Characteristics and Flow Conditions of a Forced-Vortex Impeller of an Axial Flow Fan*

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This paper deals with an experimental study of a forced-vortex impeller in an axial flow fan without inlet vanes. The impeller used is designed such as to have a forced-vortex flow at the impeller outlet on the assumption that the axial velocity component is uniformly distributed from hub to tip. Not only velocity profiles up- and downstream of the impeller but also theoretical head, loss of head and mean effective radius of the impeller, and lift and drag coefficients of each blade section are presented.

1. Introduction

Although considerable attention has been paid to examination of flow conditions in axial flow pumps and fans with free-vortex blading impellers⁽¹⁾, comparatively little is known of the flow in other blading impellers. S. Mikhail⁽²⁾ made a comparison between theoretical velocity profiles and experimental ones upstream and downstream of non-free-vortex impellers of a small axial flow fan, and presented a lift coefficient of each blade section of the rotating impeller by ignoring its drag/lift ratio, but he made no reference to flow pattern except in the design flow. K. Hatta and others⁽³⁾ presented flow conditions and rotating stall in an axial flow compressor having an impeller which was designed in such a way that the blade circulation might decrease with an increasing radius, but they made no reference to loss of head and blade characteristics on each stream surface in the impeller. J. T. Bowen and others⁽⁴⁾ compared flow conditions in a forced-vortex impeller with those in a free-vortex one of an experimental compressor with inlet vanes, but it was presumed that they were widely different from the flow conditions in an impeller of a fan without inlet guide vanes.

Although there are some questions about the similarity between a blade characteristic of a rotating impeller and its two dimensional cascade data in a wind tunnel test, it is said that both characteristics fairly agree with each other at the design point of the impeller with free-vortex blading except

near tip and hub sections⁽¹⁾. However, no work has yet been made for comparing the characteristic of a rotating blade section of a non-free-vortex impeller, in which stream surfaces are not concentric circular cylinders, with its wind tunnel test result.

This paper deals with an experimental study of a forced-vortex blading impeller of an axial flow fan having a small hub ratio, which is designed such as to have a forced-vortex flow at the impeller outlet on the assumption that the axial velocity component is uniformly distributed from hub to tip at the design flow rate. Not only the relations between overall performances and flow conditions up- and downstream of the impeller but also observed distributions of theoretical head and loss of head in the impeller, the mean effective radius of the impeller, and lift and drag coefficients of each blade section are presented.

Since a forced-vortex impeller has a small blade twist, its blade setting angle at the root does not become so large as in a free-vortex impeller in the case of adopting high outlet angle at the mean radius of the impeller blade. It is, therefore, possible in a forced-vortex impeller to take a higher outlet angle and a smaller hub ratio than a free-vortex impeller. Thus it may be possible to make the impeller diameter smaller than usual. It is an object of this paper to examine an isolated impeller characteristic and backflow conditions of such a forced-vortex impeller for putting it to practical use.

2. Experimental apparatus

Figure 1 shows a sectional elevation of an

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axial flow fan with diffusion vanes used, of which the design flow rate was $3 \text{ m}^3/\text{sec}$ and the design pressure rise 25 mmAq at the nominal rotating speed of 1450 rpm , and so its specific speed was $2000 \text{ (rpm, m}^3/\text{min, m)}$.

Figure 2 shows distributions of blade setting angle β_0 (=stagger angle), blade outlet angle β_2 , and blade solidity of the impeller, σ . A setting angle β_0 was determined such that the whirl velocity component of the discharge flow from the impeller might be a forced-vortex flow at the design flow rate while the axial velocity component might be uniformly distributed from hub to tip. The outlet angle β_2 was comparatively large and its value at the mean radius r_{2m} was 39 degrees. The aerofoil section of RAF6E was used for the impeller blades, of which there were eight. The diffusion vanes were made of flat plates of 3.2 mm in thickness, and the number of vanes was 14 . The distribution of the vane inlet angle β_{d1} is also illustrated in Fig. 2.

The layout of test circuit is shown in Fig. 3. The flow rate was measured by an orifice plate set

up in the exit duct, and the rotating speed was measured by a digital tachometer. Fluid static and total pressures were measured by the Betz projection manometers and inclined tube manometers. Since it was impossible to arrange a torque measuring instrument between the test fan and the driving motor, the input power to the motor was measured by a wattmeter instead of the input torque to the fan shaft.

Flow conditions or velocity and pressure distributions up- and downstream of the impeller were measured by Pitot tubes with yawmeters and L-type static tubes made of small stainless tubes having 1.3 mm outer diameter, which had already been calibrated in a wind tunnel before the measurement.

3. Experimental results and discussions

3.1 Overall performances and flow conditions

An overall performance of the fan used is shown in Fig. 4, where φ , ψ and τ are dimensionless values as follows:

$$\varphi = (Q/A)/r_{2m}\omega \quad \dots\dots\dots (1)$$

Fig. 1 Sectional elevation of test fan

Fig. 2 Blade angles and solidity

Fig. 3 Layout of test circuit

Fig. 4 Performance characteristics

The broken line in Fig. 6 plots the results calculated by simple radial equilibrium theory using the measured relative flow angle as described later in Fig. 9, and the agreement between the calculated and measured velocity components is nearly perfect at a higher flow rate than the design point, with the exception, of course, of the region affected by boundary layer and tip clearance. The symbol V_{2a0} in the figure is the axial velocity coefficient at the mean radius r_{2m} . Since the observed values of V_{2a} decrease rapidly near the tip, it is meaningless to make a comparison between the observed and the theoretical values of V_{2a} at the same flow rate. Therefore, the theoretical velocity profiles having the same V_{2a0} as the observed one are drawn in Fig. 6.

It is expected from the figure that the outlet backflow takes place at a higher flow rate than that of free-vortex impeller, because V_{2a} at inner radius is smaller than the one at outer radius even in the design point. However, the reason for the outlet backflow occurrence at such a very high flow rate as shown in Fig. 4 is probably due to separation in the diffusion vane near the hub as expected from the observed velocity profile near the hub⁽³⁾.

Figure 7 shows the comparison of the flow rate calculated from the observed velocity distribution with the one measured by the orifice meter, Q , in order to examine accuracies of observed values by the Pitot-tube. In the figure, Q_{p1} and Q_{p2} are the flow rates calculated from the observed velocity profiles upstream and downstream of the

Fig. 7 Comparison between flow rate obtained from velocity profiles and the one by orifice meter

impeller, respectively. It is seen from the figure that both Q_{p1}/Q and Q_{p2}/Q are nearly equal to unity except at low flow rates where the backflow takes place at the impeller inlet, and hence it is concluded that the measured values of velocity are most reliable except at a lower flow rate.

Figure 8 shows the meridional streamline shift through the impeller which is calculated between the measured stations upstream and downstream by satisfying the continuity equations graphically. The streamline is not parallel to the axial direction, but deflected radially upward toward the outer casing even at a higher flow rate than the design point unlike free-vortex impellers. The streamline deflection becomes larger with a decreasing flow rate, and the backflow takes place at inner radius of the impeller outlet and then at outer radius of the inlet with a further decreasing of flow. This phenomenon is also observed in a free-vortex impeller.

Figure 9 shows the relative flow discharge angle β_2' and the outlet deviation angle $\Delta\beta_2 (= \beta_2 - \beta_2')$. It is seen from this figure that β_2' and, therefore, $\Delta\beta_2$ become roughly constant at flow rate in higher flow region. The dash-dot line in Fig. 9 (a) was used for calculating the outlet velocity profiles which were indicated in Fig. 6 by the broken lines. The dash-dot line in Fig. 9 (b) is calculated by substituting the value given by Carter and Hughes⁽⁸⁾ into the deviation factor m in Constant's equation $\Delta\beta_2 = m\theta\sqrt{t/l}$, where θ is the deflection angle of a blade and t/l the pitch chord ratio, and it is roughly equal to the observed values except near the annular wall.

3.2 Impeller characteristics

Figure 10 shows the distributions of the theoretical head coefficient ψ_{th} and head loss coefficient Θ_i for each streamline through the impeller which are calculated from the observed values by the radial surveys. In the figure, η_i represents the isolated impeller efficiency and is given as follows:

$$\eta_i = (\psi_{th} - \Theta_i) / \psi_{th} \quad (4)$$

Even near the design flow rate, ψ_{th} is non-uniformly distributed but increases as the radius increases,

Fig. 8 Meridional streamline shift in passing through impeller

and η_i remains nearly constant except near the tip or hub of the impeller. The increase in Θ_i , and accordingly, the decrease in η_i near the casing wall, are due to the tip clearance.

Mean values of ψ_{th} , Θ_i and η_i each across the streamline which are calculated from the foregoing results except in the backflow region are shown in Fig. 11. The mean input head coefficient $\bar{\psi}_{th}$ increases linearly with a decreasing flow rate, but suddenly falls when a backflow takes place in the impeller inlet, and then also increases linearly as the flow rate decreases. This phenomenon is also observed in a diagonal flow impeller with comparatively higher outlet angles⁽⁶⁾.

The mean head loss coefficient $\bar{\Theta}_i$ gradually increases with a decreasing flow rate due to an increasing angle of attack with the blade. However, as the inflow area decreases when a backflow takes place in the impeller inlet, the inflow velocity increases with a decreasing flow rate as shown in

(a) Outlet relative flow angle

(b) Outlet deviation angle

Fig. 9 Relative flow direction at outlet

Fig. 5, and accordingly the angle of attack decreases with the flow rate. Thus, $\bar{\Theta}_i$ decreases discontinuously when the backflow takes place in the inlet as shown in Fig. 11.

There is a little change in $\bar{\eta}_i$ over a wide range of flow rates without backflow, and the highest value of $\bar{\eta}_i$ is about 92%, which is nearly equal to that of free-vortex impellers^{(6)*}. Therefore, it may be concluded that such a forced-vortex impeller as designed for uniform axial discharge velocity by ignoring the radial equilibrium condition can be put to practical use as well as a free-vortex impeller.

3.3 Mean effective radius

In order to determine the performance characteristic of an axial flow impeller analytically, it is necessary to know the blade characteristic along each streamline and to integrate the one through the entire flow section, but this method is very complicated and very unpractical for calculating the performance curve of an arbitrary impeller. Consequently, a method of calculating the performance curve by using the characteristic of a blade element at a specified radius has been adopted⁽⁹⁾. The radius is called the mean effective radius hereafter, and is defined as the one at which the theoretical

Fig. 10 Isolated impeller characteristics (1)

Fig. 11 Isolated impeller characteristics (2)

* H. D. Linhardt indicated in his report in References (1) that the maximum value of η_i was about 94% in the middle passage of a free-vortex impeller of his experimental fan.

head produced is equal to the total integrated head, that is, the mean theoretical head $\bar{\psi}_{th}$. It was indirectly revealed by comparing the overall performance curve that the mean effective radius was equal to the root mean square radius in an axial flow fan with a free-vortex impeller⁽¹⁰⁾, and it was also indicated by measured velocity distributions that in a diagonal flow impeller the effective radius was identical with the root mean square radius in both a forced- and a constant-vortex impeller as well as a free-vortex impeller⁽¹¹⁾. However, no experiments have yet been made which examine the effective radius in such an axial flow impeller with considerably small hub ratio as the impeller used here.

Figure 12 shows the effective radius ratio $\bar{R}_2$ and the ratio of input head at the root mean square radius to $\bar{\psi}_{th}$ which is indicated by the symbol Λ . It will be seen from Fig. 12 that $\bar{R}_2$ is slightly larger than unity, and so Λ is slightly smaller than unity. The discrepancy, however, being 2 or 3%, and moreover, the hub ratio of the impeller used being considerably smaller than those of usual axial flow fans and pumps, it may be concluded that the root mean square radius can be regarded as the mean effective radius even in a forced-vortex blading impeller.

Because, in the range of low flow rates at which the backflow takes place, the meridional streamlines deviate from root side to tip side as shown in Fig. 8, $\bar{R}_2$ becomes considerably larger than unity. It may, therefore, be suggested that the method of calculating the impeller performance at low flow rates by using the root mean square radius is not adequate because of large streamline shift.

3.4 Characteristics of impeller blade elements

Since the stream surfaces through the impeller used are not concentric circular cylinders even at the design point as mentioned in preceding paragraph, an attempt is made in this paragraph to compare characteristics of blade elements of a rotating impeller in a two-dimensional wind tunnel test of the same blade sections. The work done and

Fig. 12 Mean effective radius

the characteristics of each blade section are calculated from the experimental results by using the mean values of the radius and the flow velocity at points upstream and downstream of the impeller, and shown in Fig. 13, where c_L and c_D are the lift and the drag coefficients of each blade section, respectively, R_m the mean radius of the stream-surface at points upstream and downstream of the impeller. The lift coefficient c_L is uniformly distributed across the flow path at the design point, and becomes larger with a decreasing angle of attack at a low flow rate. The drag coefficient c_D remains nearly constant over a wide range of flow rates, but suddenly increases in the backflow region. The value of c_D near the impeller tip is very large under influence of the tip clearance and the boundary layers over the casing wall and the blade surfaces, thus the lift coefficient c_L becomes smaller near the tip.

Figure 14 shows the relation between c_L and angle of attack α in the region not affected by the tip clearance or the backflow, at which the thickness ratio of the blade ranges from 8 to 10%.

(a) Lift coefficient

(b) Drag coefficient

Fig. 13 Characteristics of impeller blade element

Then, in comparison with the experimental data, c_L - α characteristics of the isolated blade having the same section of 8 and 10% thickness ratio obtained by a wind tunnel test data^{(12) (13)} are also illustrated by dash-dot lines in the figure. Although no cascade data of the blade used have yet been given, it may be expected that there is little difference in characteristics between an isolated blade and a cascade having the same solidity as the impeller blade elements working in the region not affected by the tip clearance, the boundary layers and the backflow, because most solidities of the blade elements in the region are nearly equal to or lower than unity. It is seen from Fig. 14 that the observed value of c_L is in substantial agreement with the wind tunnel test data at the design flow, but is smaller at lower flow rates, especially in the inlet backflow range. The reason for this is probably a remarkable increasing of streamline shift owing to the inlet backflow, because the real angle of attack of the relative flow with the blade at this flow rate is considerably smaller than the one calculated on the assumption that the fluid flows over a concentric circular cylinder surface having the mean radius upstream and downstream of the impeller as mentioned above.

The observed values of c_D are shown in Fig. 15, where the dash-dot lines are the wind tunnel test data similar to Fig. 14. It may be seen from this figure that the observed values of c_D are spread over a certain area but nearly the same as in the wind tunnel test except in inlet backflow range. From the above results, it may be concluded that c_L and c_D determined by wind tunnel tests can be adopted to design the blade of an axial flow impeller in which small streamline shift occurs at the design

Fig. 14 Comparison of lift coefficient

Fig. 15 Comparison of drag coefficient

point.

For the relations between inlet backflow occurrence and stall of the blade element near the impeller tip, Fig. 16 shows the change in angle of attack with the blade elements near the casing wall which are calculated from the velocity measurements at the nearest position of the casing wall ($R_m = 1.33$). In the figure, the flow rate at which ψ - ϕ curve attains the maximum value is indicated by the dash-dot line, and the backflow region and the isolated blade stall zone obtained from a wind tunnel test are also illustrated. It will be seen from Fig. 16 that the maximum point of ψ - ϕ curve is in fairly good agreement with the flow rate at which the angle of attack with the blade near the tip reaches its stall angle, and it is higher than the flow rate of inlet backflow occurrence point. It may, therefore, be suggested from this figure that ψ - ϕ curve begins to fall by the stall of the blade near the tip, and further decreases with an increase in the stall region at a low flow rate⁽¹⁴⁾. Because the fluid energy given by the blade element near the tip decreases remarkably with a further increase in the stall, the fluid cannot flow toward the impeller exit having a high energy fluid but goes back to the suction duct.

The relations between the flow rate and the

Fig. 16 Angle of attack near impeller tip

Fig. 17 Angle of attack near impeller root

angle of attack with the blade elements near the hub are shown in Fig. 17, where the solid line is obtained from the observed velocity at the nearest point to the hub ($R_m=0.44$), and the dash-dot line is estimated by extending the tendency of the measured velocity to the hub surface ($R_m=0.375$). It may be seen from Fig. 17 that the backflow at the hub of the impeller outlet begins to take place when the angle of attack with the blade near the root becomes considerably higher than its stall angle. Therefore, if the backflow takes place due to a stall of the blade, it seems that the stall of the blade element near the hub occurs much later than the flow to the blade in a wind tunnel. It is said that the main reason for this is probably due to the thinner boundary layer over the blade surface near the root owing to a centrifugal effect of the fluid⁽¹⁵⁾. It is, however, doubtful that the backflow is caused by a stall of the blade, because the backflow occurrence with a decreasing flow rate is a logical result from the radial equilibrium equation for an impeller discharge flow and has no relation to the stall⁽¹⁶⁾.

4. Conclusions

In the present paper, experimental results of flow conditions and isolated impeller characteristics, that is, theoretical head, loss of head, mean effective radius, and aerodynamic characteristics of blade elements of a rotating impeller of an axial flow fan having a small hub ratio are described. The impeller used is designed such as to have a forced-vortex flow at the outlet on the assumption that the axial velocity component is uniformly distributed from hub to tip.

The following conclusions are derived from this experiment:

(1) In the forced-vortex impeller used, the axial velocity component at the impeller outlet is non-uniformly distributed but increases with an increasing radius even at the design point, while the whirl velocity component is uniformly distributed except near the annular wall, that is, the flow is a non-forced-vortex one.

(2) There is little difference in the isolated impeller efficiency between the forced-vortex impeller used and an ordinary free-vortex one.

(3) In spite of the considerably smaller hub ratio, the mean effective radius of the impeller used is nearly equal to its root mean square radius except in a low flow range where the inlet backflow takes place.

(4) In a higher flow range having a small radial shift of meridional streamline, the lift coefficient of the working impeller blade element is in fairly good agreement with its two-dimensional data in a wind tunnel test, but it becomes lower at a low flow rate. The drag coefficient of the impeller blade element is nearly equal to that of a wind tunnel test except in the inlet backflow range.

(5) The maximum point of P_T - Q curve is essentially in agreement with the flow rate at which the angle of attack with the blade element near the impeller tip reaches its stall angle.

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