A Study of Cross Flow Fan with Inner Guide Apparatus*

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This paper concerns a theoretical and experimental study of cross flow fan with inner guide apparatus. Effects of fan geometry on the performance are investigated.

Theoretically, the inner apparatus provides a shockless flow on the inner periphery of the impeller, which has an arbitrary inner blade angle.

A series of experiments have been conducted using four types of impellers with different inner blade angles β_1 and inner vanes, whose shapes are identical to theoretical stream lines. One apparatus designed with a single vane and an impeller of $\beta_1=110^\circ$ produced a high and stable performance curve sloping downwards to the flow rate.

1. Introduction

A cross flow fan invented by Mortier (1892) had a guide apparatus inside the impeller. After this, however, attention has been directed to a fan without a guide apparatus. Inner guide apparatus can give higher values of total pressure, flow rate and efficiency^{(1),(2)}, and can control the flow rate. However it seems that no systematic study has been reported so far about the fan configurations including the inner guide apparatus.

The authors found out a method of determining the potential flow inside a cross flow impeller, and investigated the influence of fan configurations on its performance. The experiments were carried out using impellers of different inner blade angles and casings of various shapes. Furthermore, the velocity distribution inside the impeller was measured. Very good agreement was confirmed between the calculated values of flow velocity with the assumption of an inviscid and incompressible flow and the measured ones of experiments.

2. Symbols

α : parameter representing the position of the vortex centre (Fig.1).
 c_r, c_θ : radial and peripheral component of the absolute velocity.
 D_2, L : outer diameter and width of the

impeller.
 H, Q : head and flow rate of the fan.
 l, t : chord length and pitch of the blade.
 u : peripheral velocity of the impeller.
 W : complex potential of the flow inside the impeller.
 α : angle of the delivery arc of the impeller.
 β_1, β_2 : inner and outer blade angles (Fig.1).
 β' : parameter representing the shape of the guide vane.
 η : efficiency.
 θ_0 : parameter representing the rotated angle of the guide apparatus (Fig.3).
 θ_c, θ_t : angular coordinates corresponding to the leading edges of the casing and the tongue (Fig.3).
 θ_g : parameter representing the size of the guide vane (Fig.3).
 ϕ : flow rate coefficient (eq.(19)).
 ψ : pressure coefficient (eq.(19)).
 ψ : stream function.
Subscripts 1, 2 : inner and outer periphery of impeller.

3. Theoretical analysis of the flow

Theoretical studies of cross flow fans without the inner guide apparatus were reported^{(3)~(5)} concerning only the case of blade angle β_1 set at 90° with the assumption of a shockless flow on the inner periphery of impeller. With the inner guide apparatus, however, a shockless flow can be realized for an arbitrary value of β_1 because the angular momentum is applied to the inside flow by the guide apparatus. Here, the potential flow inside the impeller is analyzed, assuming that the fluid is inviscid and incompressible and that the impeller has an infinite number of blades, that is, the fluid flows along the blade of the impeller.

3.1 The flow inside the impeller

With the assumption of a shockless flow on the inner periphery of impeller, we obtain the equation

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$$c_{r1}/(c_{\theta1}-u_1) = \tan \beta_1$$

hence $c_{r1} - c_{\theta1} \tan \beta_1 = -u_1 \tan \beta_1$ (const) (1)

where c_{r1} and $c_{\theta1}$ are the radial and peripheral components of absolute velocity on the inner periphery respectively, and u_1 is the inner peripheral velocity of the impeller and the angle β_1 is the inner blade angle defined as shown in Fig.1.

Fig.1 Analytical model of flow

Let W be a complex potential of the flow inside the impeller, z be a complex coordinate with the origin at the centre of impeller and $i^2 = -1$. Now we consider the flow of the complex potential W' given by

$$W' = z(i + \tan \beta_1) dW/dz \quad (2)$$

On the inner periphery $r=r_1$, W' becomes

$$(W')_{r=r_1} = r_1 \{ (c_{r1} \tan \beta_1 + c_{\theta1}) + i(c_{r1} - c_{\theta1} \tan \beta_1) \} \quad (3)$$

where the relation

$$(dW/dz)_{r=r_1} = (v_x - i v_y)_{r=r_1} \\ = (c_{r1} \cos \theta - c_{\theta1} \sin \theta) - i(c_{r1} \sin \theta + c_{\theta1} \cos \theta)$$

is substituted. Since the imaginary part of eq.(3) is constant from eq.(1), the circle $r=r_1$ is a stream line of the flow W' . Therefore, by obtaining the complex potential W' with the circle $r=r_1$ as a stream line, and transforming W' to W using eq.(2), the complex potential W of a flow inside the impeller with an arbitrary constant value of inner blade angle β_1 can be determined.

As the circulation along the inner periphery of a cross flow impeller does not vanish⁽⁶⁾, we consider the flow with a vortex of strength Γ inside the impeller as shown in Fig.1. Assume that the vortex passes through the x -axis and is located at the distance ar_1 ($0 \leq a \leq 1$) from the centre of impeller, where the origin of the coordinate system is at the centre of the impeller. And let W be the complex potential of the inside flow represented by the equation

$$W = W_0 + W_1 \quad (4)$$

where W_0 is the complex potential of a flow caused by the vortex with strength Γ located at $z=ar_1$, that is

$$W_0 = -\frac{i\Gamma}{2\pi} \log_*(z - ar_1) \quad (5)$$

and W_1 is an additional potential with no singularity in the region $|z| < r_1$ which satisfies the boundary condition (1).

Transform W , W_0 and W_1 into W' , W'_0 and W'_1 by eq.(2), then we have $W' = W'_0 + W'_1$. Now W'_0 becomes

$$W'_0 = z(i + \tan \beta_1) dW_0/dz = \frac{\Gamma}{2\pi} (1 - i \tan \beta_1) \left(1 + \frac{ar_1}{z - ar_1} \right)$$

which represents a flow with a doublet at $z=ar_1$. Therefore, W' must be a flow with the doublet at $z=ar_1$, and must have the circle $r=r_1$ as a stream line. From the principle of images, W'_1 must have a doublet at $z=r_1/a$, then it becomes

$$W'_1 = z(i + \tan \beta_1) dW_1/dz = -\frac{\Gamma}{2\pi} (1 + i \tan \beta_1) \frac{r_1/a}{z - (r_1/a)} + C$$

The second term C of the right hand side is a complex constant and never affects the flow of W'_1 , but it represents the source and circulation at $z=0$ in the flow of W_1 . Integrating the above equation with respect to z , we obtain the equation

$$W_1 = -\frac{\Gamma}{2\pi} \frac{1}{(i + \tan \beta_1)} \left\{ (1 + i \tan \beta_1) \log_* \left(z - \frac{r_1}{a} \right) - (1 + i \tan \beta_1 + \frac{2\pi C}{\Gamma}) \log_* z \right\}$$

where the integration constant is removed.

As W_1 has no singularity in the region $|z| < r_1$, C must be

$$C = -(\Gamma/2\pi)(1 + i \tan \beta_1)$$

Thus

$$W_1 = -\frac{\Gamma}{2\pi} (\sin 2\beta_1 - i \cos 2\beta_1) \log_* \left(z - \frac{r_1}{a} \right) \quad (6)$$

Substituting eqs.(5) and (6) into eq.(4), W is obtained. The value of Γ is determined such as to satisfy eq.(1), but it can easily be obtained from the circulation along the circle $r=r_1$, and then we have

$$\Gamma = 2\pi r_1 u_1 \quad (7)$$

Therefore W becomes

$$W = -r_1 u_1 \left\{ i \log_* (z - ar_1) + (\sin 2\beta_1 - i \cos 2\beta_1) \log_* \left(z - \frac{r_1}{a} \right) \right\} \quad (8)$$

Above equation represents a flow which has a vortex inside the impeller at $z=ar_1$ and a vortex and a source outside the impeller at $z=r_1/a$, satisfying the boundary condition (1) on the inner periphery of impeller.

As no friction acts on the fluid, the force applied to the fluid on the circle $r=r_1$ is perpendicular to the circle, then no angular momentum exists in the interior fluid in the case without an inner guide apparatus. If the fluid has an angular momentum, there must be a guide apparatus inside the impeller. Denoting the circle $r=r_1$ by C and the angular momentum applied to the fluid inside the circle $r=r_1$ by M , we get from Blasius' formula

$$M = \Re \left[\frac{\rho}{2} \int_C \left(\frac{dW}{dz} \right)^2 z dz \right] = 2\pi \rho (r_1, u_1)^2 \frac{a^2}{1-a^2} \sin 2\beta_1 \quad (9)$$

where $\Re[]$ means the real part of the value inside the bracket $[]$. In case of $\beta_1 = 90^\circ$, M vanishes, so the inner guide apparatus is not required. But in case of $\beta_1 < 90^\circ$, the inner guide is required in order to apply the angular momentum to the fluid. It is apparent from eq.(9) that the direction of momentum is reverse to the rotation of impeller for $\beta_1 > 90^\circ$.

From eq.(8), the stream function becomes

$$\Psi = -\frac{r_1 u_1}{2} \left\{ \log_e (a^2 r_1^2 - 2a r_1 r \cos \theta + r^2) - \cos 2\beta_1 \log_e \left(\frac{r_1^2}{a^2} - 2 \frac{r_1 r}{a} \cos \theta + r^2 \right) - 2 \sin 2\beta_1 \tan^{-1} \frac{r \sin \theta}{(r_1/a) - r \cos \theta} \right\} \quad (10)$$

Radial and peripheral velocities c_r and c_θ on the circle $r=r_1$ are

$$\left. \begin{aligned} c_r &= -2u_1 a \sin \beta_1 \frac{\cos \beta_1 (a - \cos \theta) + \sin \beta_1 \sin \theta}{1 + a^2 - 2a \cos \theta} \\ c_\theta &= u_1 + c_r \cot \beta_1 \end{aligned} \right\} \quad (11)$$

The angular coordinates θ where c_r vanishes on the inner periphery of impeller, are given by

$$\left. \begin{aligned} \theta_i &= -\beta_1 + \cos^{-1}(a \cos \beta_1) \\ \theta_c &= 2\pi - \beta_1 - \cos^{-1}(a \cos \beta_1) \end{aligned} \right\} \quad (12)$$

where $\cos^{-1}(a \cos \beta_1)$ is to take its principal value. In the region $\theta_c > \theta > \theta_i$, c_r becomes negative (suction region), whereas in the region $\theta_i > \theta > \theta_c - 2\pi$, c_r is positive (delivery region). Therefore, in order to avoid the reverse flow, the leading edges of the casing must be located at angle θ_i and θ_c given by eqs.(12). Examples of calculated values of θ_i and θ_c are shown in Fig.2.

Fig.2 Angles θ_i and θ_c related to parameter α

When $\beta_1 > 90^\circ$, the angle of delivery arc becomes larger than that of suction arc, and the difference between these arcs increases as the parameter α approaches one. On the contrary, when $\beta_1 < 90^\circ$, the flow is accelerated through the impeller. Therefore, it seems that the performance is reduced for smaller values of β_1 because of the difficulty to transform the velocity head into a pressure head.

3.2 The flow in suction region

In order to calculate the characteristics of the impeller, the velocity distribution along the outer periphery must be obtained. At first, the flow in suction region is considered. Since the flow in this region should be a potential flow, the velocity depends on the flow inside the impeller and the casing configuration. Therefore, a shockless inflow on the impeller is not always realized, and then the velocity distribution on the outer periphery cannot easily be obtained. As an example, a simple casing with two lines of $\theta = \theta_i$ and $\theta = \theta_c$ is taken in this analysis.

Let the stream function be Ψ , and the differential equation of the potential flow in suction region is

$$\frac{\partial^2 \Psi}{\partial r^2} + \frac{1}{r} \frac{\partial \Psi}{\partial \theta} + \frac{1}{r^2} \frac{\partial^2 \Psi}{\partial \theta^2} = 0 \quad (13)$$

and boundary conditions are

$$\left. \begin{aligned} \Psi &= \text{const} & \text{when } \theta = \theta_i \text{ and } \theta_c \\ \Psi &= k(\theta - \theta_i)/(\theta_c - \theta_i) & \text{when } r = \infty \end{aligned} \right\} \quad (14)$$

and from the equation of continuity on the outer periphery $r=r_2$

$$(\Psi)_{r=r_2} = -\frac{r_1 u_1}{2} \left\{ (1 - \cos 2\beta_1) \log_e (1 + a^2 - 2a \cos \theta) - 2 \sin 2\beta_1 \tan^{-1} \frac{a \sin \theta}{1 - a \cos \theta} \right\} \quad (15)$$

where k in eq.(14) is a constant and will be given by eq.(17).

The solution of eqs.(13) satisfying the condition (14) can be expressed by

$$\Psi = k \frac{\theta - \theta_i}{\theta_c - \theta_i} + \sum_{n=1}^{\infty} \frac{k_n}{r^n} \sin n\pi \frac{\theta - \theta_i}{\theta_c - \theta_i} + C \quad (r \geq r_2) \quad (16)$$

where the constants k and $k_n (n=1, 2, \dots)$ are obtained from eq.(15), and the constant C can be neglected because it never affects the flow. Then

$$k = -2r_1 u_1 \left\{ (1 - \cos 2\beta_1) \log_e ((\sqrt{1 - a^2 \cos^2 \beta_1} + a \sin \beta_1) / \sqrt{1 - a^2}) + \sin 2\beta_1 \sin^{-1}(a \cos \beta_1) \right\}$$

$$k_n = \frac{2r_1 u_1}{\theta_c - \theta_i} \int_{\theta_i}^{\theta_c} f(\theta) \sin n\pi \frac{\theta - \theta_i}{\theta_c - \theta_i} d\theta \quad (n=1, 2, \dots) \quad (17)$$

where

$$f(\theta) = -\frac{r_1 u_1}{2} \left\{ (1 - \cos 2\beta_1) \log_e (1 + a^2 - 2a \cos \theta) - 2 \sin 2\beta_1 \tan^{-1} \frac{a \sin \theta}{1 - a \cos \theta} \right\} - k \frac{\theta - \theta_i}{\theta_c - \theta_i} + r_1 u_1 \left\{ (1 - \cos 2\beta_1) \log_e (\sqrt{1 - a^2 \cos^2 \beta_1} - a \sin \beta_1) - \sin 2\beta_1 \sin^{-1}(a \cos \beta_1) \right\}$$

Substituting eqs.(17) into eq.(16), the stream function in suction region is obtained.

3.3 The flow in delivery region

Since the velocity distribution is not uniform on the outer periphery of the cross flow impeller, the total head given by the impeller differs for different points on the impeller outer periphery, so the flow is not a potential one in the delivery region. However, the velocity distribution on the outer periphery can easily be obtained from the inside flow and the outer blade angle β_2 , assuming that the fluid flows along the blade.

The total head given by the impeller can be obtained merely from the velocity on the outer periphery. In the delivery region $\theta_c \leq \theta \leq 2\pi - \theta_i$, the radial and peripheral components of the absolute velocity c_r and c_θ on the outer periphery of the impeller are

$$\left. \begin{aligned} c_r &= \frac{r_1}{r_2} c_{r1} = -2a u_1 \frac{r_1}{r_2} \sin \beta_1 \frac{\cos \beta_1 (a - \cos \theta) + \sin \beta_1 \sin \theta}{1 + a^2 - 2a \cos \theta} \\ c_\theta &= u_2 + c_r \cot \beta_2 \end{aligned} \right\} \quad (18)$$

3.4 Theoretical performance and discussions

As mentioned above, the Euler head can be calculated. These values differ from the actual ones but serve as the basis to estimate the losses and as the clues to clarifying the effects of inner guide's configuration and others. To represent the fan performance, the flow rate coefficient φ and the pressure coefficient ψ are defined as follows,

$$\varphi = Q/(LD_2 u_2) \quad \text{and} \quad \psi = H/(u_2^2/2g) \quad (19)$$

where Q is the flow rate, L is the impeller width, D_2 is the outer diameter of impeller, u_2 is the outer peripheral velocity of impeller and H is the Euler head given by the impeller.

In the foregoing discussion, the location of leading edges of the casing are considered at the points where the radial velocities on the outer periphery are zero ($c_r=0$). But no actual flow satisfies the above assumption and leading edges locate at the positions of $\theta=\theta_i'$ and θ_e'' as shown in Fig.3. The flow of the suction region in section 3.2 was analyzed for the linear casings as shown by broken lines in Fig.3.

A cross flow fan has two casings; the one near the vortex is called "the tongue" and the other simply "the casing". In the region near the tongue, $\theta=\theta_i' \sim \theta_i''$, a recirculating flow occurs, and in the region near the casing, $\theta=\theta_e' \sim \theta_e''$, the fluid leaks out from the casing. So the impeller is not effective in the region $\theta=\theta_i' \sim \theta_i''$ and $\theta_e' \sim \theta_e''$. Therefore the flow rate coefficient ϕ and the pressure coefficient ψ are given by the equations

$$\varphi = \frac{1}{2r_2 u_2} [(\psi)_{\theta=\theta_i'} - (\psi)_{\theta=\theta_e'}]_{r=r_1}$$

$$\psi = \frac{1}{\varphi} \left(\frac{r_1}{r_2} \right)^4 \left[\int_{\theta_i'}^{\theta_e'} c_{r1}^* \left(\sum_{n=1}^{\infty} n k_n^* \sin n\pi \frac{\theta - \theta_i}{\theta_e - \theta_i} - c_{\theta 1}^* \right) d\theta + \int_{\theta_e'}^{\theta_e''} c_{r1}^* \left(\frac{r_2^2}{r_1^2} + c_{r1}^* \cot \beta_2 - c_{\theta 1}^* \right) d\theta \right] \quad (20)$$

where c_{r1}^* , $c_{\theta 1}^*$ and k_n^* are dimensionless constants represented by $c_{r1}^* = c_{r1}/u_1$, $c_{\theta 1}^* = c_{\theta 1}/u_1$ and $k_n^* = k_n/(r_1 u_1 r_2^n)$. The pressure coefficient ψ is represented as the mean value by the flow rate because the total head differs for each stream line.

Fig.4 shows the ψ - ϕ curves calculated from eq.(20) for various values of a and θ_0 and for $D_1/D_2=1$, $\beta_2=25^\circ$ and $\beta_1=90^\circ$ and 110° , where $\theta_0=\theta_i' - \theta_i$ indicates the angle between the leading edge of the actual tongue and that of the ideal tongue. In calculation, the angle of suction region is set at a value given by Fig.2.

It can be seen from Fig.4 that the pressure coefficient increases as a increases or θ_0 decreases, that is, as the vortex moves toward the inner periphery of impeller or the recirculating flow decreases. Therefore, a guide apparatus which produces the minimum recirculating flow and generates a vortex near the inner periphery is recommended.

In controlling the flow rate with the guide apparatus, a peripheral movement of the apparatus is more desirable than a radial movement, because the slope of ψ - ϕ curve obtained for the peripheral movement is more real than that obtained for the radial movement. Further, for peripheral movement

of vortex, the flow pattern inside the impeller does not change and only it rotates as a whole, which means that the shape of the guide apparatus is the same at all time, and thus it suits our convenience.

The flow in a cross flow fan is not uniformly distributed; the velocities are large near the tongue and small near the casing. This flow pattern indicates that the peripheral velocity must have a direc-

Fig.3 Fan configuration

Fig.4 Calculated performance curves ($D_1/D_2=1$, $\beta_2=25^\circ$)

Fig.5 Experimental apparatus

tion contrary to the impeller rotation near the tongue in the suction region, and such a high performance as shown in Fig.4 cannot be obtained on account of the difficulty to generate the shockless in-flow. Equation (20) shows, furthermore, that the performance decreases as D_1/D_2 becomes small. For these reasons, there are large differences between actual performances and theoretical ones.

4. Experimental apparatus, results and discussions

4.1 Experimental apparatus

The experimental apparatus is shown in Fig.5. The impeller is supported and driven on one side and fixed with a ring plate on the other free side of blades, so that the guide apparatus can be inserted inside the impeller through the circular opening of the side plate. The guide apparatus is made of a circular PVC-plate fitted with a curved brass plate vane of which the shape is identical to the theoretical stream line calculated in section 3.1. Many taps are set on both the circular plate and the side plate, which are used for measuring the flow velocity by the yaw-probe and are sealed by caps when they are not used. The guide apparatus can be rotated at 7.5° intervals around the centre of the impeller. The impeller specifications are as follows:

Outer diameter $D_2=240\text{mm}$, inner diameter $D_1=200\text{mm}$, width $L=100\text{mm}$ and number of blades $z=36$, so diameter ratio is $D_1/D_2=0.83$ and width ratio is $L/D_2=0.42$.

Although the optimum value of the outer blade angle β_2 may be determined for each value of inner blade angle β_1 , an outer blade angle of $\beta_2=25^\circ$ is chosen in this work, which is optimum in case of no inner guide apparatus⁽⁷⁾. Four impellers with the blades bent as circular arc are used in this work, and their inner blade angles are 70° , 90° , 110° and 130° (pitch chord ratio $t/l=0.78$, 0.88 , and 1.02).

Most of experiments were carried out at an impeller rotating speed of $n=820\text{rpm}$, but efficiency was measured at $n=1400\text{rpm}$. For these two speeds the performance had sufficiently good similarity.

Thermo-couple type anemometer was used for measurement of flow velocity in the duct. The flow inside the impeller was measured by means of a three-hole yaw probe⁽⁸⁾ through the measuring taps.

Although fan performance depend greatly on the shape of the tongue, we adopted the tongue recommended by Murata et al.^{(9),(10)} as shown in Fig.3,

because it gave superior performance without an inner guide apparatus, and was confirmed to be favourable in this work. The experiments were carried out keeping this tongue fixed.

In Fig.6, the solid line shows the $\psi-\phi$ curve for the fan of $\beta_1=90^\circ$ impeller without an inner guide apparatus, and with casings similar to those used by Murata et al., and the broken line shows the $\psi-\phi$ curve obtained by Murata et al. There are large differences between the two performance curves in Fig.6. The impeller used in this work has a fairly smaller value of L/D_2 than that of Murata et al. So it seems that the drop in the performance was due to the smaller value of L/D_2 as reported by Коровкин⁽¹¹⁾. But it is difficult structurally to extend the width of impeller due to inner guide apparatus.

4.2 Influence of inner guide apparatus

Fig.7 shows as example of inner guide vane, which are identical to the theoretical stream lines for the blade angle $\beta_1=110^\circ$ and the parameter $\alpha=0.98$. Stream lines for $\beta_1=90^\circ$ are concentric circles centre of which is at the vortex, and those for $\beta_1=70^\circ$ have a configuration just reverse to the one in the Fig.7. Further,

Fig.6 Performance curves without inner guide apparatus

Fig.7 A shape of guide vane

Fig.8 An example of performance curves

Impeller $\beta_1=110^\circ$
Guide vane $\alpha=0.98$, $\beta_1=110^\circ$, $\theta_0=72^\circ$ (Fig. 7)
Casing B-2 (Fig. 12)
Rotating speed $n=1400\text{rpm}$

Fig.9 Influences of inner blade angle β_1 and shape of guide vane β'

the vanes of $\alpha=0.7$ were designed. Fan performances were investigated by rotating the guide apparatus with various shapes and numbers of vanes about its centre. The size of inner vane is represented by the central angle θ_g (Fig.7), and its shape by the parameter α and angle β' the same as the inner blade angle β_1 used in calculation of stream lines. The rotated angle of inner vane is represented with the central angle θ_0 between the leading edge of the tongue and the central line of the guide vane as shown in Fig.3. It was found through the experiments that the best performance was obtained for a fan with an impeller of $\beta_1=110^\circ$ and only one inner vane of $\alpha=0.98$, $\beta_1=110^\circ$ and $\theta_g=72^\circ$ shown in Fig.7. Fig.8 shows performance curves for various values of θ_0 , where the broken line shows a case without inner guide. By adequate rotation of the inner apparatus, the fan performance can be obtained as the envelope of ψ - ϕ curves in Fig.8(a), having the pressure coefficient $\psi=4$ at the flow rate coefficient $\phi=0.5$ and $\psi=2.4$ at $\phi=1.0$. The efficiencies are shown in Fig.8(b), where the chain line corresponds to this envelope. If the guide apparatus is properly rotated around its centre, the cross flow fan will have a high ψ - ϕ curve sloping downward as the flow rate coefficient ϕ increases, and have a higher efficiency than 40%.

Fig.9 shows the performance curves for blade angles $\beta_1=70^\circ$, 90° and 110° with the same casing as shown in Fig.9(a). The measurements were carried out in a wide range of the rotational angles θ_0 of the guide apparatus and for various shapes of guide vane β' . The performance curves shown in these figures were the best ones for $\beta_1=70^\circ$, 90° and 110° respectively. The impeller of $\beta_1=130^\circ$ has an extremely inferior performance as shown in Fig.10 and is not practical. It seems from these figures that the best performance is obtained for $\beta_1=110^\circ$ and the shape of the guide vane slightly affects on the performance.

There is little difference between the performance with a single vane of $\theta_g=60^\circ$ and that with one of $\theta_g=72^\circ$, but the flow occasionally becomes unstable for the vane of $\theta_g=60^\circ$. In the case of $\theta_g < 60^\circ$, the flow becomes unstable, and for $\theta_g > 72^\circ$ the flow becomes stable but reduces the performance.

Fig.11 shows the performance curves with two guide vanes and the same casing as in Fig.8; a fan with two guide vanes has a lower performance than one with a single vane. Бычков et al.⁽¹⁾ reported that the performance was improved by increasing the number of guide vanes, but this result was obtained for a fixed guide apparatus, so it seems that in a particular flow condition a fan with more inner guide vanes can achieve a higher performance of the vanes confor-

Fig.10 Performance curves for $\beta_1=130^\circ$

Fig.11 Performance curves with two guide vanes

ming to the ideal flow.

Experiments were carried out using guide vanes of thicknesses 0.5mm and 1.0mm, but the performance was little affected.

Laakso⁽¹²⁾ reported that a high pressure coefficient $\psi > 4$ is obtained for a fan with a single flat guide plate. The same guide plate was applied to the fan used in this work, but good performance was not obtained. Бычков et al.⁽¹⁾ reported that good performance was not obtained for a fan with a flat guide plate the same as the result of our work.

4.3 Influence of the shape of casing and the size of delivery arc

As mentioned in the section 3.1, the delivery arc is considered the periphery having a positive radial velocity, but it varies with the inner blade angle and the location of the vortex. Therefore, experiments were carried out with various central angles α between leading edges of the tongue and the casing. Although the theoretical prediction indicates a monotonously increasing value of α as β_1 , the experimental results show the optimum value of α in the range of $180^\circ \sim 200^\circ$ independent of β_1 . In the actual flow, the tongue is

not at the point of zero radial velocity, so a recirculating flow exists in the delivery region as mentioned in section 3.4. Therefore it is desirable to have the delivery arc wider than the suction arc.

The performance of cross flow fan depends greatly on the casing shape, so the influence of the shape was investigated using a casing with the delivery arc $\alpha=180^\circ$ and 200° . The casing used in experiments was composed of a circular arc and a straight line as shown in Fig.12. The fan with the impeller $\beta_1=110^\circ$ and the casing B-2 had the best performance, as shown in Fig.8. The dimensions of the casing B-2 are shown in Fig.3. A fan with the impeller $\beta_1=110^\circ$ and the casing A-4 also gives good performance, the same as the one with $\beta_1=110^\circ$ and B-2 shown in Fig.8.

4.4 The flow inside impeller

For comparison of the flows with and without an inner guide apparatus, the flow velocity was measured in the middle section of the impeller by means of a three-hole

yaw probe. In this measurement, a fan with the impeller of $\beta_1=90^\circ$ was used, because the flow became unstable for $\beta_1=110^\circ$ without an inner guide vane. The casing of A-4 and the inner guide vane of $\alpha=1.0$, $\beta'=90^\circ$ and $\theta_g=80^\circ$ were used. As shown in Fig.13, the stream line inside the impeller is almost the same for the full opening of the damper regardless of the presence of the guide apparatus, that is, a vortex occurs at a point near the inner periphery of the impeller about an angle 30° apart from the tongue. With the inner guide apparatus, the flow pattern does not change so much with the flow rate and the velocity diminishes as a whole, whereas without the guide, as Porter and Markland reported⁽¹³⁾, the vortex moves toward the casing along the inner periphery of impeller, but the strength of the vortex does not so much change, and a reduction of flow is caused by an increased recirculating flow in the delivery region.

In general, the flow inside a cross flow impeller consists of a forced vortex and a free vortex around it. Therefore the fan performance is improved if the forced vortex zone becomes small and stable. The guide apparatus serves for this purpose. If the position of the inner guide apparatus is so adjusted as to meet the flow rate, the flow outside the inner guide becomes a stable free vortex flow. Unless the position of the guide is not so adjusted, the radial velocity falls down as shown in Fig.14, so both coefficients ψ and ϕ become smaller than those without a guide apparatus. But in the case without an inner guide apparatus, the position of the vortex is influenced by the tongue and casing, and

the performance is generally lower than in the case with a guide on account of an increased recirculating flow.

Fig.14 shows the calculated values of radial and peripheral velocities on the circle $r=0.9r_1$ for the impellers $\beta_1=90^\circ$ and 110° , where c_r^* and c_θ^* are the dimensionless values of c_r and c_θ divided by u_1 . For comparison, the measured velocities are also shown in this figure.

It seems that the calculated values are in fairly good agreement with the measured ones for full opening of the damper. The measured values of peripheral velocity are smaller and those of radial velocity are somewhat larger than the calculated values in the suction region. This shows a less turning of the flow in passing through the suction side. This is the reason why the actual performance is inferior to the calculated one as shown in Fig.4. In addition, Fig.14 shows the measured values of radial velocity at $r=0.9r_1$. The figures show that with an inner guide vane the radial velocity decreases as a whole with the flow rate.

5. Conclusions

A simple method was proposed to obtain

Fig.14 Velocity distributions on the circle $r=0.9r_1$ inside impeller

the potential flow inside the impeller with an arbitrary inner blade angle. The method was applied to calculate a flow in a cross flow fan.

For a cross flow fan with an inner guide apparatus, it was verified theoretically that a shockless inflow could be achieved on the inner periphery of impeller even if the inner blade angle β_1 was not equal to 90° .

In this study, the fan with an inner guide apparatus was shown to give an impeller of $\beta_1=110^\circ$ and $\beta_2=25^\circ$.

Inner guide apparatus having only one vane of $\theta_g=72^\circ$ similar to the stream line gave the best performance.

With adequate control of inner guide apparatus, a high and stable performance curve was obtained which had the slope downwards to abscissa and a pressure coefficient $\psi=4$ at flow rate coefficient $\phi=0.5$ and $\psi=2.4$ at $\phi=1.0$.

The measured flow velocity inside the impeller showed that in the presence of the inner guide the reduction of flow rate was caused by a reduced velocity, while in the absence of the guide it was caused by an increased circulating flow due to the peripheral movement of the vortex.

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