ANALYSES ON FLOW FIELDS AND PERFORMANCE OF A CROSS-FLOW FAN WITH VARIOUS SETTING ANGLES OF A STABILIZER

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A cross-flow fan is generally used on the region within the low static pressure difference and the high flow rate. It relatively makes high dynamic pressure at low rotating speed because a working fluid passes through an impeller blade twice and blades have a forward curved shape. At off-design points, there are a rapid pressure head reduction, a noise increase and an unsteady flow. Those phenomena are remarkably influenced by the setting angle of a stabilizer. Therefore, it should be considered how the setting angle of a stabilizer affects on the performance and the flow fields of a cross-flow fan. It is also required to investigate the effect of the volumetric flow rate before occurring stall. Two-dimensional, unsteady governing equations are solved using a commercial code, STAR-CD, which uses FVM. PISO algorithm, sliding grid system and standard k - ε turbulence model are also adopted. Pressure and velocity profiles with various setting angles are graphically depicted. Furthermore, the meridional velocity profiles around the impeller are plotted with different flow rates for a given rotating speed.

Keywords: Archimedes Spiral, Cross-Flow Fan, Stabilizer, Setting Angle, Sliding Grid System.

NOMENCLATURE

b	width [mm]
C	flow velocity [m/s]
D	impeller diameter [mm]
M	Mach number
N	revolution per minute [rpm]
Q	volumetric flow rate [m^3/min, CMM]
U	circumference velocity of impeller [m/s]

GREEK SYMBOLS

β	blade angle of impeller [deg.]
ε_r	gap between the impeller and rearguider
ε_s	gap between the impeller and stabilizer
θ_{es}	setting angle of stabilizer [deg.]

SUBSCRIPTS

2	outlet of impeller
3	basic circle of rearguider
d	design point,
m	radial direction
r	rearguider
s	stabilizer, constant pressure
θ	tangential direction

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1. INTRODUCTION

A cross-flow fan is also comprised in the blower, which has generally pressure difference below 10 mAq. It consists of an impeller, a stabilizer and a rearguider. When it is applied to an air conditioner, heat exchager should be added (see Fig. 1). The cross-flow fan with those elements has been used in the wide range of industries: ventilating devices in mining, building, automobile, etc. It is recently adopted in the indoor units of an air conditioner as an appliance. Fluids pass through an impeller blade twice and the impeller has a large absolute flow velocity because of a forward curved blade. This can reduce the rotating speed to achieve the equal

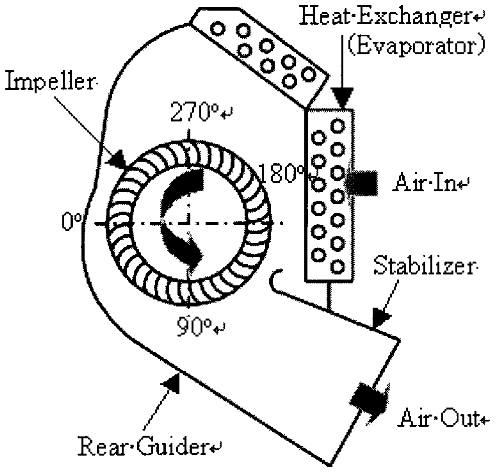


Fig. 1 The schematic diagram of a cross-flow fan.

pressure difference at the same flow rate, comparing with the other types of a blade. Then, a cross-flow fan can be applied to the small-size air conditioning fields. There exists a large impact loss in blades with a small setting angle. In general, the efficiency is commonly increased as the setting angle of a blade is larger. Typically the efficiency of a cross-flow fan is 30 to 40% due to a forward curved blade with a small setting angle.

In addition, the dynamic pressure is converted into the static pressure in a stabilizer and a rearguider. Therefore, the flow behaviors in a stabilizer and a rearguider are more important than the others, such as radial and backward curved blades. Because the rearguider and the stabilizer as a scroll in the turbo fan have significant correlations to the performance, the characteristics of the static pressure difference and the efficiency can be investigated through the modification of those shapes.

There are two different types of vortices in the flow fields of a cross-flow fan. One is an eccentric vortex, i.e., forced vortex, induced by recirculation from the stabilizer to the impeller, and the other is the free vortex in the rearguider. Resulting from these vortices, the impeller has two regions that work as a turbine and a pump. Especially, the control for the location and the intensity of the eccentric vortex are connected directly with the fan performance. There are several studies on shape parameters of the components of a cross-flow fan, but design theories like a turbo pump have not been established yet. The analytical methods are mainly not theoretical ways but reiterated case studies by empirical and numerical methods.

Eck[1] studied the entire theories on the cross-flow fan using experimental and analytical methods. First, he studied the eccentric vortex through flow visualization. Tsurusaki et al.[2,3] measured the velocity of the internal flow in the cross-flow fan using PTV (particle tracking velocimetry). They also found the path line and velocity distribution using a digital camera and calculated the generation and the diffusion of the vorticity induced by the eccentric vortex. Yamafugi and Nishihara[4] clarified the generation procedures of irregular main flow by LDV and disclosed the production of vortex shedding at a blade tip. It is verified that the principle of the eccentric vortex could be comprehended from those phenomena. Murata et al. [5-7] inquired that the setting angle, the gaps between components and the rearguider shape are important design parameters with the influence on the fan performance. In case of a small variation of Reynolds number with a little change on the diameter and the rotational velocity, it is reported that the flow and pressure coefficients are valid by means of the study on the scale effect versus Reynolds number. In this study, the flow behavior and the performance of the cross-flow fan with various setting angles of the stabilizer are numerically investigated.

2. MODEL OF THE CROSS-FLOW FAN

2.1 SHAPE OF THE REARGUIDER

The flow behavior of a cross-flow fan has the same nature as that of a turbo pump because of the incompressibility of flows in the fan ($M<0.3$). In a turbomachinery, there is the largest loss within an impeller and next is occurred within a scroll. Therefore, it is very important how to design the optimal shape of the rearguider.

A working fluid acquires the energy, passing through blades of the impeller twice. Some particular part of the impeller only discharges air into the rearguider. In this study, the rearguider curve from the starting point to the exit duct is designed with Archimedes spiral. That spiral is known as one having the excellent pressure recovery in the general scroll of a turbomachinery. Aforementioned, as flowing out within the

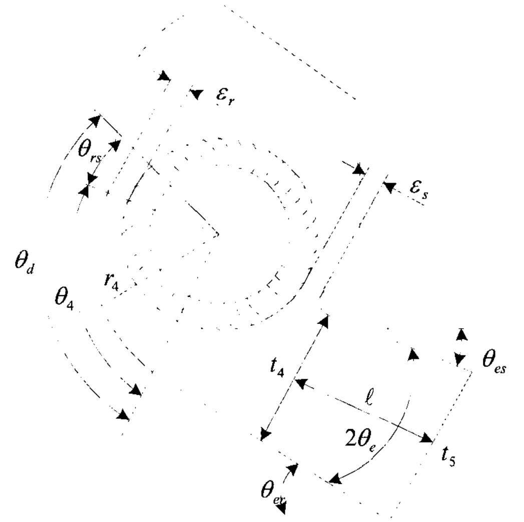


Fig. 2 Schematic diagram and design parameters of indoor unit of a cross-flow fan.

restricted region of the impeller, the value of a discharge angle (θ_d) is not 360° but empirical values. The relevant formula for the spiral is as follows:

$$r_4(\theta) = r_3 \exp[Q_d\theta/(C_{\theta 2} \cdot r_2 \cdot b_2 \cdot \theta_d)] \quad (1)$$

where,

$$r_3 = r_2 + \varepsilon_r$$
$$C_{\theta 2} = C_{m2} / \tan\alpha_2$$
$$\alpha_2 = \tan^{-1}[W \sin\beta_2 / (U + W\cos\beta_2)]$$
$$U = \pi DN / 60$$
$$W = C_{m2} / \sin\beta_2 = (Q_d / A_{exit} \sin\beta_2).$$

The referred variables are shown in Fig. 2. The rearguider curve is made by Eq. (1) at the design point, 4.5 CMM.

2.2 SETTING ANGLE OF THE STABILIZER

In this study, the modeled shape of a stabilizer is chosen with a circle one because it has an excellent performance proven by Koo et al.[8] The general design parameters and setting angles (θ_{es}) are shown in Table 1. Especially, the setting angle is varied from 19° to 27°. The gap between the impeller and the stabilizer is fixed at 3.5 mm.

Table 1. Design parameters of the modeled cross-flow fan.

D_2	95 mm
β_2	29°
ε_r	5.9 mm
θ_{es}	25°, 23°, 21°
ε_s	3.5 mm
Q_d	4.5 CMM (ϕ=0.51)

3. NUMERICAL METHOD

The conservative equations for unsteady, turbulence and viscous flows are as follows:

$$\frac{1}{\sqrt{g}}\frac{\partial}{\partial t}(\sqrt{g}\rho) + \frac{\partial}{\partial x_j}(\rho u_j) = 0 \quad (2)$$

$$\frac{1}{\sqrt{g}}\frac{\partial}{\partial t}(\sqrt{g}\rho u_i) + \frac{\partial}{\partial x_j}(\rho \tilde{u}_j u_i - \tau_{ij}) = -\frac{\partial P}{\partial x_i} + s_i \quad (3)$$

where t is the time, $\sqrt{g}$ the matrix equation of a tensor, ρ the density, u the velocity and s_i the source of momentum, respectively.

These governing equations were discretized by the finite volume method (FVM) to find the solutions for flow variables. For pressure-velocity coupling, PISO (Pressure-Implicit with Splitting of Operators) algorithm was adopted. This scheme is based on the higher degree of the approximate relation between the corrections for pressure and velocity. One of the limitations of the SIMPLE and SIMPLEC algorithms is that new velocities and corresponding fluxes do not satisfy the momentum

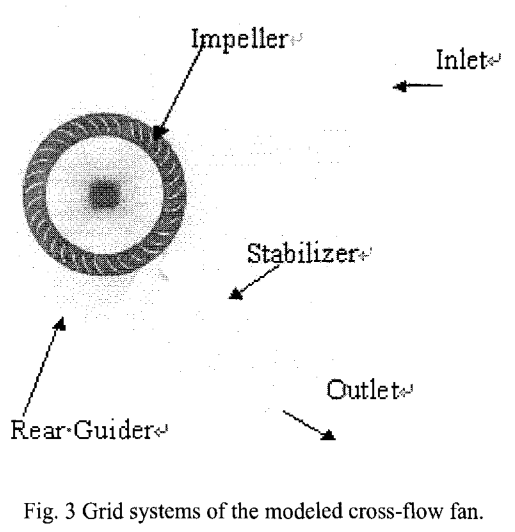


Fig. 3 Grid systems of the modeled cross-flow fan.

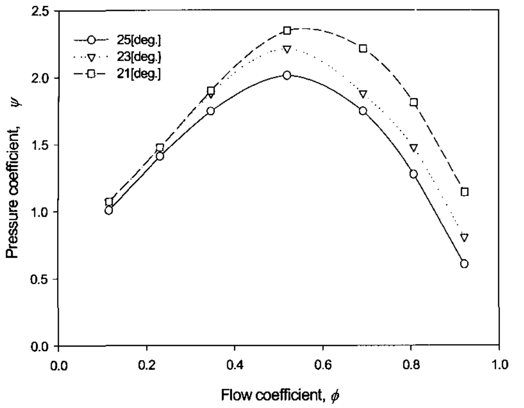
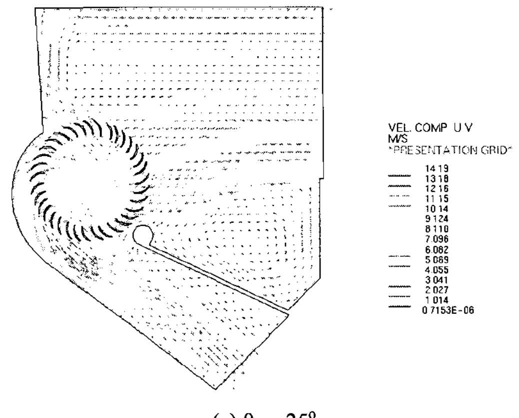


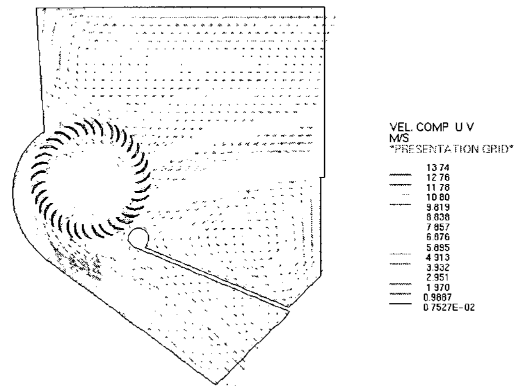
Fig. 4 Pressure coefficients versus flow coefficient.

balance after the pressure-correction equation solved. As a result, the calculation must be repeated until the balance is satisfied. To improve the efficiency of this calculation, the PISO algorithm performs two additional corrections: neighbor and skewness correction. The hybrid scheme was used to handle the convection and diffusion terms in the governing equations. The standard k - ε turbulence model with the wall function was adopted to simulate the behaviors of turbulence. Analyses were carried out until solutions reach steady state.

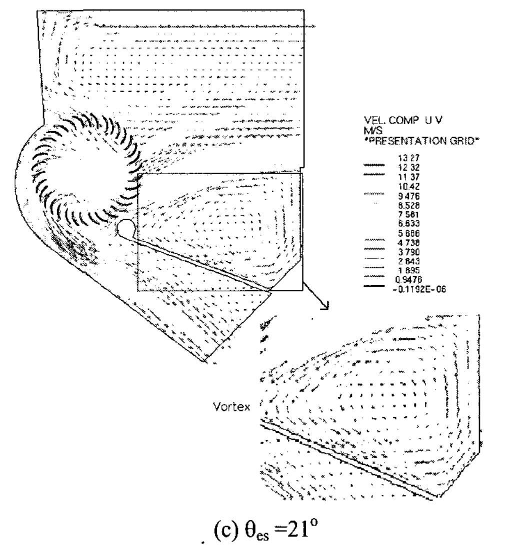
In addition, the multi-block method was used to form the complicated geometries of a cross-flow fan. The numerical domain consists of the inlet region, the impeller, the rearguider, the stabilizer and the exit duct (see Fig. 3). Those are needed to solve the complicated relations among the elements of the fan. Because the flow is complicate in a narrow region, the grids are enhanced near to each gap: ε_r and ε_s. A sliding interface between the rotating and stationary parts was applied to simulate the rotating impeller, using a commercial code, STAR-CD.[9] The sliding grid provided with an event module made simulations of the unsteady and the rotating flow behaviors in interface. It does not change the original geometry and grid number but varies the relative location of the grid to the stationary part. The grid system was made by the assumption that the flow is two-dimensional. In this study, the number of cells was 65,000 to reduce the calculating time. To minimize the error induced by the cell number, the grid number was increased up to 80,000. However, there was no more change in the value of flow variables.



(a) θ_{es} =25°



(b) θ_{es} =23°



(c) θ_{es} =21°

Fig. 5 Velocity distributions of the modeled cross-flow fan at design point.

Pressure and outlet boundary conditions were respectively adopted for the inlet and the outlet boundaries as shown in Fig. 3. The attachment boundary condition was used on the interface among the rotating cells near the impeller and the immovable cells. The rotating cells will be linked with the next fixed cells along the rotating direction and new regions be produced at that time. No-slip condition was adopted on walls and it was assumed that there is no mass flux to normal direction for a wall. Wall functions were employed to reduce the number of cells for calculating turbulence variables.

4. RESULTS AND DISCUSSION

In order to investigate the performance of a cross-flow fan, the pressure coefficient ($\psi=\Delta p/(0.5\rho U_2^2)$) against the flow coefficient ($\phi=q/(b_i d_2 U_2)$) is plotted in Fig. 4. It is seen that the maximum pressure coefficient is occurred at the design point (ϕ=0.51). As the setting angle (θ_{es}) decreases, the pressure coefficient becomes higher. Especially, the difference of the pressure coefficient is small on the left side of the design point and that obviously large on the right. This phenomenon might be resulted from the strong unsteady flow fields that induce a stall at the low flow rate. The stall causes loss and makes the difference of the pressure coefficient be smaller.

The distributions of the velocity vectors in a cross-flow fan with various setting angles are shown in Fig. 5. The working fluid passes the first blade cascade through the right side of the impeller and enters the second cascade of the left side. The fluid obtains the momentum from the centrifugal force and the Coriolis force generated by the rotation of the impeller. The working fluid is also discharged into the rearguider within the discharge region (θ_d). However, some of the primary discharged fluid is recirculated to the impeller, which results in the eccentric vortex. This vortex is occurred necessarily when a cross-flow fan works. The recirculation is an essential factor to produce the loss of the cross-flow fan. As the setting angle decreases, the inlet area of the diffuser (t_4) is decreased (refer to Fig. 2) and much less flow is recirculated. Meanwhile, some of the working fluid is discharged from the upper region of the impeller. That is generated by a diffusion of the main flow inside the impeller and separated into two directions. Some of that returns to the inlet and the other enters the rearguider. This

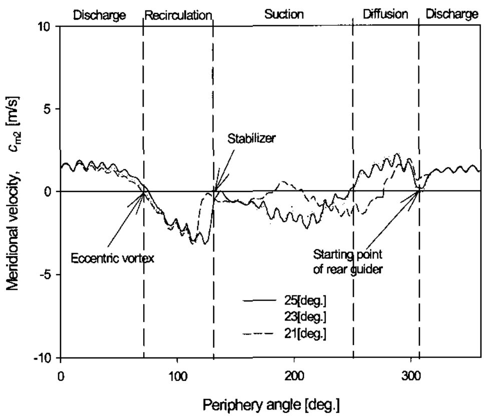


Fig. 6 Distribution of meridional velocity at 1 CMM (ϕ=0.11).

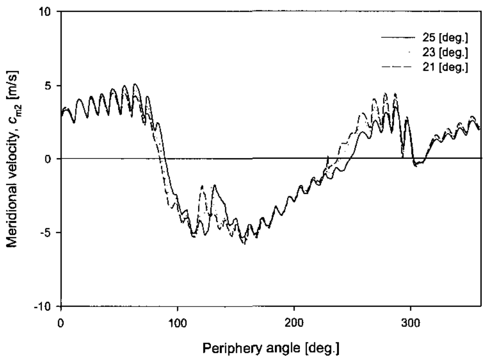


Fig. 7 Distribution of meridional velocity at 4.5 CMM (ϕ=0.51).

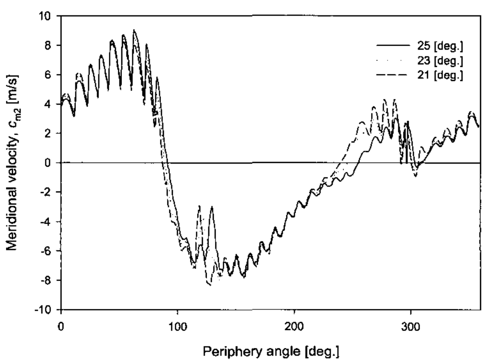


Fig. 8 Distribution of meridional velocity at 8 CMM (ϕ=0.91).

separation has an influence on stability rather than the performance because it is just the diffusion flow. Therefore, it should be also thoroughly considered at the low flow rate.

In order to elucidate the velocity distribution around the impeller in detail, the meridional velocity profiles are prepared in Figs. 6 to 8. If the value of the velocity is positive, the flow is discharged from the impeller. On the contrary, for a negative number, it is denoted that there exists suction phenomena. As mentioned earlier, the eccentric vortex is an important parameter on the design of the cross-flow fan and the recirculation induced by the eccentric vortex makes it difficult to determine the design flow rate in the design process. It is also noted that the distance from the eccentric vortex to the stabilizer is important for improving the performance. If the position of this vortex is located near the stabilizer, the performance becomes higher as being generally known. As the setting angle is smaller, this distance becomes smaller. That makes the recirculation quantity be reduced at all flow rates. Therefore, in case of the smaller setting angle, we may deduce that the pressure coefficient is higher than the others. As the flow rate increases, the region of the recirculation is diminished because the discharge region is enlarged.

5. CONCLUSIONS

In this study, the flow behaviors in a cross-flow fan with various setting angles of a stabilizer are studied. The following conclusions are obtained:

(1) As the setting angle decreases, the pressure coefficient is increased and the inlet area of the diffuser is decreased.
(2) In that case, much less flow is recirculated to the impeller. This phenomenon may reduce the loss and lead to increase the velocity in the whole domain.
(3) The recirculation region is reduced as the flow rate increases, because the discharge region of the impeller is enlarged.

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