Scale Effects in Cross Flow Fans*

(Effects of Fan Dimensions on Flow Details and the Universal Representation of Performances)

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Laser doppler velocimeter (LDV) measurements of velocity fields and the determination of static pressure distributions in the rotors of cross flow fans with good geometrical similarity but different dimensions have been performed, and the peculiar scale effects appearing in the flow details have been shown. The internal flow fields tend to be more active with increasing rotor size, and normalized velocity, vorticity, velocity fluctuations and pressure drops in the eccentric vortex increase as the fan dimensions increase. The mechanisms of generation and dissipation of vorticity in the eccentric vortex in rotor have been discussed, and it is suggested that the scale effects may be attributed to differences in the rate of diffusion of the circulation vortices in the recirculatory region. By analyzing the influences of Reynolds number and fan dimensions on performance curves, a new procedure for representing the performance of cross flow fans in a universal form has been proposed.

Key Words: Turbomachinery, Flow Measurement, Cascade, Internal Flow, LDV, Pressure Distribution, Reynolds Number, Scale Effect

1. Introduction

The dimensions of rotors markedly affect the performance curves of cross flow fans, even those with good geometrical similarity. In a previous paper⁽¹⁾, systematic tests using three test fans with good geometrical similarity but different sizes were reported, and details of effects of the rotor dimensions, so-called scale effects, appearing in the performance curves have been discussed. The results indicate that the conventional affinity law, which is considered to be useful for many types of turbomachines, does not hold for cross flow fans, but evident scale effects do appear in the performance curves. It has been pointed out that not only the Reynolds number but also the fan dimensions must be taken into account to evaluate the fan performance. However, it is also essential to clarify the details of scale effects appearing in the flow details such as internal flow fields of fans as well as

the operational conditions of cascades of airfoils of fan rotors in order to understand the scale effects of cross flow fans.

Theoretical studies of internal flow fields of cross flow fans^{(2),(3)} and many experiments on flow visualization⁽⁴⁾, measurements of the internal flow of rotors with a Pitot tube⁽⁵⁾⁻⁽⁷⁾ as well as an attempt to observe the unsteady velocity fields relative to the cascade of airfoils of rotors using a hot-wire anemometer⁽⁸⁾ have been reported. On the basis of this research, the details of cross flow fan operation have been clarified. However, the operation of cross flow fans is quite unstable and very sensitive even to tiny disturbances in comparison with many other types of fans. For example, the internal flow fields are sometimes affected by even very small probes inserted in the rotors. Therefore, precautions must be taken not to introduce any small disturbances into experiments on the scale effects appearing in the internal flow fields of cross flow fans. On the other hand, since the scale effects in the performance curves appear to be quite complicated, as discussed in the previous paper⁽¹⁾, it appears difficult to evaluate the scale effects quantitatively. However from the point of

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view of practical use, data obtained from many tests using various fans with different dimensions have to be qualitatively compared, and the results must be applied to other fans with different dimensions. Thus it is essential to determine a general procedure to represent fan performance in a universal form.

In the present study, flow details of velocity fields in the rotors are investigated with laser doppler velocimeter (LDV) and the static pressure distributions on the side walls of the rotors are determined in order to study the scale effects appearing in the internal flow of cross flow fans without introducing any unexpected disturbances. Then, the effects of Reynolds numbers and the dimensions of rotors on the performance curves are quantitatively analyzed using the data given in the previous paper⁽¹⁾, and a new procedure to represent the performance of cross flow fans in a universal form is proposed.

2. Experimental Apparatus and Procedure

2.1 Test fans and measuring instruments

A schematic diagram of the test facility is given in Fig. 1. The test fans are mounted on the surface plate ① for LDV measurements. Main components of fans such as the frame ②, a test rotor ③, side walls ④ and the casing, are the same as those used in the

Fig. 1 Experimental facility

previous paper⁽¹⁾. The LDV system is composed of a one-component general-purpose optics with Ar-ion laser of maximum output 2 Watt. The system consists of a laser tube ⑥, an optical assembly with a backscatter section ⑦, a front lens ⑩ with focusing length of 310 mm and a photomultiplier section ⑰, and these components are assembled on a common mounting bench. The LDV system is set on the X-Y table ⑪ located on the measuring surface plate. The measuring point of LDV can be traversed in two horizontal directions by moving the X-Y table using stepping motors ⑮ and the locations are determined with optical linear encoders ⑬. It is traversed in a vertical direction by moving the upper one of two mirror units ⑧ ⑨, which moves with the front lens mounting rig along a linear traversing unit ⑫, using a pulse motor ⑯, and its vertical location is determined with a linear encoder ⑭. The traversing of the measuring points and the acquisition of the data are automatically controlled with a microcomputer. A clear flat glass observation window for the LDV measurement is installed in the end wall of the rotor. In addition, another type of end wall with many pressure taps in various radial locations, which can be rotated in an azimuthal direction, is used for the static pressure measurement.

2.2 Measurement procedure

A block diagram of the measuring system is illustrated in Fig. 2. The velocity data obtained from the counter processor of the LDV system were acquired with an interface unit and stored in the microcomputer to carry out simple processing such as time-mean averaging or root-mean averaging in real time⁽⁹⁾. Since the LDV system has only one-component optics, the orthogonal components of velocity can be resolved by rotation of the optics in the range of $\pm 45^\circ$ around the mean flow direction, and time-mean velocity vectors and velocity fluctuations were calculated from these data. An aerosol of distilled water generated with an ultrasonic vibrator was used for seeding for the LDV measurement. The static pressure on the end wall of the rotor was measured with a pressure transducer and the output signals were digitized with an A/D converter interface and processed in real time and stored in the microcomputer to obtain the time-mean pressure. These data in the microcomputer were transferred to a mini-computer using a high-speed computer network and all further data processing was carried out on the mini-computer.

3. Experimental Results and Discussion

The operating points, with approximately equal Reynolds numbers, at which flow details were

Fig. 2 Block diagram of measuring system

Fig. 3 Operating points investigated

Fig. 4 Similarity of the internal flow fields

investigated with the LDV are shown in Fig. 3. In these tests the flow velocities were determined in the plane of the midwidth of the rotor. The flow parameters are represented in the normalized forms nondimensionalized with the outer diameter of the rotors D_2 , the tip speed of rotors u_{t2} and the air density ρ . The outer diameters of the three rotors R_1, R_2 and R_3 used in the study are 75, 135 and 200 mm, respectively.

3.1 The relation between internal flow fields and performance

The operating points on stable performance curves, points ① and ②, are selected as typical operating points at which the flow coefficient Φ and the total pressure rise coefficient Ψ are effectively the same. The corresponding flow details are indicated in Fig. 4. In the figure, velocity vectors, streamlines and veloc-

ity fluctuation $\overline{u \cdot v}$ are illustrated, where u and v are the velocity fluctuation in the direction of and orthogonal to the streamlines, respectively. The normalized flow parameters for the two cases agree with each other fairly well, which indicates that satisfactory similarity of the inner flow fields is attained in these cases. Further investigation reveals that the velocity fluctuations $\overline{u \cdot u}$ and $\overline{v \cdot v}$ also exhibit good similarity as in the case of $\overline{u \cdot v}$. On the other hand, flow fields at the operating points ③ and ④ on the unstable performance curves are as illustrated in Fig. 5. Although Φ and Ψ at these points are approximately equal, the flow fields exhibit quite poor similarity, and it appears that the details of the two flows are essentially different. These findings show that the normalized flow parameters Φ and Ψ are

Fig. 5 Similarity of the internal flow fields

approximately equal when the internal flow fields are fairly similar, whereas the flow fields are not necessarily the same even when Φ and Ψ are equal. These results may arise partly because the parameters Φ and Ψ are defined under the assumption of similarity of the operating conditions of the rotor cascade, and partly because the fan performance is determined as the averaged performance of the rotor cascades over all azimuthal locations.

3.2 The scale effects appearing in the details of the internal flow

The flow fields in the rotor operating at specific points on the performance curves at similar Reynolds numbers will be discussed. Figure 6 illustrates the details of the internal flow of rotors operating at points ⑤ - ⑦ in Fig. 3. Figures 6(a) - (c) show the normalized flow details, time-mean velocity vectors, streamlines Ψ_{sl} , vorticity ω , velocity fluctuation $\overline{u \cdot u}$ and static pressures on the end wall of the rotor C_p . The velocity vectors and the configurations of streamlines are quite different depending on the dimensions

Fig. 6 Flow details at the peak pressure points

of the rotor. The velocity for the smallest rotor R_1 is the lowest and velocity increases with the dimensions of the rotor. The vorticity ω in the through-flow region, where the flow passes through the rotor from the suction side to the discharge side, is fairly small. Such a region corresponds to the specific area where total pressure of the flow is maintained almost constant as shown by Murata and Nishihara⁽⁸⁾. On the other hand, the vorticity increases in the recirculatory region on the discharge side, where flow patterns with closed streamlines across the rotor cascade are formed, which indicates that an eccentric vortex is formed in the rotor in all cases. The strength of the vorticity, expressed as an absolute value, in the eccentric vortex is largest for the largest rotor diameter R_3 and it gradually decreases by a factor of 1/2 as the dimensions of the rotor decrease from R_3 through R_2 to R_1 . This finding shows that scale effects appear in the strength of the eccentric vortex. For further discussion of the flow details in the rotor, the velocity fluctuations $\overline{u \cdot v}$ for the three test rotors are also compared in Fig. 6. The equal-intensity contours are mostly perpendicular to the streamlines, which represents the decay of the wake behind the cascades of airfoils of the rotor. A scale effect clearly appears in the velocity fluctuation as well, and normalized intensity of the velocity fluctuation increases with the dimensions of the rotor. In addition, the distributions of the static pressure on the end wall of the rotor C_P are quite similar to the vorticity distributions described above, and the pressure drop in the eccentric vortex rapidly diminishes with decrease in the dimensions of the rotor. To summarize the results obtained here, the internal flow tends to become more active with increase in the dimensions of the rotor, and normalized flow parameters such as time-mean velocity, vorticity, velocity fluctuation or pressure drop in the eccentric vortex, increase with the dimensions of the rotor. Further investigation has been carried out for the flow details at operating points ⑧ – ⑩ corresponding to the peak pressure points on the unstable performance curves, which shows that similar scale effects to those described above also appear under these flow conditions although some minor numerical differences are observed.

3.3 Scale effects appearing in the operation of the rotor cascades

The flow parameters near the cascade of airfoils along the periphery of the rotor were calculated from the data described above using interpolation or extrapolation. Figure 7 illustrates the azimuthal distributions of normalized flow parameters relative to the moving cascades of airfoils at the inner radius of the rotors, namely relative velocity to the cascade

Fig. 7 Flow conditions at the inner radius of the rotor

W_1 and relative flow angle β_{w1} . In the figure, the azimuthal coordinate θ is measured in the direction of motion of the rotor. The symbols θ_c and θ_z denote the locations of the edges of a rear wall and a vortex wall of the casing, respectively. The parameter θ_d indicates the specific point at which the radial velocity component V_r changes from negative to positive, i.e., the flow direction changes from inflow to outflow at that point. A symbol θ_v indicates the azimuthal location of the center of the eccentric vortex, at which V_r changes from positive to negative. The location of the center θ_v is almost constant independent of the dimensions of the rotor but θ_d appreciably shifts in the direction of rotation of the rotor with increase in the dimension of the rotor.

Next, changes in the operating conditions of the cascades of airfoils along the direction of motion of the rotor will be discussed. The flow angles β_{w1} relative to the airfoils moving on the suction side near $\theta = \theta_z$ have relatively small values of $15^\circ - 30^\circ$, and W_1 is fairly large. When the ring cascade is developed into a linear cascade at the outer radius, the stagger angle is about 30° with a solidity of as much as 1. However, the camber angle of the cross section of the airfoils of the cascade is about 70° , which is too large for the conventional linear cascade, therefore some local separations must occur on the airfoils of the cascade as suggested in a series of photographs presented by Yamafuji⁽⁴⁾. In fact, the outlet airflow angles from the cascade are quite far from the outlet angle of the cascade. Then, an increase in β_{w1} and a decrease in W_1

occur and the cascade may experience a heavy stall as the airfoils approach $\theta = \theta_d$. Such a stalled flow condition near θ_d can be observed from the hot-wire traces of the relative velocity to the cascade of airfoils obtained by Murata and Nishihara⁽⁸⁾. Then, β_{w1} relative to the airfoils of the cascade entering the discharge side decreases more rapidly with the diameter of the rotors, but its values remain fairly small, as low as $10^\circ - 30^\circ$, in a wider peripheral region. This finding implies that the operating conditions of the cascade can be restored to the favorable flow conditions in this region. When the airfoils of cascades approach the eccentric vortex more closely, increase in W_1 with increase in the rotor dimensions becomes more marked as mentioned above. It is well known that most of the work done by the rotor cascade is transferred during the movement on the suction side⁽⁶⁾; thus the favorable operation of the rotor cascade in this region contributes to an increase in the total discharge pressure rise of the fan more effectively as the dimensions of the rotor are increased. The linear cascade developed from the ring cascade at the inner radius of the rotor has the stagger angle of -35° , which is never used in the conventional linear cascade arrangement. Moreover, the camber angle of airfoils, about 70° , is also too large for this rotor. Judging from these circumstances, it is reasonable to consider that the cascade even with incidence angles as low as $10^\circ - 30^\circ$ does not recover from the stalled condition. Furthermore, a sudden increase in β_{w1} and a decrease in W_1 occur as the airfoils approach the center of the eccentric vortex, and the cascade of airfoils may stall heavily again. When the cascades of airfoils have passed the center of the eccentric vortex, they experience a rapid increase in β_{w1} and an increase in W_1 , after which the airfoils leave the discharge side to return to the suction side. The azimuthal extent indicated by r.r in Fig. 7 means the recirculatory region calculated from the configurations of the streamlines in Fig. 6, which rapidly decreases with increase in the dimensions of the rotor. Such a change contributes to an increase in the flow rate at the peak pressure point more effectively as the dimensions of the rotor increase. In addition, the recirculatory region mostly coincides with the area filled with vorticity of high intensity; thus it can be concluded that the eccentric vortex in the rotor of the cross flow fan has considerable influence on the performance curves from the point of view of the flow rate characteristics as well as the discharge pressure characteristics.

3.4 Discussions of the formation of the eccentric vortex and the mechanism of the onset of the scale effects

As mentioned above, the major feature of the scale effects appearing in the flow fields can be represented by the differences in the configuration of the eccentric vortex which dominantly influences the internal flow of the rotor. It is reasonable to understand that the eccentric vortex is composed of the circulation vortices shed from the moving airfoils of the cascade due to their unsteady operating conditions and that the eccentric vortex is maintained by the supply of shedding vortices from the cascade of airfoils as discussed by Yamafuji⁽⁴⁾ and Lajos⁽¹⁰⁾. For the rotor cascade tested, the supply of circulation vortices is produced in the following manner. The airfoils passing near $\theta = 180^\circ$ where the cascade operates in fairly favorable conditions enter the recirculatory region, then they rapidly drop in a heavily stalled condition before they reach the center of the eccentric vortex. In this process, circulation vortices around the airfoils with the same sense as the rotor rotation are shed from the individual airfoils toward the outer region of the rotor. These vortices are taken into the rotor again as the cascade moves from the center of the eccentric vortex to the boundary of the recirculatory region. Since the flow around these airfoils is in the process of recovering from the heavily stalled condition, the airfoils travel with increasing circulation counter to the rotor rotation around them, and therefore circulation vortices with the same sense as the rotor rotation are shed again from the airfoils towards the inside of the rotor. Thus, there must be some mechanisms for the dissipation of the circulation vortices in the recirculatory region in order to maintain steady eccentric vortex strength. The natural dissipation of the circulation vortices due to the action of the viscosity might be considered as a mechanism. However, the rate of decay of the vortices along the streamlines is negligibly small as shown in Fig. 6, and it is also reported by Lajos⁽¹⁰⁾ that the vortices along the streamlines are maintained fairly constant. Therefore, the natural decay of the circulation vortices cannot be a major mechanism of the dissipation of the vortices. It is more realistic to consider the mechanism to be the diffusion effects of the circulation vortices, and the carry-over effects on the diffused vorticity due to the through flow of the fan. Namely, the circulation vortices shed from the airfoils of the cascade circulate along the streamlines of the eccentric vortices, and some of the vortices diffuse to the outside of the recirculatory region, and are carried away to the exit of the fan by the through flow. This process may cause the dissipation of the circulation

vortices in the rotor.

When the location and the strength of the eccentric vortex are discussed from the point of view of the energy, they are determined such that the work done by the rotor cascade in the recirculatory region balances the dissipation due to the action of the viscosity⁽¹¹⁾. On the other hand, from the point of view of the total circulation in the rotor, they are determined such that the rate of generation of the circulation shed from the airfoils in the circulatory region balances the rate of its dissipation. Therefore, if any small differences in the process of dissipation of the circulation vortices occur, the configuration of the eccentric vortex formed in the rotor gradually changes in time even for fans with geometrical similarity, which may lead to non-negligible changes in the operating conditions of the cascade as well as in the performance curves of fans.

It seems to be very difficult to identify the dominant factor which gives rise to the changes in the rate of the dissipation of the circulation vortices from the present study. But it is worth noting that the operating conditions of cross flow fans are quite unstable and are accompanied by random fluctuations of the flow conditions; moreover the internal flow field has quite high turbulence. Judging from these circumstances, it appears that the rate of dissipation of the circulation vortices is modified by small differences in the action of the viscosity and velocity fluctuations as well as by the turbulence of the internal flow.

4. A Universal Representation of the Fan Performances

4.1 Influences of the Reynolds numbers

The performance curves of a cross flow fan at various rotational speeds of rotors are schematically shown in Fig. 8. Since the influences of the Reynolds

Fig. 8 Effects of rotational speeds on the performance curves

number Re_c on the performance curves of cross flow fans are more marked than those of other types of fans, the performance curves cannot be represented as a single curve. The corresponding operating points at the same throttle position are not identical but spread along a quasi-parabolic line. When these performance curves are reduced to a single curve by compensating for Re_c , it seems reasonable that the data are so compensated that the operating points at the same throttle position are identical. We assume that the flow coefficient Φ_j and the total pressure rise coefficient Ψ_j at the Reynolds number $(Re_c)_j$ can be represented in the following form:

$$\Phi_r = \frac{\Phi_j}{(Re_c)_j^{n_1}} = \frac{\Phi_{j+1}}{(Re_c)_{j+1}^{n_1}} \quad (j=1, 2, 3, \dots)$$

$$\Psi_r = \frac{\Psi_j}{(Re_c)_j^{n_2}} = \frac{\Psi_{j+1}}{(Re_c)_{j+1}^{n_2}}$$

If the constant parameters n_1 , n_2 are well determined, the performance of the cross flow fan can be represented in a universal form with the reduced flow coefficient Φ_r , reduced total pressure rise coefficient Ψ_r and the parameters n_1 , n_2 .

Parameters n_1 and n_2 were calculated from the corresponding data at the same throttle position but with different rotational speeds using the following equations:

$$n_1 = \frac{\ln(\Phi_j/\Phi_{j+1})}{\ln\{(Re_c)_j/(Re_c)_{j+1}\}} \quad (j=1, 2, 3, \dots)$$

$$n_2 = \frac{\ln(\Psi_j/\Psi_{j+1})}{\ln\{(Re_c)_j/(Re_c)_{j+1}\}}$$

The results indicate that the calculated n_1 and n_2 are not exactly constant for all cases but are somewhat scattered, thus their mean values are calculated for every test rotor and used in the following discussions. The validity of the procedure for compensating the influences of Re_c is demonstrated in Fig. 9. These results correspond to the data for the test rotor R_2 , for which the most wide-ranged performance tests were carried out⁽¹⁾, and indicate that the influences of Re_c are effectively compensated on the whole except for some data for especially large flow rates or quite low Reynolds numbers. Further investigations reveal that similar verification can be expected for the other data.

4.2 Influence of fan diameter

Parameters n_1 and n_2 obtained for the three test rotors are given in Fig. 10, which shows that n_1 is approximately proportional to the diameter of the fan D_2 , whereas parameter n_2 is almost constant independent of D_2 . The relations between n_1 , n_2 and D_2 were determined by the method of least squares to be,

$$n_1 = 1.02 \times 10^{-3} \times D_2$$

$$n_2 = 0.173$$

where, the unit of D_2 is mm. From these relations, the parameters n_1 and n_2 were evaluated for every

Fig. 9 Compensation of the influences of the Reynolds numbers

Fig. 10 The relations between n_1 , n_2 and D_2

test rotor and performance curves⁽¹⁾ were reduced from the form of $\Phi-\Psi$ to the form of $\Phi_r-\Psi_r$ in order to verify the procedure.

The performance curves represented by the conventional form of $\Phi-\Psi$ and by $\Phi_r-\Psi_r$ for the three test rotors operating at maximum and minimum rotational speeds are given in Fig. 11. The present procedures are not necessarily satisfactory for all cases since Re_c and D_2 are not fully compensated in some cases, especially for some data of unstable performance. However, it can be appreciated that the representation of performance curves with Φ_r and Ψ_r effectively approaches a universal one compared with the conventional $\Phi-\Psi$ representation. Some experimental results given by Murata and Nishihara⁽¹²⁾, which were obtained with fans fairly similar to those used in the present experiments, were also reduced to

$\Phi_r-\Psi_r$ relations as illustrated by broken lines in the figure. Non-negligible differences appear in the large-flow-rate region, but this may be partly due to differences in the geometrical configurations of the fan, such as the ratio of inner to outer diameter, ratio of width to diameter, or number of rotor blades. Thus it can be concluded that the compensating procedure presented here is also essentially valid for these data.

For Reynolds numbers higher than the critical values $Re_c = (1.0 - 1.5) \times 10^4$, the influence of Re_c becomes negligible as mentioned in the previous paper⁽¹⁾; however the influences of the rotor dimension still remain. In this region the actual Reynolds numbers may be replaced by the critical value Re_c in the reduction procedure. The parameters n_1 , n_2 presented here are introduced to deal with the fact that the performance of the cross flow fans does not obey the conventional affinity law of fans. For other types of fans, such as axial or radial flow fans, in which only negligible scale effects appear and performance curves can be represented by conventional $\Phi-\Psi$ relations, the parameters corresponding to n_1 and n_2 may be chosen to be zero. Therefore, the newly introduced parameters Φ_r , Ψ_r do not conflict with the conventional ones but are reasonable parameters which represent the performance of various types of fans in a universal form.

5. Conclusions

The internal flow fields of the rotors of cross flow fans are determined by LDV and wall static pressure

Fig. 11 A universal representation of performance curves

measurements, and the scale effects appearing in the flow details have been shown. On the basis of these results, the operating conditions of the rotor cascade have been discussed and a mechanism of the onset of the scale effect has been suggested. From further analysis of the data of fan testing, the influences of the Reynolds numbers and the rotor dimensions have been evaluated, and a procedure for universal representation of fan performance has been proposed. The results obtained can be summarized as follows:

(1) When the similarity of the internal flow field is maintained, performance of cross flow fans represented by the conventional parameters Φ - Ψ is mostly the same even for rotors with different dimensions. However, the internal flow details are not necessarily the same even when Φ and Ψ are equal.

(2) Remarkable scale effects appear in the flow details at particular rotor operating points such as at the peak pressure points. The internal flow fields tend to be more active with increasing dimensions of the rotor, and normalized flow parameters increase with the diameter of the rotor.

(3) The cascade of airfoils passing through the discharge side, where most of the work done by the rotor is transferred to the flow, operates under favorable conditions with fairly small incidence angles, and the extent of this region increases with the dimensions of the rotor, which contributes to the large increase in discharge pressure. In addition, the extent of the recirculatory region formed on the discharge side rapidly diminishes with increasing fan diameter, which also contributes to the increase in flow rate at

the peak pressure point.

(4) It appears that dissipation of the circulation vortices flowing along the eccentric vortex may be caused by the diffusion effects of the vortices and also by the carry-over effects on the diffused vorticity due to the through flow of the fan. Any small differences in the rate of diffusion may cause non-negligible changes in the internal flow details, which may lead to the onset of the remarkable scale effects in the fan performance.

(5) The influences of the Reynolds number and the dimensions of the rotor are quantitatively evaluated using two parameters, n_1 and n_2 , which were determined in relation to the diameter of the rotor. On the basis of these results, a procedure for representing fan performance in a universal form has been proposed and its validity has been demonstrated.

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References

- (1) Tanaka, S. and Murata, S., Scale Effects in Cross Flow Fan; 1st Report, JSME Int. J., Ser. B, Vol. 37, No. 4 (1994), p. 844.
- (2) Coester, R., Theoretische und Experimentelle Untersuchungen an Querstromgebläsen, Mitt. aus den Institute für Aerodynamik, Zürich, No. 28

- (1959), p. 1.
- (3) Ikegami, H. and Murata, S., A Study of Cross Flow Fan ; Part 1, Techn. Rep. Osaka Univ. Vol. 16, No. 731(1966), p. 557.
 - (4) Yamafuji, K., Studies on the Flow of Cross-Flow Impellers, 1st Report; Bull. of the JSME, Vol. 18(1975), p. 1018.
 - (5) Murata, S., et. al., A Study of Cross Flow Fan with Inner Guide Apparatus, Bull. of the JSME, Vol. 21 (1978), p. 154.
 - (6) Nakamura, K. and Okutani, K., An Experimental Investigation into the Internal Flow in a Cross Flow Fan, Turbomachinery, (in Japanese), Vol. 11, No. 4(1983), p. 196.
 - (7) Nakamura, K., et. al., Automatic Measurements of Internal Flow of Cross Flow Fan and their Analysis, Turbomachinery, (in Japanese), Vol. 14, No. 2 (1986), p. 67.
 - (8) Murata, S. and Nishihara, K., An Experimental Study of Cross Flow Fan ; 2nd Report, Bull. of the JSME, Vol. 19 (1976), p. 322.
 - (9) Tanaka, S., An Application of Personal Computers for Laboratory Automation, Turbomachinery, (in Japanese), Vol. 15, No. 7 (1987), p. 411.
 - (10) Lajos, T., Investigation of the Flow Characteristics in the Impeller of the Tangential Fan, Proc. 5th Conf. Fluid Machinery, Budapest (1975), p. 567.
 - (11) Yamafuji, K., Studies on the Flow of Cross Flow Impellers, 2nd Report ; Bull. of the JSME, Vol. 18 (1975), p. 1425.
 - (12) Murata, S. and Nishihara, K., An Experimental Study of Cross Flow Fan ; 1st Report, Bull. of the JSME, Vol. 19 (1976), p. 314.
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