Mean Velocity Characteristics behind a Forward-Swept Axial-Flow Fan*

H.J. MYUNG** and J.H. BAEK**

Detailed measurements in the flow behind a highly forward-swept axial-flow fan at three operating conditions were performed using three-dimensional LDV system. Various phenomena related to the leakage flow, leakage vortex, hub vortex and wake were investigated. At the design condition, where no distinct leakage vortex appears behind trailing edge, the leakage flow interacts with the boundary layer of the pressure side inside the blade passage, which directly influence the wake characteristics behind the trailing edge. At the low loading condition, the leakage vortex has the characteristics of the velocity defect and this vortex induces radially inward flows at both the suction and pressure sides in the wake near the tip region. As the loading increases, radially-outward velocity component increases and the position of the maximum value in the wake is closer to the tip region. At the design and high loading condition, hub vortex induces radially inward flow at the pressure side and wakes near the hub region decays very quickly due to the interaction with the large hub vortex.

Key Words: Forward-Swept Axial-Flow Fan, Wake, Leakage Flow, Leakage Vortex, Hub Vortex, LDV

1. Introduction

The wake, developed behind an axial-flow fan, is a source of loss which deteriorates the efficiency and also affects the noise and vibration characteristics. In view of noise, wake generates noise by the interaction with the objects behind the trailing edge as well as wake itself is a source of noise. Because of these importance, studies have long been performed on the wake flow such as the characteristics of the mean velocities, turbulence and decay characteristics⁽¹⁾⁻⁽⁴⁾. However, since most measurements in the above studies cover only several radial points, the phenomena, related with the three-dimensional complex flow and radial velocity which contribute to the spanwise mixing, could not be investigated comprehensively. Especially related to the leakage flow, only rough explanations have been given because of the lack of the knowledge of the flow data inside the blade pas-

sage.

The overall flow field behind a trailing edge of an axial compressor were investigated by various authors⁽⁵⁾⁻⁽¹⁶⁾. They found that the flow is very complex due to the leakage flow, casing wall boundary layer, blade boundary layer, centrifugal force, trailing vortices, horse shoe vortex and interaction of such phenomena.

Since most of the previous studies have been focused on the compressor, researches on the fan, especially on the forward-swept fan which is widely used nowadays, are very insufficient. In this study, detailed measurements were made to investigate comprehensively the flow behind a forward-swept axial-flow fan, especially focused on the leakage flow, hub vortex and trailing-edge wake flow. The tip-region flow induced by this fan was discussed intensively in the previous paper by present authors⁽¹⁷⁾ with detailed measurements. Therefore the leakage flow characteristics behind this fan can be referred clearly from the paper.

The objectives of the present study are summarized as follows.

- 1) To investigate the characteristics of the

* Received 16th October, 1998

** Department of Mechanical Engineering, Pohang University of Science and Technology, San 31, Hyoja Dong, Pohang, 790-784, Korea. E-mail: jhbaek@Vision.Postech.ac.kr

leakage flow and its effect on the wake flow according to the loading conditions.

2) To study the influence of hub vortex on the wake flow

3) To investigate the characteristics of the radial flow in the wake

4) To offer basic data for the future study of turbulence and noise

Nomenclature

D_c : casing diameter

D_h : hub diameter

L : shaft power

PS: pressure side

ΔP_s : static pressure difference between the inlet and outlet of the duct

Q : flow rate

R_{tip} : tip radius

r : radial coordinate

Δr : radial dimension of the LDV measuring volume

SS: suction side

U_{tip} : circumferential velocity of the tip

V : phase averaged relative velocity

v : fluctuating velocity component

x : axial coordinate

Δx : axial dimension of the LDV measuring volume

Ω : angular velocity of the rotor

ϕ : flow coefficient

η : static efficiency

θ : circumferential coordinate

$\Delta\theta$: circumferential dimension of the LDV measuring volume

ψ : static pressure coefficient

Subscripts

r : radial component

s : streamwise component

x : axial component

θ : circumferential component

2. Experimental Set-Up

The test rig was designed to investigate the flow phenomena in the axial flow fan and the three dimensional LDV was used to measure the velocity. The test rig consists of an axial flow fan, fan tester, duct system and a driving unit, as shown in Fig. 1(a).

2.1 Test rig

The shape and dimensions of the axial flow fan are shown in Fig. 2, and its geometric specifications are presented in Table 1. The blade is highly forward swept for the purpose of reducing the noise generated by the interaction of the inlet flow disturbance and the blade leading edge. The thickness of the blade is

Table 1 Geometric configuration of the axial-flow fan

Number of blade	3
Tip radius	113.5mm
Hub radius	40.0mm
Tip chord length	220mm
Hub chord length	85mm
Solidity at the tip	0.93
Solidity at the hub	1.0
Maximum blade thickness	3mm
Sweep angle	42.5°
Outlet flow angle at the midspan	60.5°
rpm	1500

Fan Tester

1. Rectangular chamber
2. Meshes
3. Five nozzles
4. Pressure measurement taps for flow rate
5. Pressure measurement taps for static pressure
6. Auxiliary sirocco fan

Duct Systems

7. Circular duct
8. Bell mouth

Driving Unit

9. Cone for clean inlet flow
 10. Torque meter
 11. AC servo motor
 12. Encoder
- $L1 = 0.6m, L2 = 1m$

(a)

(b)

Fig. 1 (a) Test rig and LDV system, (b) Coordinate system

constant as a whole and the shape of the leading and trailing edges and the tip section is rather sharp. The blade is of a circular-arc type camber-line with the leading and trailing edges slightly modified.

The fan tester, made according to the ASHRAE standard, consists of a stagnation chamber, several layers of mesh, five nozzles, and an auxiliary fan. The

Fig. 2 Photograph and dimension of the fan; (a) Front view, (b) Side view, (c) Dimension of the fan

dimensions of the rectangular chamber are $1.2\text{ m} \times 1.2\text{ m} \times 3\text{ m}$. The flow rate is obtained by the pressure difference between the inlet and outlet of nozzles, which is controlled by the auxiliary sirocco fan. The total length of the duct system is 1.6 m including the bell mouth and its inner diameter is 237 mm . The tip clearance is 5 mm which is equivalent to 2.4% of chord length. The window for LDV measurement is made of plexiglass plate of 0.8 mm thickness. The driving unit consists of a torque meter, an encoder and an AC servo motor.

2.2 LDV system and uncertainty analysis

Throughout the experiment, the flow field was measured by the 3D LDV system made by DANTEC operating in the back-scattering mode. The LDV system consists of a light source, a transmitter, probes, fiber optic cables, a 3D traverse and a signal processor. LEXEL 5Watt Ar-ion laser was used as the light source. Green and violet beams are used to measure the axial and radial velocity components in the non-orthogonal coordinate system as shown in Fig. 1(b). The angle between the two probes is 30° degree. The blue beam alone is used to measure the

Fig. 3 Measurement positions

circumferential velocity component. The diameter of the front lens of the probe is 85 mm with the focal length of 600 mm . The dimension of the measuring volume in each coordinate direction, Δr , $\Delta \theta$ and Δx are approximately $340\text{ }\mu\text{m}$, $89\text{ }\mu\text{m}$ and $92\text{ }\mu\text{m}$, respectively. The DANTEC three-dimensional traverse is used to move the two probes, and the resolution of the traverse is 0.05 mm . The velocity components are obtained by the processor using auto covariance method. An encoder is used to obtain the phase averaged data from the LDV measurement. The number of pulses per one revolution from the encoder is 3600 , which means the angular resolution of the velocity is 0.1° degree. The paraffin was used for generating seed particles of the average size $1\text{ }\mu\text{m}$.

For the correction and calibration of the LDV system in the present experiment, an apparatus providing accurate reference velocity was made. It consists of a circular plate rotating by a servo motor and $5\text{ }\mu\text{m}$ hot wire probe attached to the plate. The diameter of the hot wire is the same order of magnitude as the fringe width, thus the clear velocity can be obtained when the wire passes through the measuring volume. In the case of the three-dimensional LDV system, the orthogonal velocity components are usually obtained from the non-orthogonal LDV data by the coordinate transformation. In this case, the angle between the two probes is an important factor influencing the accuracy of the velocity component. In the present experiment, the angle was set to 30° and the accuracy of the three-dimensional LDV system was examined by the calibration apparatus and it was ascertained, by a careful adjustment, that the uncertainty of the radial component can be lowered below 1.5% .

Statistical uncertainty can be estimated as stated in Snyder et al.⁽¹⁸⁾. The uncertainty of radial velocity is 1.7% , and the uncertainty of axial and circumferential velocity is lower than that of the radial velocity.

2.3 Measurement positions

Measurement positions and geometric dimensions of the fan inside the duct are shown in Fig. 3. The

Fig. 4 Performance characteristic

Fig. 5 Circumferentially averaged axial velocity at position 3

number of radial measurement positions is 24 with the increment of 3 mm and the measuring range of r/R_{tip} is from 0.40 to 1.004 at position 1. At position 2 and 3, 16 radial positions of $8/R_{tip}$ with the increment of 2 mm is ranged from 0.74 to 1.004.

3. Results and Discussions

3.1 Overall performance

The performance curves are plotted in Fig. 4. Measurements were performed at three operating conditions, A, B and C. The A, B, and C points corresponds to the conditions of the small flow rate ($\phi = 0.21$), the maximum efficiency ($\phi = 0.25$), and the large flow rate condition ($\phi = 0.30$), respectively. The flow coefficient, ϕ , static pressure coefficient, ψ , and static efficiency, η are defined as follows.

$$\phi = Q / [U_{tip} \pi (D_c^2 - D_h^2) / 4] \quad (1)$$

$$\psi = \Delta P_s / (\rho U_{tip}^2 / 2) \quad (2)$$

$$\eta = \phi \psi [\rho U_{tip}^3 \pi (D_c^2 - D_h^2) / 8] / L \quad (3)$$

Circumferentially averaged axial-velocity profiles behind the trailing edge at the above three

Fig. 6 Secondary flow field at position 3; (a) $\phi = 0.25$, (b) $\phi = 0.30$, (c) $\phi = 0.21$

flow operating conditions are shown in Fig. 5. The differences in the profiles will be clarified at the later sections.

3.2 Exit flow field

The secondary flows at position 3, which are defined by velocity components perpendicular to the circumferentially averaged relative flow directions, are represented in Fig. 6 at three operating conditions. In the previous study, the present authors⁽¹⁷⁾ showed that, at the design condition, leakage vortex is formed at the front part of the tip region but as the flow moves downstream, the strength of the vortex becomes very weak at the mid-chord region and disappears almost completely before it reaches the trailing edge, resulting in no signal of its existence in the figure of the secondary flow at position 3. A weak vortex denoted by A1 is a kind of a second vortex

generated by the leakage flow at the rear part of the tip region. There is also another vortex denoted by A2 generated by the separation of inlet flow at the inlet part of the hub. In case of $\phi=0.21$, the leakage vortex decays more quickly than that at the design condition and the size of the hub vortex becomes a little bigger as shown in Fig. 6(c). On the other hand, at $\phi=0.30$, a distinct leakage vortex denoted by B in Fig. 6(b) appears near the pressure side behind the trailing edge, which is formed at slightly further downstream position than that for the design condition. But no distinct vortex is observed at the hub region which was developed for the higher loading condition.

The secondary flows in Fig. 6 offer a lot of information on phenomena which are to occur behind the trailing edge of various axial-flow fans. So far some researchers observed distinct leakage vortex behind the trailing edge but others didn't. Recently the numerical result of Furukawa et al.⁽¹⁹⁾ and the experimental one by present authors⁽¹⁷⁾ showed that vortex can be extinguished inside the blade passage. Furukawa et al.⁽¹⁹⁾, showed that the decay is the result of the vortex breakdown, while the present authors' results⁽¹⁷⁾ showed that the vortex can disappear very fast mainly due to vortex squeezing and turbulence.

Let's consider the difference between the flow with vortex and flow without it. Since in this experiment, velocity fields are thoroughly investigated for 3 loading condition of $\phi=0.21$, 0.25 and 0.30 , the difference due to the loading can be investigated. The hub vortex in case of $\phi=0.25$ and $\phi=0.21$ can be generated generally in the fan used for home appliances, other electronic systems and ventilating systems. One of the distinct differences that arise due to the difference of the loading is the radial flow characteristics. Several studies^{(20),(21)} showed that both the convection by the mean radial flow and turbulence contribute to the spanwise mixing. However the effect of the radial flow on the performance and flow characteristics has not been investigated thoroughly.

The radial flows in case of $\phi=0.25$ and $\phi=0.21$ are to be investigated first. It is well known that the radial outward-flow is due to the imbalance between the centrifugal force and the radial pressure gradient. If other secondary-flow effects do not exist, radially outward flow is to occur at both the pressure and suction sides at the wake region, except the region close to the hub and tip. However the present experiment showed that radially outward flow exists at the suction side while inward flow at the pressure side. The maximum value of outward flow is located at the position slightly away from the wake center toward the suction side, which is clearly observed from Fig. 8

Fig. 7 Radial velocity (V_r/U_{tip}) contour at position 3; (a) $\phi=0.25$, (b) $\phi=0.30$, (c) $\phi=0.21$

and Fig. 9 with the circumferential angle as x coordinate value. The angle 0 degree in these figures corresponds to the position of the left bottom part in Fig. 7 and 180 degree corresponds to the right bottom part. It is known that the magnitude of the radially outward flow has higher value at the case of higher loading⁽²⁾. This fact is confirmed from the present results that the value at $\phi=0.21$ is larger than that at $\phi=0.25$. It seems that the radially inward flow at the pressure side is mainly affected by the hub vortex. The information on the radial flow is given more clearly by the contour plot represented in Fig. 7. In the wake region, for $\phi=0.25$, radially outward velocity has the maximum value at the position of one-third span from the tip with the 8% of the tip velocity. In case of $\phi=0.21$, maximum outward value at the one-fifth span from the tip with the 10%. This confirms that the effects of the centrifugal force is larger in case of $\phi=0.21$. The

(a)

(a)

(b)

(b)

(c)

(c)

Fig. 8 Velocities at $r/R_{tip}=0.714$ at position 3;
(a) $\phi=0.25$, (b) $\phi=0.30$, (c) $\phi=0.21$

Fig. 9 Velocities at $r/R_{tip}=0.925$ at position 3;
(a) $\phi=0.25$, (b) $\phi=0.30$, (c) $\phi=0.21$

maximum value of the radial outward velocity in the hub vortex is 12% for $\phi=0.21$.

In case of $\phi=0.30$, unlike the other two cases, radially inward flows are noticed at both the suction and pressure sides near the tip region due to the leakage vortex. The radial flow in Fig. 9(b) represents the leakage vortex at the wake region, which differs distinctly from those of the other two cases. At the suction side, radially inward flow exists up to the two-fifths span from the tip, where the maximum

inward value is 15%, which is relatively large. Radial outward velocity has the maximum value at the three-fifth span from the tip with the 6%. It was found that the radial flow behavior is affected by the flow rate especially when the hub has the blunt shape at the exit as in the case of the present experiment, showing radially inward flow as a whole. Therefore the radially inward flow becomes larger in case of $\phi=0.30$, partly due to the large flow rate condition.

Figure 10 represents the axial velocity distribution.

(a)

(b)

(c)

Fig. 10 Axial velocity (V_x/U_{tip}) contour at position 3; (a) $\phi=0.25$, (b) $\phi=0.30$, (c) $\phi=0.21$

As shown in Fig. 8 and 9, axial velocity represents overall characteristics and gives useful information near the casing although it does not represent the exact shape of the wake. As expected, its value near the casing is very low, mainly due to the leakage flow with low momentum and its interaction with the passage flow through the blade. The discrepancies caused by the loading conditions can be clearly explained by the axial velocity distribution. In case of $\phi=0.30$, the velocity distribution at both the pressure and suction sides are almost the same, which means that pressure difference between the suction and pressure sides is very low.

Figure 11 shows the flow yaw angle defined as $\tan^{-1}(V_\theta/V_x)$. As expected, the yaw angle is relatively larger near the tip region at all the three cases. At ϕ

(a)

(b)

(c)

Fig. 11 Contour of flow yaw angle at position 3; (a) $\phi=0.25$, (b) $\phi=0.30$, (c) $\phi=0.21$

$=0.30$, though the value is as a whole small compared to the other two cases, the angles are large in the upper part of the leakage vortex region due to the overturning effect of the vortex. At the suction side in the wake region, the yaw angle is smaller than that of the free stream, which is thought to be caused by the radially outward flow. Near the tip, the yaw angles at $\phi=0.25$ are found to be larger than those at $\phi=0.21$ due to the leakage vortex inside the blade passage, that is, in case of $\phi=0.21$ the leakage vortex decays more quickly than at $\phi=0.25$, so the overturning effect of the vortex persists a little longer at $\phi=0.25$. This explains the higher value for $\phi=0.25$ exists at the tip.

3.3 Wake characteristics

In the former section, the overall flow characteris-

(a)

(b)

(c)

Fig. 12 Streamwise velocity contour (V_s/U_{tip}) at position 3; (a) $\phi=0.25$, (b) $\phi=0.30$, (c) $\phi=0.21$

tics at position 3 behind the trailing edge were discussed. In this section, the investigation is focused on the structure of the wake.

Streamwise velocity contours at position 3 are shown in Fig. 12 for the overall view and the detailed profiles of the streamwise velocities at position 1, 2 and 3 at each operating conditions are represented in Fig. 13, 14 and 15, respectively. In Fig. 13(a), the wake due to the boundary layer developed on the walls of the blade is clearly shown, where the velocity at the edge of the pressure side is very low due to the interaction with the leakage flow. This low velocity region due to the leakage flow extends deep into the passage from the edge. From the previous results⁽¹⁷⁾, the effect of the leakage vortex reaches nearly the midspan and after the decay of the leakage vortex, the

(a)

(b)

(c)

Fig. 13 Streamwise velocity at $\phi=0.25$; (a) position 1, (b) position 2, (c) position 3

flow at that region interacts with the boundary layer on the pressure side of the blade. Since the flow of these has the low momentum, and cannot be completely

(a)

(b)

(c)

Fig. 14 Streamwise velocity at $\phi=0.30$; (a) position 1, (b) position 2, (c) position 3

mixed with the boundary layer, the distinct wake region denoted by dashed area appears at position 1. Prato and Lakshminarayana⁽⁴⁾ shows results in the case that the leakage flow does not interact with the

Fig. 15 Streamwise velocity at position 3 at $\phi=0.21$

wake at the near wake region. Depending on the operating condition, tip clearance and solidity, there are some former results that the leakage flow interacts with the pressure side of the blade. The present results at $\phi=0.25$ shows the wake characteristics when such interaction occurs.

At position 2, 10 mm downstream from the position 1, the distinct wake cannot be observed at position 2, which means severe mixing occurs with the leakage flow. The velocity defect due to the hub vortex is clearly seen in Fig. 13(c). At position 3, the wake is not identified due to the severe mixing with the hub vortex. The wake as well as the low momentum region of leakage flow decays moving downstream. As is well known, the decay of the wake is due to the mixing with the free stream and momentum exchange between either side of the wake. Also it is expected that the radial flow contribute to the decay of the wake. The decay of the leakage flow seems largely due to the interaction with the throughflow.

Figure 14 shows the wake characteristics at $\phi=0.30$. In this case, it is observed that both at position 1 and 2 distinct wake at the trailing edge appears, which means that the interaction of the wake with the leakage flow is weaker than that at $\phi=0.25$. The velocity defect due to the leakage vortex in this case is different from that at $\phi=0.25$. The streamwise velocities at the vortex region is lower than that of the velocity at the wake edge of the pressure side. This trend persists at position 3. From these results the velocity defect is found to be the characteristic of the leakage vortex. At $\phi=0.30$, the radial extent of the leakage flow is much reduced than that at $\phi=0.25$, which means that the strength of the leakage flow is weaker for the lightly loading irrespective of the existence of the vortex. At the hub region, as distinct

(a)

(a)

(b)

(b)

(c)

(c)

Fig. 16 Radial velocity at $\phi=0.25$; (a) position 1, (b) position 2, (c) position 3

Fig. 17 Radial velocity at $\phi=0.30$; (a) position 1, (b) position 2, (c) position 3

vortex does not exist, the trailing edge wake at the hub side is clearly identified at position 3. The region which neither the leakage flow nor hub vortex affect is present at the midspan region. At $\phi=0.21$, the leakage flow region exists more widely than that at $\phi=0.25$ because the strength of the leakage flow is stron-

ger.

Fig. 16 - 18 shows radial velocity profiles on three axial cross section planes at three operating conditions. As a preceding investigation of turbulence characteristics in the wake, the possible effect of the radial flows on the turbulence characteristics will be

Fig. 18 Radial velocity at position 3 at $\phi=0.21$

discussed in this section. The comparison of the velocity gradient gives information about the contribution of each velocity component to the generation of turbulence. Since the turbulence in the wake of axial flow rotor is characterized as anisotropy with higher radial component, the effect of the radial velocity on turbulence is of much concern. The production terms of the Reynolds-stress equations in x , θ , r directions are given as follows.

$$\frac{\overline{v_x^2}}{2} : -\overline{v_x} \frac{\partial V_x}{\partial x} - \overline{v_r v_x} \frac{\partial V_x}{\partial r} - \overline{v_x v_\theta} \frac{\partial V_x}{\partial \theta} \quad (4)$$

$$\frac{\overline{v_r^2}}{2} : -\overline{v_r} \frac{\partial V_r}{\partial r} - \overline{v_\theta v_r} \frac{\partial V_r}{\partial \theta} - \overline{v_r v_x} \frac{\partial V_r}{\partial x} + \overline{v_r v_\theta} \frac{V_\theta}{r} \quad (5)$$

$$\frac{\overline{v_\theta^2}}{2} : -\overline{v_\theta} \frac{\partial V_\theta}{\partial \theta} - \overline{v_r v_\theta} \frac{\partial V_\theta}{\partial r} - \overline{v_x v_\theta} \frac{\partial V_\theta}{\partial x} - \overline{v_\theta^2} \frac{V_r}{r} \quad (6)$$

The values of $\partial V_r/(\partial \theta)$ and $\partial V_s/(\partial \theta)$ on the suction side of the wake of position 1 at $r/R_{tip}=0.9$ for $\phi=0.25$ is $1350s^{-1}$ $517s^{-1}$ respectively. It is expected that the radial velocity gradient in the θ -direction contributes comparable amount to the generation of r component of turbulence. From Fig. 16 it is found that $\partial V_r/\partial r$ is small compared to the other components. However $\overline{v_r^2}$ is far much larger than the other shear stress components, so it may cause $\partial V_r/\partial r$ contribute much to the generation of r component. From the rough investigation of the mean velocity gradient, it is found that the radial velocity due to the centrifugal force and the other secondary flow effects contributes significantly to the generation of turbulence. Detailed investigation of turbulence including the redistribution effects by rotation will be given in later separate paper in detail.

In general the gradient in the direction of s , $\partial/\partial s$

Fig. 19 Decay characteristics of the wake at $\phi=0.25$; (a) $r/R_{tip}=0.90$, (b) $r/R_{tip}=0.95$, (c) $r/R_{tip}=1.004$

in the wake is neglected since it is considered to be small compared with the $\partial/(\partial \theta)$. But the wake of a rotor decays very fast compared to that of single stationary airfoil. So it is needed to investigate whether the $\partial/\partial s$ can be neglected or not, especially related to the turbulence. $\partial V_s/\partial s$ at the suction side of the wake at $r/R_{tip}=0.90$ of position 1 is $57s^{-1}$. The value of $\partial V_s/(\partial \theta)$ at position 1 was $517s^{-1}$. In Eq. (4) $\partial V_x/\partial x$ contributes to turbulence in x -direction in the form of $-\overline{v_x^2} \partial V_x/\partial x$ and similarly $-\overline{v_s^2} \partial V_s/\partial s$ contribute to the generation of s component of turbulence. $\overline{v_s^2}$ is far much larger than the other stress in Eq. (4), so it is expected that $-\overline{v_s^2} \partial V_s/\partial s$ term affects significantly on the turbulence characteristic. The sign of this term is always negative in the wake region

Fig. 20 Decay characteristics of the wake at $\phi=0.30$; (a) $r/R_{tip}=0.90$, (b) $r/R_{tip}=0.95$, (c) $r/R_{tip}=1.004$

and reduce the s component turbulence. On the other hand it is expected that the effect of $\partial V_r/\partial s$ and $\partial V_\theta/\partial s$ is considered negligible as can be seen in Eq. (5) and (6).

The decay characteristics of the wake at three radial positions are shown in Fig. 19 and Fig. 20. Comparing the two cases of $r/R_{tip}=0.9$ and 0.95 , where distinct wake exists, the wake decays more quickly at $\phi=0.30$, which is consistent with the other results. It seems that the smaller width of the wake and larger velocity gradient in the wake cause the momentum transfer occur more rapidly. Also the larger free stream velocity can be another possible reason of the rapid mixing. The present experiment may also confirm the large radially inward velocity to contribute to the fast decay.

4. Conclusions

Detailed measurements in the flow field behind a highly forward-swept axial-flow fan at three operating conditions were performed using three-dimensional LDV system. The following conclusions can be drawn.

1) At the design condition ($\phi=0.25$), the leakage flow interacts with the pressure side inside the blade passage, thus directly influence the wake characteristics behind the trailing edge. The velocity of the wake edge at the pressure side is very low at the interaction region and the low momentum flow due to the leakage flow exists very widely which interacts with the wake and free stream moving downstream. At the design condition, distinct wake decays very quickly due to the interaction with the leakage flow.

2) At the low loading condition ($\phi=0.30$), radially inward flows exists at both the suction and pressure side in the wake near the tip region due to the effect of the leakage vortex.

3) At the low loading condition, the streamwise velocity at the leakage vortex region is lower than that of the velocity of the wake edge at the pressure side. This velocity defect is the characteristic of the leakage vortex.

4) As the loading increases, radially outward velocity increases and the position of its maximum value is close to the tip region.

5) At the design and high loading ($\phi=0.21$) conditions, trailing edge wake at the hub region decays very quickly due to the interaction with the large hub vortex. Also radially inward flow exists at the wake region near the pressure side, which is mainly caused by the hub vortex.

6) The rotation effect and secondary flows which cause radially outward flow in the wake region contributes significantly to the generation of radial-component turbulence.

7) The streamwise variation of the velocity in the wake cannot be neglected especially related to turbulence characteristic.

8) The wake at light loading condition decays faster than that at the design condition due to the rapid mixing and momentum exchange.

References

- (1) Reynolds, B., Lakshminarayana, B. and Ravindranath, A., Characteristics of the Near Wake of a Compressor of a Fan Rotor Blade, AIAA Journal, Vol. 17, No. 9 (1979), p. 959-967.
- (2) Ravindranath, A. and Lakshminarayana, B., Mean Velocity and Decay Characteristics of the Near and Far-Wake of a Compressor Rotor

- Blade of Moderate Loading, *Journal of Engineering for Power*, Vol. 102 (1980), p. 535-548.
- (3) Davino, R. and Lakshminarayana, B., Turbulence Characteristics in the Annulus-Wall Boundary Layer and Wake Mixing Region of a Compressor Rotor Exit, *Journal of Engineering for Power*, Vol. 104 (1982), p. 561-570.
 - (4) Prato, J. and Lakshminarayana, B., Investigation of Compressor Rotor Wake Structure at Peak Pressure Rise Coefficient and Effects of Loading, *Journal of Turbomachinery*, Vol. 115 (1993), p. 487-500.
 - (5) Hunter, I.H. and Cumpsty, N.A., Casing Wall Boundary Layer Development Through an Isolated Compressor Rotor, *ASME Journal of Engineering for Power*, Vol. 104 (1982), p. 805-818.
 - (6) Pandya, A. and Lakshminarayana, B., Investigation of the Tip Clearance Flow Inside and at the Exit of a Compressor Rotor Passage, Part I: Mean Velocity Field, *ASME Journal of Engineering for Power*, Vol. 105, No. 1 (1983), p. 1-12.
 - (7) Inoue, M. and Kuroumaru, M., Three-Dimensional Structure and Decay of Vortices Behind an Axial Flow Rotating Blade Row, *Journal of Engineering for Gas Turbines and Power*, Vol. 108 (1984), p. 561-569.
 - (8) Inoue, M., Kuroumaru, M. and Fukuhara, M., Behavior of Tip Leakage Flow Behind an Axial Compressor Rotor, *Journal of Turbomachinery*, Vol. 108 (1986), p. 7-14.
 - (9) Murthy, K.N.S. and Lakshminarayana, B., Laser Doppler Velocimeter Measurement in the Tip Region of a Compressor Rotor, *AIAA Journal*, Vol. 24 (1986), p. 807-814.
 - (10) Inoue, M. and Kuroumaru, M., Structure of Tip Clearance Flow in an Isolated Axial Compressor Rotor, *ASME Journal of Turbomachinery*, Vol. 111 (1989), p. 250-255.
 - (11) Store, J.A. and Cumpsty, N.A., Tip Leakage Flow in Axial Compressors, *ASME Journal of Turbomachinery*, Vol. 113 (1991), p. 252-259.
 - (12) Goto, A., Three Dimensional Flow and Mixing in an Axial Flow Compressor with Different Rotor Tip Clearances, *ASME Journal of Turbomachinery*, Vol. 114 (1992), p. 675-685.
 - (13) Stauter, R.C., Measurement of the Three-Dimensional Tip Region Flow Field in an Axial Compressor, *ASME Journal of Turbomachinery*, Vol. 115 (1993), p. 468-476.
 - (14) Lakshminarayana, B., Zaccaria, M. and Marathe, B., The Structure of Tip Clearance Flow in Axial Flow Compressors, *ASME Journal of Turbomachinery*, Vol. 117 (1995), p. 336-347.
 - (15) Copenhaver, W.W., Mayhew, E.R., Hah, C. and Wadia, A.R., The Effect of Tip Clearance on a Swept Transonic Compressor Rotor, *ASME Journal of Turbomachinery*, Vol. 118 (1996), p. 230-239.
 - (16) Suder, K.L. and Celestina, M.L., Experimental and Computational Investigation of the Tip Clearance Flow in a Transonic Axial Compressor Rotor, *ASME Journal of Turbomachinery*, Vol. 118 (1996), p. 218-229.
 - (17) Myung, H.J., Rew, H.S. and Baek, J.H., Measurements of Tip Clearance Flow in a Highly Forward-Swept Axial-Flow Fan, Submitted for Experiments in Fluids.
 - (18) Snyder, P.K., Orloff, K.L. and Reinath, M.S., Reduction of Flow Measurement Uncertainties in Laser Velocimeters with Nonorthogonal Channels, *AIAA Journal*, Vol. 22, No. 8 (1984), p. 1115-1123.
 - (19) Furukawa, M., Saiki, K., Nagayoshi, K., Kuroumaru, M. and Inoue, M., Effects of Stream Surface Inclination on Tip Leakage Flow Fields in Compressor Rotors, *ASME GT Paper 97-GT-43* (1997).
 - (20) Adkins, Jr. G.G. and Smith Jr, L.H., Spanwise Mixing in Axial-Flow Turbomachines, *Journal of Engineering for Power*, Vol. 104 (1982), p. 97-110.
 - (21) Gallimore, S.J. and Cumpsty, N.A., Spanwise Mixing in Multistage Axial Flow Compressors: Part 1—Experimental Investigation, *Journal of Turbomachinery*, Vol. 108 (1986), p. 2-16.
 - (22) Dring, R.P., Radial Transport and Momentum Exchange in an Axial Compressor, *Journal of Turbomachinery*, Vol. 115 (1993), p. 477-486.