Unsteady Pressure Measurements around Rotor of an Axial Flow Fan under Stable and Unstable Operating Conditions

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This study presents some measurement results on the unsteady pressure fields around rotor under stable and unstable operating conditions of an axial flow fan. The unsteady static pressure of rotor passage was measured by using high frequency pressure transducers mounted on the casing wall. The measurements on the unsteady total pressure at rotor inlet and outlet were also conducted with specially designed high frequency total pressure probe. Double Phase-Locked Ensemble Averaging Technique was used for analysis of pressure fluctuations around the rotor at rotating stall onset point. From the results, the unsteady pressure fields during stable and unstable operations of the axial fan were investigated and compared with each other. Particularly one period of rotating stall could be divided into two regions, stalled flow and unstalled flow region respectively. Furthermore the former could be also classified into two zones, bubbled and disturbed region by their features. The flow characteristics for each zone were described in detail and the static and total pressure fields were also analyzed in terms of the pressure distribution along pressure side and suction side on the blade tip profile.

Key Words: Axial Flow Fan, Rotating Stall, Unsteady Pressure Field, Bubbled Region, Disturbed Region, Double Phase-Locked Ensemble Averaging Technique

1. Introduction

Unsteady flow phenomena such as periodic rotating stall, which is one of the most serious instabilities, and broadband pressure fluctuation with non-periodic characteristics deteriorate the performance and stability of turbomachinery in vibration, noise and blade excitation. Thus, the mechanism of these unsteady phenomena is worth analyzing in the aspect of avoiding and inhibiting such instabilities. To understand and control the rotating stall, detail measurements and analysis on the structure of stall cell and flow characteristics inside stall cell are essentially required.

Many studies on the flow structure of the stall cell and the rotating stall for the turbo-compressor have been conducted since Emmons et al.(2) had described the propagation mechanism for the stall cell. Larguier(3) sketched the flow pattern in the stall cell by analyzing the unsteady pressure field measured on the casing wall of an axial flow compressor. Kato et al.(4) and Poensgen and Gallus(5) investigated the flow structure of the stall cell in a single stage axial flow compressor. Shiromi et al.(6) studied the behavior of the stall cell by measuring unsteady pressure in the diagonal flow fan and Soundranayagam and Elder(7) also studied on stall in a low hub-tip ratio fan. Shin and Kim(8) observed the flow characteristics at the impeller blade exit of the centrifugal compressor under rotating stall, which were analyzed by Double Phase-Locked Ensemble Averaging Technique.

Meanwhile, during stable operation of an axial fan, many investigations(9) – (12) on the leakage flow of the rotor have been conducted, which is largely affecting to performance and stall. Particularly Furukawa et al.(13) found out that the tip leakage vortex breakdown played a major role in the characteristic of the rotor performance at near stall conditions and gave a significant effect to the stall onset.

In this study, measurements and analysis of unsteady static pressure field on the casing wall and the variation of total pressure in span direction upstream and downstream of the rotor were carried out to understand the flow characteristics and structure of stall cell under rotating stall.

Nomenclature

bpf : blade passing frequency
\( f_{rs} \): rotating stall frequency  
\( f_s \): shaft frequency  
\( N \): number of sample, number of measured group  
\( p_i \): instantaneous pressure  
\( p_t \): total pressure  
\( P.S \): pressure side  
\( S.S \): suction side  
\( U \): rotor tip speed  
\( V_x, V_t \): axial and tangential velocity  
\( \Delta p \): pressure difference in the inlet and outlet of the rotor  
\( \phi \): flow coefficient  
\( \rho_{air} \): density of air  
\( \psi \): pressure coefficient

2. Test Facility and Instrumentation

An industrial axial flow fan was used as a test model in this study, which has ten rotor blades of adjustable pitch type as shown in Fig. 1 (a). The chord length of the rotor blade is 138 mm at hub and 107 mm at tip respectively. The rotor driven by 11 kW electric motor with a rotational speed of 1750 rpm has tip clearance of 1.5 mm and hub-tip ratio of 0.5. The casing has inner diameter of 710 mm.

Thirteen high frequency pressure transducers were mounted on the casing wall of the rotor passage as shown in Fig. 1 (b). The distance between each measuring point is 11 mm and the p1 is positioned at 38.5 mm upstream of the leading edge of the blade tip and 310 mm downstream of the inlet bell-mouth. The p13 is positioned at 49.5 mm apart from trailing edge. Figure 2 illustrates the schematic diagram for measurement. The system for unsteady pressure measurements on the casing wall consists of high frequency pressure transducer (Kulite, XCS-062), strain gauge amplifier (Instruments Group, 2260), low pass filter (Kron-Hite, 3384), A/D Board (Data Translation, DT3003-PGL) and waveform analyzer (Analogic, D6500E). Figure 3 shows the total pressure probe, which was traversed along spanwise to measure unsteady total pressure field in upstream and downstream of the rotor, p1 and p13 in Fig. 1 (b).

Instantaneous pressure signals were taken with the sampling rate of 0.083 msec (12 kHz), which is corresponded to 40 circumferential sampling points for one blade passage. A reference pressure signal for phase-locked averaging was also taken from the pressure transducer installed at the position of 60 degree in circumferential direction from p1, which was sampled with the same condition except being filtered with 100 Hz, which was decided from the propagation speed of the stall cell. In this investigation, Phase-Locked Ensemble Averaging Technique (PLEAT) was used as a post-process to analyze the measured signals. Details of measurement method and procedure were described in the previous study(14).
3. Fan Characteristics and Spectrum Analysis

The characteristics of the test fan are shown in Fig. 4 (a). Flow coefficient ($\phi$) and pressure coefficient ($\psi$) are defined as following Eqs. (1) and (2).

$$\phi = \frac{V_x}{U}$$  \hspace{1cm} (1)

$$\psi = \frac{\Delta p}{\rho_{air} U^2/2}$$  \hspace{1cm} (2)

Figure 4 (b) indicates RMS (Root Mean Square) values of measured pressure fluctuation using high frequency pressure transducers on the casing wall at the position (p13). RMS values are defined as following Eq. (3).

$$RMS = \sqrt{\frac{1}{N} \sum_{i=1}^{N} (p_i - p_{mean})^2}$$  \hspace{1cm} (3)

where, $p_{mean} = \frac{1}{N} \sum_{i=1}^{N} p_i$

where, $p_i$ is instantaneous pressure and $N$ means number of sample.

Figure 4 (c) shows the wall static pressure signals at p1 in the stable and unstable operating range and Fig. 4 (d) corresponds to their amplitude spectra. As shown in the characteristic curve of Fig. 4 (a), as the flow rate decreases further from $\phi = 0.370$, the pressure sharply drops at $\phi = 0.316$ where the rotating stall firstly occurs. At this time, as shown in (b), RMS value greatly increases. When the flow rate decreases further to $\phi = 0.262$, the pressure is restored and RMS value decreases. As shown in the wall static pressure fluctuation and its spectrum at $\phi = 0.389$ in Fig. 4 (c) and (d), the influence by blade passing dominates flow characteristics in stable operating range.

The stalled flow region is indicated as ‘A’ at $\phi = 0.316$ as shown in (d) and the un stalled flow region indicated as ‘B’ shows the similar pressure pattern with the stable operating range, $\phi = 0.389$. In the result of spectrum analysis for the stall onset point, the frequency component for the rotating stall and its harmonic frequency except the blade passing frequency newly appeared. The RMS value at this range suddenly increased by appearance of the rotating stall accompanied the pressure fluctuation with large amplitude. It is a factor of causing noise and vibration.

If flow rate is reduced further to $\phi = 0.262$, the feature of whole signal is changed to the type of less periodic random signal contrary to $\phi = 0.389$ and $\phi = 0.316$, and it involves broadband frequency spectrum. At this flow rate, the component of rotating stall exists as narrowband type in the broadband frequency and the amplitude suddenly decreases. As proved in the research by Haupt et al.(1), such broadband pressure spectrum is due to reverse flow effect. In this range, the propagation speed of the stall cell and its pressure fluctuation amplitude abruptly decrease from 16.3 Hz to 8.4 Hz and the RMS value also decreases as shown in (b).

4. Unsteady Wall Pressure Measurement

4.1 Stable operation

Figure 5 shows the phase-averaged wall static pressure distributions, which is normalized by dynamic pressure as shown in the denominator of the Eq. (2). Figure 5 (a) indicates pressure distributions along chordwise
(a) Along chordwise on blade tip (near stall, $\phi = 0.370$)

(b) On +12.5% of chord from leading edge with flow rate

Fig. 5 Distributions of phase-averaged wall static pressure

at the operation point near stall ($\phi = 0.370$) and (b) corresponds to those at 12.5% from blade leading edge with various flow rates. The pressure difference between the pressure and the suction side indicates the maximum value near the leading edge of the fan blade along chordwise. As the flow rate decreases to the operation point near stall, the pressure difference at 12.5% downstream from the blade leading edge gradually increases and reaches to about two times of that at maximum flow rate. The pressure difference in a blade passage leads to leakage flow through the blade tip clearance and flow instability can be caused by its development\(^{(15)}\).

Figure 6 illustrates the contours of RMS value for the wall static pressure at various flow rates. The RMS value can be defined by changing time mean value of the pressure in the previous Eq. (3), $p_{\text{mean}}$, into phase-averaged value. The RMS values are relatively very large at the suction side of the blade, which are denoted by white spots in the plots. As the flow rate decreases, they are gradually increased and move toward the blade leading edge. This is mainly influenced by the tip vortex intensifying due to increase of the pressure difference during the stable operation of the fan. Consequently as the flow rate is reduced to near stall onset point, intensity of the tip vortex due to leakage flow develops further and the flow instability is also increased.

4.2 Unstable operation

Figure 7 shows unsteady wall static pressure contour at stall onset, $\phi = 0.316$, which was generated by Double Phase-Locked Averaging Technique, and the variation of RMS value is shown at top of the figure, which could be obtained by the following Eq. (4).

$$\text{RMS}(s) = \sqrt{\frac{1}{N} \sum_{i=1}^{N} (p(s)_i - \text{DPLA}(s))^2}$$  \hspace{1cm} (4)

where, $p(s)_i$ is instantaneous pressure and DPLA(s) is phase-averaged value for $N$ measuring groups, which was treated by adjusting rotating stall and blade phase through the Double Phase-Locked Averaging Technique. Contrary to stable operating region where the pressure distribution was repeated in the blade passages and was influenced by the blade passing effect, it is affected by the stall cell in this case. The pressure distributions within blade passages are therefore characterized by the location of the rotating stall cell.

If the RMS value of the Eq. (4) is small, it means that the flow characteristics in the region seem to be on steady state similar to those at stable operating range as shown in the previous plot, Fig. 4 (c), while if the value is large, it means unsteady random signal. Therefore, one period of rotating stall can be divided into two regions according to the RMS value as illustrated in Fig. 7. One is the unstalled flow region denoted as ‘A’ where the RMS value is low. The other is the stalled flow region denoted as ‘B’ and ‘C’ where most of blades are influenced by the rotating stall cell. In this zone, the RMS value is relatively high compared to that of region A.

Figure 8 shows detailed static pressure contours on the rotor casing wall. Plot (a) means the pressure field observed at stable operating point near stall ($\phi = 0.370$) and (b) – (d) correspond to those at rotating stall onset,


\[ \phi = 0.316 \]

Figure 9 demonstrates the pressure distributions along the pressure and suction side of blade tip for typical case in three regions of A, B and C in one period of rotating stall.

As shown in the figure, the stalled flow region (region B and region C) is influenced by rotating stall, the pressure difference between the pressure side and the suction side of the blade significantly decreased, especially around the leading edge.

4.2.1 (a) Unstalled flow region (A)

In comparison of the case of the blade a3 and a4 in Fig. 9 (a) with that of \( \phi = 0.389 \) in the stable operating range, the distribution on the pressure side was maintained equally, while the pressure on the suction side increased. In this region, the pressure difference of both sides somewhat decreased quantitatively, but similar characteristics with the distribution at \( \phi = 0.389 \) of the stable operating range are shown. And the slope of the pressure distribution on the suction side behind 63\% of the chord length is decreased in the blade a4, which is closer to stalled flow region than the blade a3. This trend becomes more apparent in the blade b1 of the stalled flow region, such that nearly same pressure is maintained from the measuring point p7 to p9. It is believed that this result is due to separation bubble incurred on the suction surface of the blade.

4.2.2 (b) Stalled flow region (B, C)

This region could be divided into two regions according to their characteristics. The number of stall cells is one and the rotating direction is the same with that of the rotor. The characteristics in the two regions are summarized as follows:

a. Bubbled region (B): As shown in Fig.8 (c), high-pressure zone is concentrated near the measuring point p3 (17 mm upstream the leading edge) and relatively low pressure is distributed on the downstream of the trailing edge compared to other regions. This result is caused by the blockage effect of the stall cell. Comparing to the blade located at the unstalled region, for the pressure distribution of the blade b3 positioned nearly on the middle of this region, the pressure difference between both sides of the blade decreased significantly. This pattern is periodically repeated by rotating stall, and ultimately decreases the performance of the fan.
b. Disturbed region (C): In this region, the high-pressure zone on upstream of the leading edge, which was the dominant characteristics of bubbled region, disappeared and the pressure downstream the rotor is relatively high compared to bubbled region.

Assuming that the rotating direction of the rotating stall cell and that of the rotor is same, the incidence angle of blades ahead of stall cell is decreased, and those behind of the stall cell have an increased incidence angle. Therefore, the blades ahead of the stall cell are not stalled, while those behind of it are stalled\(^\text{16}\). As shown in the blade c3 of Fig. 9 (b), this region named as disturbed region and located behind of the bubbled region is under stalled condition, which means that the pressure difference between the pressure and the suction side decreased significantly like the bubbled region. And, as shown in blades (c3 and c9) of Fig. 9 (b), the lowest pressure peak on the suction side moved to p4 toward the leading edge, which demonstrates different pattern from the unstalled flow region. This pressure distribution shows a similar result with Schulz's investigation\(^\text{17}\) illustrated for the static pressure distribution on the blade surface of the annular compressor cascade with high incidence angle. As mentioned before, it is considered that this result is caused by increase of the incidence angle due to the influence of the stall cell blockage in the bubbled region. Comparing to Inoue’s observation\(^\text{18}\), this tendency is consistent with the result measured at corresponding position to the disturbed region.

In the blade c9, adjacent to the unstalled flow region, the pressure pattern on both sides of the blade becomes similar to that of the unstalled flow region. The pressure distribution of the blade a’3, which comes after the stalled flow region, has the similar feature to that of the stable operating range.

5. Total Pressure Measurement

Figure 10 shows total pressure distributions with flow rate, which were measured at 4 mm downstream of the rotor. Whereas there are no remarkable differences with flow rate in the distributions of the total pressure at mid-span and hub, some noticeable effects are observed near tip region.

From Fig. 10 (a), which is corresponded to the result measured at 15 mm from blade tip, as the flow rate is decreased toward near stall onset (\(\phi = 0.370\)), the total pressure near the pressure side of the blade steeply increases.
while the loss of the total pressure near the suction side relatively grows and expands. Therefore high blade loading is generated in each blade passage.

It is considered that the generation of the low total pressure zone near tip is due to the leakage interaction caused by mixing with the main flow. This interaction zone can be extended to about 15 – 20% of the blade span\(^{(15)}\).

Figure 11 shows time mean distributions of the total pressure in span direction upstream and downstream the rotor at flow rates corresponding to near stall (\(\phi = 0.370\)) and under stall condition (\(\phi = 0.316\)) respectively.

As shown in Fig. 11 (a), considering that the measured position is inlet of the rotor, it can be observed that the reverse flow of high-pressure zone exists near the tip upstream the rotor and high-pressure loss zone exists between 10% and 50% of blade span under stall condition. This result is because incoming flow is forced toward hub due to the blockage effect of stall cell near the casing, therefore the very low momentum flow zone is locally created between the blocked zone (the zone interacted by incoming flow and reverse flow near the casing) and incoming flow at mid-span. Consequently, this low momentum flow results in high-pressure loss.

The incoming flow between mid-span and hub, passed through the rotor inlet, is accelerated toward the casing downstream the rotor and relatively low absolute velocity in mid-span in order to preserve the mass conservation in the rotor inlet and outlet. Therefore, as shown in Fig. 11, the high-pressure zone is located near the tip and the large pressure loss zone is distributed in the mid-span.

Figure 12 (b) – (e) show the phase-averaged pressure patterns for instantaneous total pressure signals upstream and downstream the rotor (measuring position p1 and p13 depicted in Fig. 1) at various positions in blade span direction and Fig. 12 (a) is result of wall static pressure measurements on the casing wall at rotor inlet (p1) and outlet (p13). As shown in (b) – (e), the results clearly show that two regions, which is unstalled and stalled flow region observed in the result of wall static pressure measurements, exist in one period of rotating stall and repeats periodically.

In Fig. 12 (b), considering that the measured position is inlet of the rotor, it can be clearly observed that the stalled flow region is accompanied by strong reverse flow effect with high total pressure zone. As shown in (c), in comparison with the result at rotor inlet, it is observed that the stalled flow region enlarges in circumferential direction at rotor outlet. As shown in the descriptions on the leakage vortex breakdown\(^{(13),(19)}\) and in the vorticity contours illustrated in Hoying’s simulation\(^{(20)}\), it is believed that this circumferential enlargement is caused by the development of the blade tip vortex, flow separation and the growth of the boundary layer. Also, the mean pressure level and pressure fluctuation in stalled flow region is relatively high compared to that of the unstalled flow region.

Figure 12 (c) – (e) also show that stalled flow region is reduced in circumferential direction at mid-span compared to the result at tip. The effect of stall cell almost

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Fig. 11 Total pressure distributions at rotor inlet (a) and outlet (b) near stall (\(\phi = 0.370\)) and stall onset (\(\phi = 0.316\)) conditions

Fig. 12 Double phase-averaged static and total pressure distributions along spanwise at \(\phi = 0.316\)
6. Conclusions

The instantaneous measurements of wall static pressure and total pressure at inlet and outlet of the rotor blade were performed. From the measured unsteady flow fields, the characteristics of the rotating stall were discussed and they lead to the following conclusions.

(1) One period of rotating stall can be divided into three regions according to the characteristics of each region; unstalled flow region, bubbled region and disturbed region. In the disturbed region, the high-pressure zone appeared upstream of the leading edge in the bubbled region was not observed, but most of blades were on stalled condition. The lowest pressure peak on the suction side moved toward upstream to 25% of blade chord compared to the unstalled flow region.

(2) As the flow rate decreases to near stall onset point, the static pressure difference between pressure and suction side of the blade gradually increases and is maximized near the blade leading edge. At the same time the intensity of the tip vortex due to leakage flow develops further and the flow instability is affected by it.

(3) At the tip region, as the flow rate is decreased toward near stall onset, the total pressure at the pressure side strongly increases whereas the low total pressure zone is generated by leakage interaction caused by mixing leakage flow through the tip clearance with the main flow. Then high blade loading is generated in each blade passage.

(4) While the stalled flow region was accompanied with strong reverse flow near the tip at the inlet of the rotor, the region at rotor outlet was slightly enlarged in circumferential direction compared to that of rotor inlet and it was reduced at mid-span.

References

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