Unsteady Behavior of a Radial Fan in a Pulsating Flow Field

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Abstract

The unsteady behavior of a radial fan in a pulsating flow field was studied experimentally. In a pulsating flow, compressor and turbine operating points do not match their steady performance curve, but exhibit a hysteresis loop instead. The shape of the hysteresis loop depends on the pulsating frequency. Measurements to calculate the Bode diagram of the fan and the phase lag of pressure and flow rate indicate that the phase lag controls the hysteresis loop. Moreover, to clarify the influence of air column resonance, the diffuser pump test was performed as a basic study. Computational fluid dynamics (CFD) analysis of the radial fan was carried out, and its results were compared with the experiment results. The unsteady curve calculated by CFD formed a hysteresis loop as observed in the experiment results.

Keywords
Radial fan — Pulsating flow — Hysteresis loop

INTRODUCTION

To prevent global warming and conserve natural resources, it is important to improve automobile engine efficiency. Downsizing engines with turbocharging is considered an effective way to achieve this purpose. In general, a turbocharger is designed based on steady flow performance, while the actual exhaust and intake gas of an engine exhibit unsteady flow owing to the rapid opening and closing of engine valves. According to Mareli et al., under a pulsating flow, compressor and turbine operating points do not match their performance curve for a steady flow, but form a hysteresis loop instead.¹⁰ Yoshiki et al. performed a turbocharger turbine test under pulsating flow and discussed the influence of the shape of pulsating waves on turbine performance.⁹¹⁰ In fact, any turbo machinery, and not just the turbocharger, may exhibit the same phenomenon under pulsating conditions. Nevertheless, only few attempts have been made to understand this phenomenon. Therefore, the detailed mechanism of the hysteresis loop has not been clarified as yet. The purpose of this work is to clarify the mechanism of the hysteresis loop.

To accomplish this purpose, a radial fan in a pulsating flow was experimentally studied. Although a turbocharger attached to an automobile engine is operated in compressible fluid, the flow in the radial fan can be treated as that of incompressible fluid. In this work, for simplicity, the mechanism of the hysteresis loop in incompressible fluid was investigated first. In this experiment, the operating point of the radial fan and pulsating frequency were changed to investigate their influence on the hysteresis loop. The fan resistance is related to the instability of the fan and is calculated to investigate the influence on the hysteresis loop.

Moreover, a computational fluid dynamics (CFD) analysis of the radial fan with unsteady RANS code for the pulsating flow was performed. The CFD results were compared with the experiment results. The internal flow of the fan was computed for the boundary condition of pressure in the pulsating flow in the experiment. The comparison of internal flows under steady and unsteady conditions is discussed in this paper.

1. Test Apparatus and Method

1.1 Radial fan test apparatus

The experimental investigation was performed at the radial fan test facility. The test facility is shown in Figure 1. In the experiment, 11 impeller vanes and 9 return vanes were considered. The impeller had a diameter of 0.15 m. In this experiment, the pressure and velocity were measured at the inlet and outlet of the fan. For the measurement, pressure transducers and hot wires were set at the fan inlet and outlet. To plot the performance curve, the flow coefficient and pressure coefficient were calculated. Here, the flow rate was calculated by using the velocity at the fan inlet. In order to investigate the influence of the operating point on the hysteresis loop, the rotational speed of the impeller and flow rate at the fan outlet were changed. Because the Mach number at the inlet of the fan was approximately 0.07, the flow in the radial fan could be treated as incompressible fluid flow. The Reynolds number was about 1.9×10⁵. Rotational and stationary disks with holes were used as a pulse-generating device (Figure 2).
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1. Steady and Unsteady Performance Test

The pulse-generating device in the downstream region of the fan was used. The pulsating frequency could be changed freely by changing the rotational speed of the rotational disk. The steady performance and the unsteady performance tests were carried out. In these tests, the flow coefficient at the fan outlet was changed from 0.03 to 0.10 in steps of 0.01, and the rotational speed of the impeller was 3000 rpm. In the unsteady test, the frequency was changed from 10 Hz to 110 Hz in steps of 10 Hz. The time resolution of the measurements was 8 kHz.

Moreover, to grasp the transfer matrix of the fan, a Bode diagram was plotted. For this purpose, a frequency analysis of pressure and flow rate at the inlet and outlet was carried out.

1.2 Diffuser pump test apparatus

As a basic study on air column resonance, the diffuser pump test was carried out. To investigate the pressure pulsation in the diffuser pump piping system, a diffuser pump test apparatus shown in Figure 3 was used. The working fluid was air, and the Mach number at the impeller inlet was approximately 0.06; hence, the flow in the pump could be treated as incompressible fluid flow. The suction pipe and the discharge pipe were open to the atmosphere. There were 7 impellers and 13 diffusers. For measurement, the pressure transducers were set at the inlet of the impeller and at the shroud wall between the impeller and the diffuser, as shown in Figure 3. Frequency analysis was performed by using the unsteady pressure data obtained from the pressure transducers.

2. CFD Method

Three-dimensional (3D) unsteady CFD was performed for comparison with the experiment and to investigate the mechanism of the hysteresis loop. CFD software ANSYS/CFX15.0 was used for calculations. Reynolds-averaged Navier–Stokes equations were solved using the RANS-SST model. The rotor/stator interface was treated as a frozen-rotor in steady CFD or rotor-stator interaction in unsteady CFD. The boundary conditions were the flow rate at the inlet and the static pressure at the outlet.
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The static pressure at the outlet was pulsated in the form of a sine curve to calculate the flow under pulsating condition. In addition, all solid wall surfaces were considered as non-slip walls. The mesh of the fan is shown in Figure 4. The domains consisted of inblock, impeller, return vane, and outblock. There were about 16.2 million nodes in the entire fan stage, which includes 7.5 million in the inblock and outblock, 2 million in the impeller, and 6.7 million in the return vane. The average y plus was about 1.27 at the impeller and about 2.59 at the return vane. The frequencies of the pulsating flow were 10 Hz and 80 Hz, and the flow coefficient was 0.08.

There were about 31 timesteps in one rotation, i.e., about 11.7° per step. The number of iterations for unsteady CFD was 40.

3. Experimental Results and Discussion

3.1 Radial fan test result

The results of the experiment for the steady and unsteady conditions are shown in Figure 5. The steady performance curve and part of the unsteady curve forming the hysteresis loop changed when the flow coefficient and pulsating frequency were changed. The rotational speed of the impeller was 3000 rpm. It was confirmed that the operating point of the radial fan in the pulsating flow did not lie on the steady performance curve, but formed a hysteresis loop instead. In addition, Figure 6 shows the hysteresis loops when the time-averaged flow coefficient was changed from 0.03 to 0.10 in steps of 0.01 at a constant pulsating frequency of 40 Hz. The figure shows that the size of the hysteresis loops increased with flow coefficient, but the shapes of all loops appeared to be similar, regardless of the flow coefficient. On the other hand, pulsating frequency appeared to influence the shapes of the hysteresis loops. Figure 7 summarizes the hysteresis loops when pulsating frequency was changed from 10 Hz to 110 Hz in steps of 10 Hz for a constant time-averaged flow coefficient of 0.10 for clarifying the influence of pulsating frequency in detail. The shapes of the hysteresis loops were confirmed to depend on pulsating frequency. While the hysteresis loop at low pulsating frequency followed the steady performance curve, the higher the pulsating frequency, the larger was the deviation of the hysteresis loop from the steady performance curve. At the maximum pulsating frequency in this experiment, 120 Hz, the angle formed by the steady performance curve and the long axis of the hysteresis loop was approximately 90°.
Moreover, to find the transfer matrix of the fan, a Bode diagram was plotted. Figure 8 shows the Bode diagram of the radial fan for an impeller rotation speed of 3000 rpm and a time-averaged flow coefficient of 0.10. Pulsating frequency was changed from 10 Hz to 110 Hz in steps of 10 Hz. The diagram shows that while the pressure magnitude decreased and the phase lag between \( P_1 \) and \( P_2 \) remained more or less constant, the flow rate magnitude attained its minimum value at approximately 55 Hz, and accordingly, the phase lag between \( Q_1 \) and \( Q_2 \) increased. Possible causes include the fact that pulsating frequency is related to the phase lag, and the phase lag is related to the shape of the hysteresis loop. Otherwise, the influence of air column resonance will be reflected in the Bode diagram. This air column resonance will be discussed below.

For investigating the origin of pulsating frequency characteristics, the time variation in total pressure coefficient and flow coefficient are shown in Figure 9. Pulsating frequency was set as 10 Hz, 40 Hz, and 80 Hz. It was clarified that there existed a phase lag between these two waves. To determine the relation between the lag and pulsating frequency, a cross spectrum was calculated. Figure 10 shows the phase lag between the total pressure coefficient and flow coefficient. Pulsating frequency was changed from 10 Hz to 110 Hz in steps of 10 Hz, and flow coefficient was changed from 0.03 to 0.10 in steps of 0.01. This indicates that the phase lag decreased with pulsating frequency. In addition, when pulsating frequency was less than about 50 Hz, the phase lag increased with the flow coefficient; on the other hand, when pulsating frequency exceeded 50 Hz, the phase lags showed agreement regardless of flow coefficient. Thus, it can be concluded that these phase lag characteristics control the shape of hysteresis loop, as shown in Figure 7.
3.2 Air column resonance test result

To clarify the influence of air column resonance, the diffuser pump test was carried out as a basic study. Figure 11 shows the result of the frequency analysis for frequencies under 200 Hz. The rotational speed of the impeller ranged from 500 rpm to 2192 rpm. As seen from Figure 11, the dominant frequencies at the impeller inlet and outlet were 21 Hz, 40 Hz, and 64 Hz. Moreover, the length of the suction pipe was set as 0 mm, 1300 mm, and 2280 mm, and similar measurements were performed to investigate the relationship between the piping length and the resonance frequency. Table 1 lists the theoretical values and the actual measurements for each piping length. Theoretical values were calculated by using expression (1). Here, the inlet of the suction pipe and the outlet of the discharge pipe were treated as open end, and the pump was replaced by the equivalent length. According to the data in Table 1, the actual measurements agreed approximately with the theoretical values. Therefore, these dominant frequencies were the air column resonance frequencies.

\[ f_n = \frac{nC}{2l_{total}} \]  

(1)

The Bode diagram of the radial fan (Figure 8) was analyzed based on this basic study. As indicated in the figure, the flow rate magnitude was minimum at about 55 Hz. To examine whether the air column resonance frequency was related to this characteristic, the theoretical value of the resonance frequency was calculated. The inlet of the suction pipe and the pulsation-generation device were treated as open end. Using expression (1), the first mode of the resonance frequency was found to be 45 Hz. It is concluded that air column resonance is not the cause of the characteristics of the Bode diagram.

4. CFD Results and Discussion

Figure 12 shows a comparison of the steady performance curve calculated by CFD and the experimental results. The CFD results agreed with the experimental results qualitatively, but there were a little quantitative deviation between the CFD and experimental results. This deviation is attributed to the insufficient number of mesh at the impeller outlet. The unsteady CFD results are shown in Figure 13. Pulsating frequencies were set as 30 Hz, 60 Hz, and 90 Hz. It was revealed that the unsteady curve calculated by CFD formed a hysteresis loop as in the case of the experiment results.
5. Conclusion

The results of investigation of the unsteady behavior of the radial fan in a pulsating flow field lead to the following conclusions.

- The performance curve of the radial fan in pulsating flow was not on the steady performance curve, but a hysteresis loop.
- The size of the hysteresis loops increased with the flow coefficient, but the shapes were independent of the flow coefficient.
- The shapes of the hysteresis loops depended on pulsating frequency.
- The phase lag of pressure and flow rate at the fan inlet controlled the shape of the hysteresis loop.
- The air column resonance is not related to the shape of the hysteresis loop.
- The unsteady curve calculated by CFD also showed a hysteresis loop, as in the experiment results.

NOMENCLATURE

\[ A \] Cross-section area [m^2]
\[ B \] Height of the impeller at the outlet [m]
\[ D \] Diameter of the impeller [m]
\[ l \] Length [m]
\[ Re \] Reynolds number \( = \frac{UD}{\nu} \)
\[ P \] Pressure [Pa]
\[ Q \] Flow rate \( = AV \) [m^3/s]
\[ U \] Rotational velocity of the impeller at the outlet [m/s]
\[ V \] Velocity [m/s]
\[ N \] Rotational speed [rpm]
\[ \omega \] Angular velocity [rad/s]
\[ f \] Frequency [Hz]
\[ C \] Speed of sound [m/s]
\[ n \] Integer value
\[ \tau \] Torque [N \cdot m]
\[ \rho \] Density [kg/m^3]
\[ \nu \] Kinematic viscosity [m^2/s]
\[ \phi \] Flow coefficient \( = \frac{Q}{\pi DBU} \)
\[ \psi \] Total pressure coefficient \( = \frac{P_1 - P_2}{\rho U^2} \)

ACKNOWLEDGMENTS

The continued support of SIP is greatly appreciated. We acknowledge the help of Mr. Hiramatsu, Mr. Komaki, and Mr. Miwa, in particular, for providing their experimental data for use in the studies.

REFERENCES
