Performance and Operational stability of Axial-flow Fan with Upstream Filter

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Abstract

In this study, an anisotropic filter was installed between a disk-shaped obstacle and fan, with the disk on the fan intake side, and the effects of the difference in transmissivity on fan performance and the occurrence of flow instability were investigated both experimentally and analytically. The results showed that the anisotropic filter suppressed the flow instability with periodic pressure fluctuations that occurred in the low-flow region. In addition, when a coarse filter with the highest permeability in this condition range was installed, a larger differential pressure was observed over the entire flow range compared with the case with only a disk. It was thus confirmed that the filter was effective in both suppressing flow instability and improving fan performance.

Keyword: *Axial-flow fan, Flow instability, Performance curve, Swirl flow*

1. Introduction

Because of their high flow rate and low differential pressure, large axial fans are used for ventilation inside buildings such as factories and in tunnels. In addition, because of their simple structure and ease of miniaturization, they are also used for cooling heating elements inside high-density spaces such as the inside of personal computers. Some of these fans are equipped with a dust proof filter on the fan intake side to prevent dust and other particles from passing through the fan, contaminating the fan blades, and entering the structure. Incidentally, it is known that the installation of a filter or structure on the intake side of an axial-flow fan reduces the airflow-pressure characteristics compared to an environment where there are no obstacles around the fan (such as during performance tests before shipping) [1]-[4]. Assuming an environment where there is an obstacle on the fan intake side, such as a desktop PC, an obstacle is installed on the fan intake side, and the effect of the disk diameter and distance from the fan (distance between obstacles) on fan performance has been investigated, and it has been reported that fan performance drops significantly when the diameter is above a certain level and the gap is below a certain level . Furthermore, the occurrence of flow instability, in which disturbances with pressure fluctuations propagate in the circumferentially at low flow rates and disturb stable continuous operation [1-2]. Our research group has investigated for suppression of the flow instability. For example, by placing a cross plate between the fan and the disk obstacle and weakening the two-dimensionality of the swirling flow between the two plates, the pressure fluctuation caused by the disturbance was effectively damped, but the flow instability has not been eliminated yet [3]. In addition, the coarser cross plates cannot be expected to provide dust suppression, and other filter are required, which may lead to an increase in pressure loss, thus providing few advantages in practical applications. Therefore, we proposed a method of installing an isotropic porous filter with a dust suppression function on the fan intake side between the disk obstacle and the fan [4], and succeeded in suppressing the flow instability under a wider range of conditions than when the cross plate was installed. However, the obtained results were fragmentary and did not lead to a systematic investigation of the optimal filter porosity, characteristics (anisotropic or isotropic), and installation position.

In this study, a filter was installed on the fan intake side in the presence of a disk obstruction, and the effects which the roughness of the filter and the distance between the obstruction and the fan influence the fan performance and the occurrence of the flow instability were investigated experimentally and analytically. In this paper, three types of anisotropic filters with different permeability (pressure drop) were investigated under the constant conditions of disk diameter D_d = 320 mm, fan diameter D_f = 130 mm, and dimensionless disk-to-fan inlet distance G/D_f = 0.15, where the instability of interest is generated with reference to previous studies [4]. Performance tests were conducted to draw flow-pressure difference curves and to measure the unsteady velocity and pressure fields between the disk and the fan inlet.

Experimental and CFD Setup

Fig. 1 shows a schematic of the experimental apparatus. The test fan (Oriental Motor MS14-BC, hub diameter D_h = 75 mm, number of blades: 5) is incorporated into a plenum tank, with an anisotropic filter and a disk-shaped obstacle above the fan. Air, the working fluid, is drawn from the atmosphere by the test fan into the channel between the disk and fan.

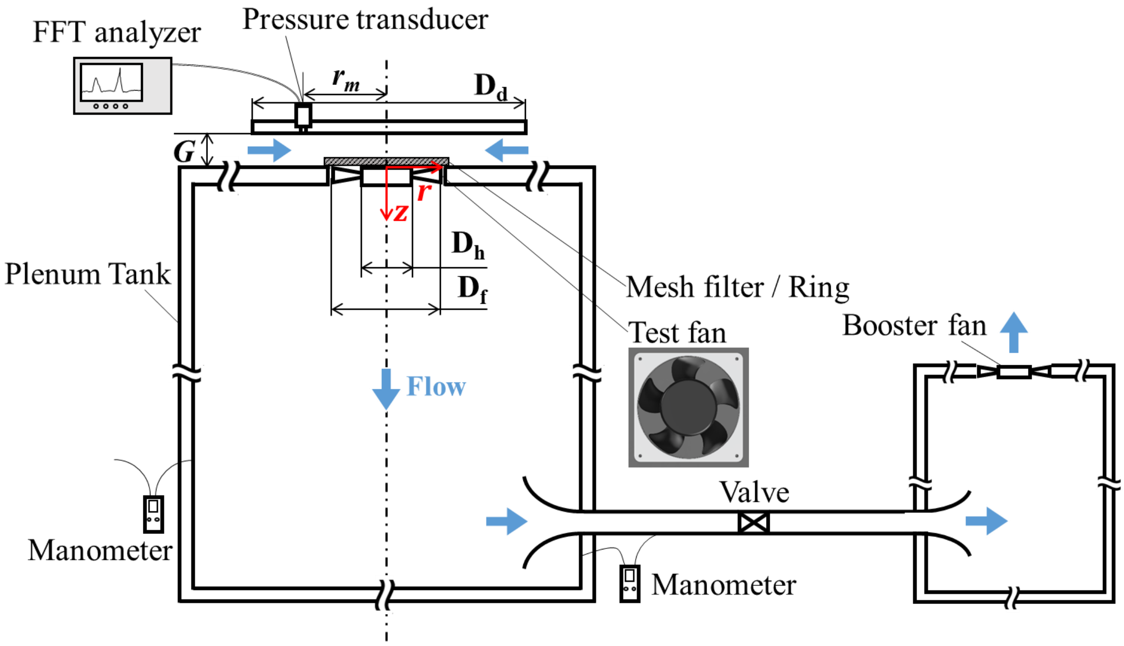


Fig. 1 Experimental apparatus

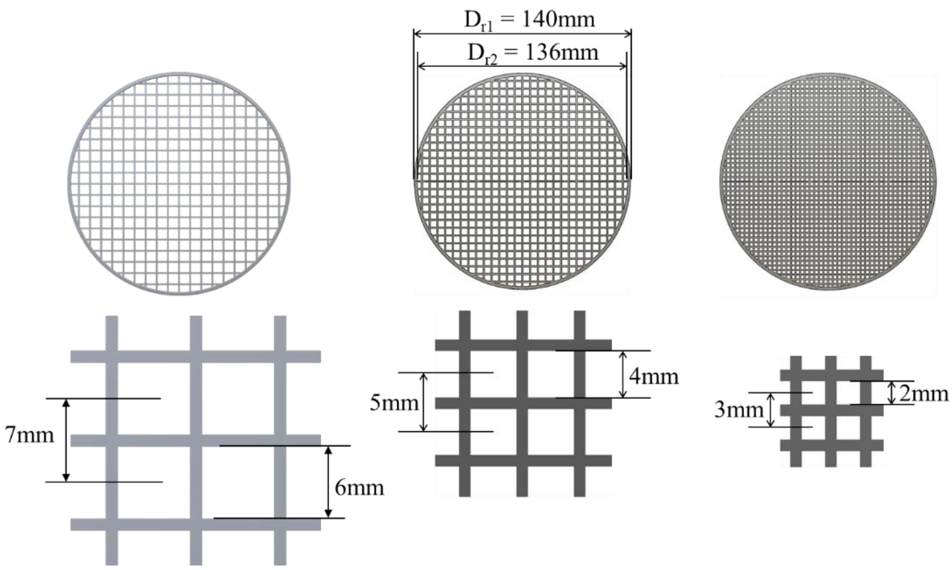


Fig. 2 Specifications of installed item near the axial-flow fan inlet:
(a) coarse, (b) medium, and (c) fine filters

A pre-swirling flow is generated in the channel and released into the plenum tank after passing through the fan. In this experiment, the fan speed was varied from 2000 to 2600 rpm, and a manometer (DPC-201N, Okano Corporation) was used to measure the pressure difference. The flow rate Q was regulated by the rotational speed of the booster fan and the valve opening, as illustrated in Fig. 1. Q was determined by multiplying the spatial average velocity, calculated from the differential pressure between the plenum tank and the downstream bellmouth immediately, by the passage cross-sectional area. The fan inlet–outlet pressure difference ΔP_s is the time-averaged pressure difference between the atmosphere and plenum tank. The fan performance was evaluated the flow coefficient Φ – pressure coefficient Ψ curve based on the blade tip peripheral velocity $U_{\theta T}$. Φ and Ψ were calculated as shown in Equations (1) and (2), respectively.

$$\Phi = \frac{Q}{U_{\theta T}\pi(D_f^2 - D_h^2) / 4} \tag{1}$$

$$\Psi = \frac{\Delta P_s}{0.5\rho U_{\theta T}^2} \tag{2}$$

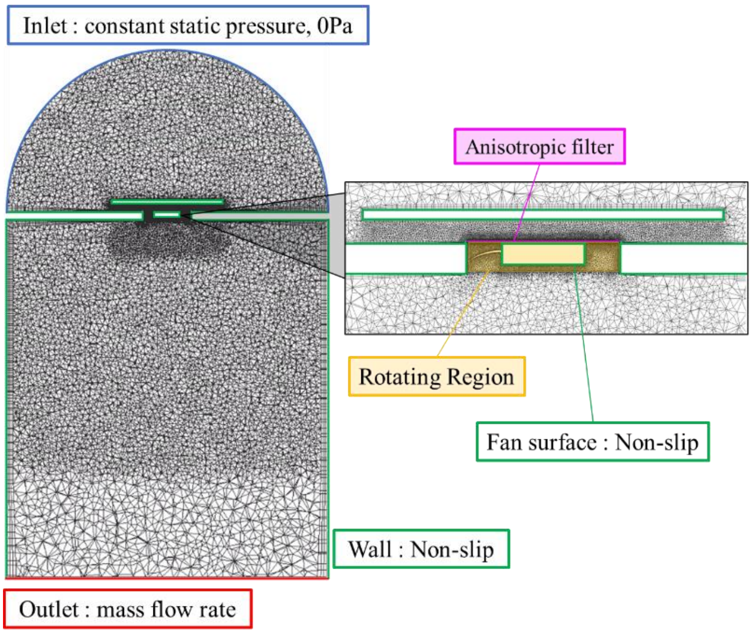


Fig. 3 Boundary conditions and mesh for CFD analysis of filter with upstream obstacle

where ρ is the air density. The pressure fluctuations associated with the flow instability occurring in the channel between the disk and fan were measured by connecting a pressure transducer (sampling frequency: 128 Hz) to the static pressure holes in the disk, and a fast Fourier transform analyzer was used for spectral analysis of the obtained time waveforms.

Fig. 2 shows the lattice geometries of three different anisotropic filters (outer diameter Dr_1 = 140 mm, frame width 2 mm, and thickness L = 1 mm) with different permeabilities, where (a), (b), and (c) represent the coarse, medium, and fine filters, respectively. The velocity V and pressure drop ΔP_L through the filter installed in the straight pipe channel, obtained in preliminary experiments. The following equation, which are based on the Darcy–Forchheimer law, was used to calculate the permeability K_P and coefficient of inertia resistance K_L.

$$\Delta P_L = \frac{\mu L}{\mathrm{K_P}} V + \mathrm{K_L} \frac{\rho L}{2} V^2 \tag{3}$$

where μ is the viscosity coefficient, ρ is the air density, and V is the velocity in the pipe.

Fig. 3 shows the meridian cross section and boundary conditions of the analysis model with a filter. The flow field was assumed to be a 3-dimensional incompressible viscous flow, and the general-purpose code SC/Tetra V14 (Software Cradle Inc.) was used for the unsteady calculations (governing equations: Unsteady Reynolds-Averaged Naiver–Storks (URANS) equations, turbulence model: standard k-ε model, total number of meshes with obstacles only: approximately 5.3 million, total number of meshes when the fine filter is installed: 11.8 million). As boundary conditions, a constant static pressure was applied to the open area upstream of the fan, a constant mass flow rate was applied to the bottom of the plenum as the outlet of the computational domain, and non-slip conditions were applied to the walls (fan, disk, and plenum tank surface). The anisotropic filter specifies a resistance coefficient C_f and a power law α [5], which were calculated assuming that the V-ΔP_L relationship as obtained from the preliminary experiment described above is a power law approximation as follows.

$$\frac{\Delta P_L}{L} = \frac{1}{2} \rho C_f V^{\alpha} \tag{4}$$

For simplicity, the fan speed was kept constant at 2500 rpm, and discontinuous joints were used in the fan rotation region.

Results and Discussion

Fig. 4 shows the Φ–Ψ curves obtained under the constant G/D_f = 0.15. For reference, the results for $G/D_f = \infty$, where no disk exists, are also shown. As in the previous studies [1-4], where only a disk was installed, Ψ was lower in the entire flow range compared with the case with no disk. In the low flow region, $\Phi \lesssim 0.1$, the case with only a disk had an ascending characteristic, and then it exhibited a significant decrease in Ψ. In addition, it is experimentally confirmed that the flow instability in this study occurs within the same range.

When an anisotropic filter is installed in addition to the disk, the remarkable positive-sloping characteristic observed when only a disk is used is not observed. In addition, the negative-sloping characteristic is generally observed even in the low-flow range, which indicates that stable operation is being achieved. The performance of the fine and medium filters

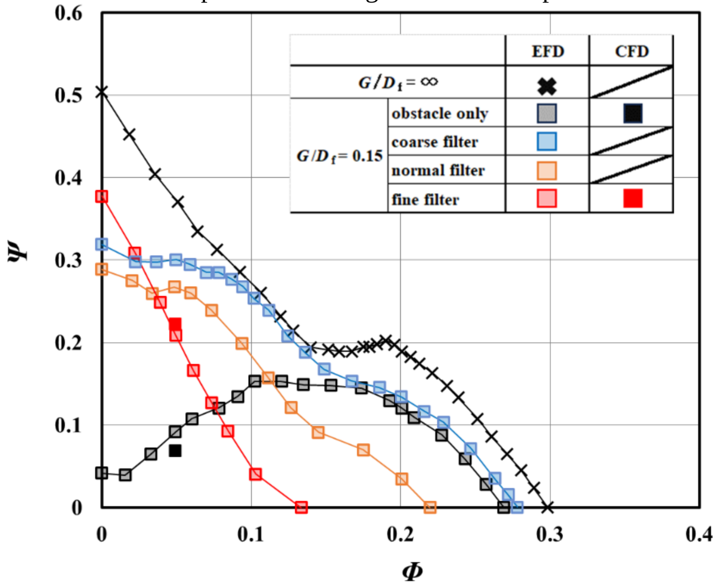
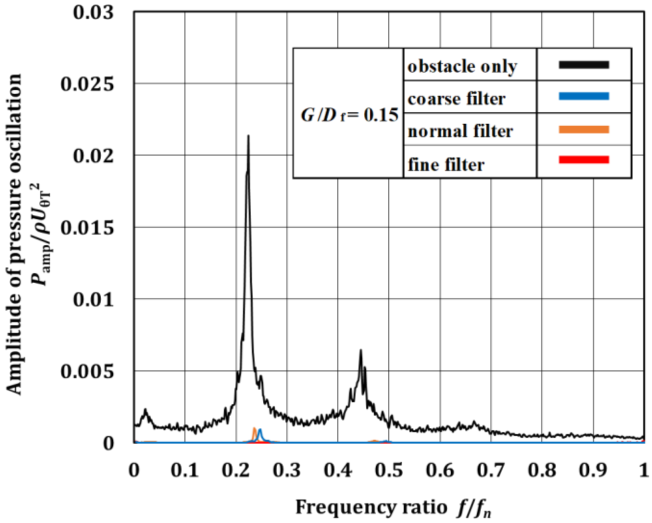


Fig. 4 Pressure performance curves experiment and CFD



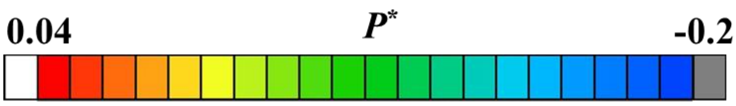
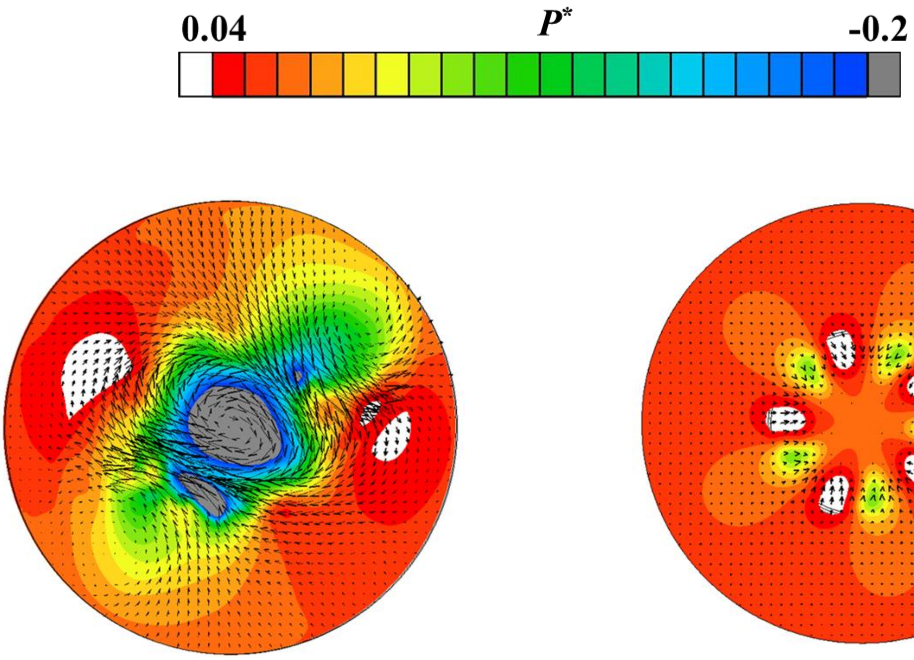


Fig. 5 FFT result for $G/D_f = 0.15$ at $\Phi = 0.049$ from experiment ($r_m/D_f = 0.85$)

exceeded that of the disk-only case in the low flow rate range, and the performance of the coarse filter exceeded that of the disk-only case in the entire flow rate range. However, as the filter permeability decreases, the pressure drop increases, and the maximum flow coefficient at $\Psi = 0$ and the maximum pressure coefficient at $\Phi = 0$ decrease.

Fig. 5 and 6 show the spectrum analysis results of the experimental pressure fluctuation waveform ($r_m/D_f = 0.85$) and the instantaneous velocity vector and static pressure distribution in the r-θ cross-section between the disk and fan inlet ($z/D_f = -0.077$) obtained from the CFD analysis under the conditions $G/D_f = 0.15$ and $\Phi = 0.049$, respectively. Comparing the results for the case with only the disk and the case with a fine filter in addition to the disk, Fig. 5 shows that the dimensionless frequency $f/f_n \approx 0.22$ (where f_n is the rotation frequency of the fan) is almost completely suppressed when the fine filter is installed.

This corresponds to the fact that, in Fig. 6(a), flow instability is observed with a cell structure in which a high-pressure and low-pressure region appear in pair. However, in Fig. 6(b), when a fine filter is installed, only the pressure distribution owing to the passing of the five rotor blades is observed, and no flow instability is generated. In addition, focusing on the difference in transmissivity, Fig. 5 shows that the coarse and medium filters were able to suppress the dominant component to approximately one-twentieth of that observed with the disk alone, although not to the extent of eliminating it, indicating that the anisotropic filter is effective in suppressing the flow instability.



(a) Obstacle only (b) With fine filter

Fig. 6 Pressure and velocity vector distribution from a top view (z/D_f = -0.077), for G/D_f = 0.15 at Φ = 0.049, from the CFD analysis

The results obtained in this study confirm that the coarse filter is the best choice because it improves fan performance over the entire flow range and successfully suppresses flow instability, resulting in a more stable operation than that with the disk alone.

Conclusion

In this study, an anisotropic filter was installed between an obstacle and fan in an environment with the obstacle on the fan intake side (G/D_f = 0.15), and an attempt was made to clarify the effect of the difference in transmissivity on fan performance and the occurrence of flow instability. The results of both experiments and CFD analysis confirmed that the installation of a coarse filter with the highest permeability in this condition range can suppress the flow instability with periodic oscillations and improve fan performance. However, further

r investigations are required to determine the optimal anisotropic filter permeability and installation position.

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