

Compressible Simulation of Flow and Sound around a Small Axial-Flow Fan with Flow through Casing Slits

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ABSTRACT

Small axial fans are used for cooling electronic equipment and are often installed in a casing with various slits. Direct aeroacoustic simulations and experiments were performed with different casing opening ratios to clarify the effects of the flow through the casing slits on the flow field and acoustic radiation around a small axial fan. Both the predicted and measured results show that aerodynamic performance deteriorates at and near the design flow rate and is higher at low flow rates by completely closing the casing slits compared with the fan in the casing with slits. The predicted flow field shows that the vortical structures in the tip vortices are spread by the suppression of flow through the slits at the design flow rate, leading to the intensification of turbulence in the blade wake. Moreover, the pressure fluctuations on the blade surface are intensified, which increases the aerodynamic sound pressure level. The suppression of the outflow of pressurized air through the downstream part of the slits enhances the aerodynamic performance at low flow rates. Also, the predicted surface streamline at the design flow rate shows that air flows along the blade tip for the fan with slits, whereas the flow toward the blade tip appears for the fan without slits. As a result, the pressure distributions on the blade and the torque exerted on the fan blade are affected by the opening ratio of slits.

1. Introduction

Small axial-flow fans are widely used to cool electronic equipment, such as personal computers and video cameras. Past research [1,2] showed that the tip clearance flow affects the aerodynamic and aeroacoustic performance of an axial-flow fan. Longhouse [1] showed that fan peak efficiency dropped with increasing clearance and that tip clearance noise contributed to the overall noise up to 15 dB. Fukano and Jang [2] measured the flow of two tip clearances at 1.6% and 3.5% and showed that the velocity fluctuations of the tip leakage vortex become more intense with a larger tip clearance, where both the discrete frequency and broadband noise levels were increased.

The flow fields around the blade tip have been studied [3-6]. Camussi *et al.* [3] measured the flow field around a stationary isolated blade installed between endplates with a tip clearance and presented the relationship between the flow perturbation generated at the tip and far-field sound pressure. Also, the turbulence structure of the tip leakage flow around a cascade of blades was investigated by Muthanna *et al.* [4]. Recently, simulations have been used to clarify the flow field. Boudet *et al.* [5] predicted the flow around the tip by compressible large-eddy simulations (LES) and investigated the wandering of tip leakage vortices at the Mach number based on the tip velocity of 0.22. Moghadam *et al.* [6] clarified the effects of tip clearance and flow rate on the tip leakage vortex by compressible LES at the Mach number of 0.136.

Various passive or active control methods of the flow around the blade tip have been investigated to enhance aerodynamic and aeroacoustic performance [7-11]. Aktürk and Camci [7] modified the blade shape in one such attempt at a passive control methodology. Tip treatment with a bump on the pressure side was effective for reducing

the tip leakage mass flow rate. Moreover, the methodologies for active control of the tip clearance flow have also been studied [8,9]. Morris and Foss [8] enhanced fan performance for middle-to-high flow rates by delivering high-momentum air into the tip clearance region using a shroud with a pressurized plenum. Neuhaus and Neise [9] conducted a study on the steady injection of compressed air through slits installed in casing to improve aerodynamic and aeroacoustic performance.

The control methodologies for modification of the tip clearance flow without extra input energy have been also investigated [10,11]. Eberlinc *et al.* [10] presented a way to increase aerodynamic performance with self-induced trailing edge blowing, where the internal flow in a hollow blade was used. Also, by the incompressible flow simulation and aeroacoustic analysis, Itagaki *et al.* [11] predicted the aerodynamic performance and level of radiated sound for the fan in the casing with slits, where the slits were installed for the enhancement of the aerodynamic performance. However, in the actual installation of the fan for industrial products, the slits can be partially or completely closed due to other components in the products. The variation of flow field around a fan due to the suppression of the flow through the slits has not been sufficiently clarified.

The objective of this study is to clarify the effects of the opening ratio of casing slits on the flow field and acoustic radiation around an axial-flow fan. Direct aeroacoustic simulations based on the compressible Navier–Stokes equations were performed in the experiments for this study. The first novelty of this paper is to clarify the detailed flow and aeroacoustic phenomena related with the flow through the casing slits including the effects of closing the slits for the first time. This knowledge contributes to the development of fans for enhancement of aerodynamic and aeroacoustic performances

considering the various installation conditions. Moreover, another novelty of this paper is to show the usefulness of the direct aeroacoustic simulations at the low Mach number based on the tip velocity of 0.017, where the flow and acoustic radiation can be directly predicted. This Mach number is the lowest value comparing with the past direct aeroacoustic simulations of fans to the knowledge of the authors. The present in-house computational methodologies are validated by the experimental results.

2. Methodologies

2.1 Flow Configurations

Figure 1 shows the configuration of a small axial-flow fan in a casing. As shown in Fig. 1, the number of rotor blades is $Z_b = 5$, and four struts ($Z_s = 4$) are installed downstream of the rotor. The blade shape was determined referring to the literature [11] and the details of the shape are shown in Fig. 1. The blade chord length at the blade tip is $C = 22.3$ mm. The fan diameter is $D_f = 37$ mm, and the hub-to-tip ratio is $D_h/D_f = 0.49$. The outer side length of the casing is $L = 40$ mm, and the inner diameter of the casing is $D_s = 39$ mm. The tip clearance is $\sigma = (D_s - D_f)/2 = 0.04C$. The cross-section of the duct is a square with the side of $4L$.

The width of the slit channels in the casing is 3 mm, as shown in Fig. 1, where the slit shape is determined according to the literature [11]. Three casings with different opening slit ratios of $R = 0.0$ (no slit), 0.17, and 0.34 are used to clarify the effects of the flow through the slits on the flow field around the fan. The above-mentioned blade shape was designed for the slit conditions of $R = 0.34$, and the conditions of $R = 0.0$ and 0.17 are corresponding to the cases with the slits completely and partially closed. For $R = 0.17$ and 0.34, the opening ratio is adjusted by changing the spacing between the slits and the number of slits.

Table 1 shows the parameters of the operating and flow conditions. The rotational speed is $n = 50.4$ s⁻¹, and the blade tip velocity is $V_{\text{tip}} = 5.86$ m/s. The Reynolds number, based on the blade tip velocity and chord length, $\text{Re} = V_{\text{tip}}C/\nu$, is 8,600, and the Mach number, based on the blade tip velocity, is 0.017. In the experiments, aerodynamic performance was measured in the range of the flow coefficient of $\phi \equiv Q/(A \cdot V_{\text{tip}}) = 0.0$ –

0.31, where Q is the volumetric flow rate and A is the flow passage area of 0.001 m^2 . The computations were performed at four flow rates: a low flow rate, the design flow rate ($\phi = 0.19$), a flow rate slightly greater than the design flow rate, and the maximum flow rate.

2.2 Experimental Methods

2.2.1 Experimental setup

Figure 2 shows the experimental setup. The origin of the coordinate is set to the center of the front surface of the hub. The x -axis is set along the axial direction of the fan, the y -axis is along the vertical direction, and the z -axis is normal to the x -axis and y -axis. As shown in Fig. 2, the fan was installed in a flow duct. A bent duct and a duct with sound-absorbing material were installed downstream of the fan to reduce the noise from the outside of the experimental equipment.

In the test facility for fan performance measurement specified in JIS B 8330 corresponding to ISO 5801, a duct with a circular cross-section with the same diameter of the fan is used. In this test, the fan was installed in the duct with a relatively larger rectangular cross-section with the side length of $4D$ considering the flow through the casing slits.

2.2.2 Measurement methodologies

The flow rate was evaluated by using the orifice flow meter installed downstream of the test section referring to ISO 5167-1. The pressure difference across the orifice plate was measured by differential manometer (DPC-500N12, OKANO WORKS, LTD). Aerodynamic performance was evaluated at each flow rate by the pressure difference

across the baffle plate, ΔP , measured by the above-mentioned differential manometer at the positions of $x/D = -1.0$ and 3.0 . The duration time and sampling frequency of the measurement are 300 s and $2,000$ Hz. For the evaluation of the aerodynamic performance, the static pressure coefficient, ψ , was computed as follows:

$$\psi = \frac{2\Delta P}{\rho V_{\text{tip}}^2}, \quad (1)$$

The sound pressure level was measured by a 1/2-inch microphone (UC-53A, RION CO., LTD) that was flush-mounted along the upstream duct wall at $x/D = -2.5$. Compared with the sound measurement methodology specified in JIS B 8346 corresponding to ISO 3744-1981 and 3746-1979, this measurement position was set closer to the fan in order to measure relatively weak aerodynamic noise from the small fan at high accuracy.

The velocity fluctuations in the wake of the blade were measured by a hot-wire anemometer at the radial position of $r/(D/2) = 0.92$. The duration time of the measurement was 30 s, and the sampling frequency was $64,512$ Hz ($1,280$ n).

2.2.3 Measurement uncertainties

The main sources of the uncertainties of the flow coefficient and pressure coefficient were the uncertainties of measuring the pressure difference across the baffle or orifice plate, the rotational speed, and the fluid density. Referring to the literature [12], the uncertainty of flow coefficient was estimated to be 0.0050 , which was corresponding to 2.6% of the measured coefficient at the design flow rate ($\phi = 0.19$) for $R = 0.34$. The uncertainty of the pressure coefficient was 0.0045 , which was corresponding to 2.2% of the measured pressure coefficient at the design flow rate.

To estimate the uncertainty of measuring the sound pressure level, the repeatability variation of the measured sound pressure level was investigated by 10 measurements. The variation of the sound pressure level at the blade passing frequency of nZ_b was 1.7 dB. Also, the uncertainty of the measurement position of the hot-wire anemometer was estimated to be 0.2 mm. The uncertainty of the turbulence intensity due to this effect was estimated by utilization of the computational results and found to be $0.002V_{\text{tip}}$, which was corresponding to 7% of the measured turbulence intensity ($R = 0.34$).

2.3 Computational Methods

2.3.1 Governing equations and finite difference scheme

The three-dimensional compressible Navier–Stokes equations were solved to directly simulate the flow and aerodynamic sound from the fan. The sixth-order-accurate compact finite difference scheme (the fourth-order-accurate scheme at the boundary) [13] for convective and viscous terms. The time integration was done using the third-order-accurate Runge–Kutta method.

The volume penalization method [14] was used to reproduce rotating or complex-shaped objects, such as a fan and casing, utilizing rectangular grids. Before the simulation, the spline function of the blade shape was calculated with respect to its axial, radial and circumferential position. The spline function was changed according to the rotation of the blade during the simulation. The external force called the penalization term V was added to the right side of the three-dimensional compressible Navier–Stokes equations as follows:

$$(\mathbf{Q}_c)_t + (\mathbf{E} - \mathbf{E}_v)_x + (\mathbf{F} - \mathbf{F}_v)_y + (\mathbf{F} - \mathbf{F}_v)_z = V \quad (2)$$

$$\mathbf{V} = -(1/\gamma - 1)\chi \begin{pmatrix} \nabla \cdot \rho \mathbf{u} \\ 0 \\ 0 \\ 0 \\ 0 \end{pmatrix}, \quad (3)$$

$$\gamma = 0.25, \quad (4)$$

$$\chi = \begin{cases} \min(1, |\theta - \theta_b| / \Delta\theta) & \text{(solid)} \\ 0 & \text{(fluid)} \end{cases}, \quad (5)$$

where $\mathbf{Q}_c$ is the vector of the conserved variables; $\mathbf{E}$, $\mathbf{F}$, and $\mathbf{G}$ are the inviscid flux vectors; and $\mathbf{E}_v$, $\mathbf{F}_v$, and $\mathbf{G}_v$ are the viscous flux vectors. In the present simulation, the coefficient γ was determined to be 0.25, so that the sound wave is almost completely reflected (reflectivity: 99%) on the objects without computational unsteadiness [15]. The variable χ is the mask function and was set to be 0 outside the solid and 1 inside the solid, except for the region near the surface of the blade. The mask function was smoothly changed in the region near the surface to minimize the radiation of artificial pressure waves [16]. The mask function was determined according to the circumferential angle from the surface, $\theta - \theta_b$, as shown in Eq. (5), where θ_b is the angle of the nearest blade surface and $\Delta\theta$ is the angle between neighboring grid points.

LES were performed to clarify the unsteady flow and acoustic fields without excessive computational cost; no explicit subgrid-scale (SGS) model was used. A tenth-order spatial filter dissipated the turbulence energy in the grid scale, which should be transferred to the SGS eddies [17]. The filter also suppressed the numerical instabilities associated with the central differencing in the compact scheme. The coefficients of the filter were determined according to the values of those used by Gaitonde and Visbal [18].

2.3.2 Computational domain and boundary conditions

Figure 3 shows the computational domain. It is divided into three regions, a vortex region, a sound region, and a buffer region with different grid spacing. The grid spacing changes smoothly between regions.

In the flow region, the flow structures, such as the boundary layer, wake, and vortices around the rotating blades and struts, were captured by the minimum grid spacing $\Delta/C = 1/223$. As will be discussed in Sec. 4.3, the predicted power spectrum of velocity fluctuations in the wake indicates that the present computation suitably represents turbulence energy cascade. In the sound region, the acoustic propagation was captured where the maximum grid width was 48 mm. More than seven grid points were used per acoustic wavelength of $4nZ_b$ (1,008 Hz) for intense tonal sound occurring in some conditions. Also, in the buffer region, the grid width was stretched to weaken the acoustic wave near the artificial outflow boundary.

Non-reflecting boundary conditions were used on the inflow and outflow boundaries [19-21]. The inflow velocity was asymptotically given in the upstream buffer region [22] so that the flow rate approaches the target flow rate. Non-slip and adiabatic boundary conditions were adopted on the surfaces of the duct and baffle plate, while slip conditions were adopted in the upstream region of $x < -6L$ referring to the position of the inlet bell mouth in the experiments. The rotational velocity and ambient temperature were given for the grid points existing in the rotating fan.

3 Validation of Computational Methods

To investigate the effects of the grid resolution on the predicted results, the computations were also performed for the case without a slit by using coarser and finer grids with the minimum grid spacing around the fan of $\Delta/C = 1/112$ and $1/319$ compared with the present grid of $\Delta/C = 1/223$. The total number of grid points were 43 and 299 million for coarse and fine grids, respectively, while the number of the present was approximately 160 million. Figure 4 shows the pressure rise predicted by utilization of these grids for the opening ratio of $R = 0.0$ and 0.34 along the experimental results, where the static pressure coefficient ψ is shown versus the flow coefficient ϕ . By a Grid Convergence Index (GCI) method [23], the apparent order of the numerical accuracy was estimated to be 0.68 based on the predicted pressure coefficients around the design flow rate ($\phi = 0.19$) for $R = 0.0$. Also, the numerical uncertainty for the present grid regarding the quantification of grid independence was estimated to be 5% .

The predicted static pressure rise and radiated sound by the fan are compared with the measured results. As shown in Fig. 4, the predicted results show that the aerodynamic performance deteriorates at the design flow rate of $\phi = 0.19$ for the case of no slit ($R = 0.0$) compared with the case with slits. Meanwhile, the performance becomes higher at lower flow rates. These results relatively agree with the measured results. This means that the present computational methodology is appropriate to predict the effects of the flow through slits on the flow around the fan.

The predicted and measured sound pressure spectra at $2.5L$ position upstream of the fan are shown in Fig. 5. Both the measured and predicted results show that the sound pressure level at the frequency of $4nZ_b$ is higher for $R = 0.34$ compared with $R = 0.0$. Also, the sound pressure level in the low-frequency range around nZ_b increases for

$R = 0.0$. These results also agree with the measured results. Therefore, it has been demonstrated that the computational method used in this study is useful to clarify the flow field and acoustic radiation around the fan.

4 Results and Discussion

4.1 Fan performance

Figure 6 shows the measured curves of the static pressure rise for the opening ratio of $R = 0.0$, 0.17, and 0.34. In addition, computational results at four flow rates for $R = 0.0$ and 0.34 are shown. The experimental results show that the pressure rise is almost the same in the flow rate range greater than $\phi = 0.22$ irrespective of the value of the opening ratio and is reduced at and near the design flow rate of $\phi = 0.19$ for the case of no slit compared with the cases with slits. The pressure rise of $R = 0.17$ is slightly higher overall than that of $R = 0.34$ and the difference in the pressure rise increases with decreasing flow rate, especially in the flow rate range smaller than the design flow rate of $\phi = 0.19$. The pressure rise at low flow rates becomes higher for the case with the slits completely closed compared with the other cases.

4.2 Vortical structures around the blades

Figure 7 shows the predicted vortical structures represented by the isosurfaces of the second invariant of the velocity gradient tensor based on the instantaneous absolute velocity around the fan at the design flow rate ($\phi = 0.19$). The tip vortices are composed of fine-scale vortices formed near the blade tip for both conditions of $R = 0.0$ and 0.34. The structured tip vortices are convected downstream for $R = 0.34$. In the case of no slit

($R = 0.0$), the active fine-scale vortices in the tip vortices are widespread between the blades. This is due to the difference in the axial velocity. Figure 8 shows the distributions of the axial velocity on the planes of $z = 0$ and $x_c/C = 0.8$. The axial velocity around the blade tip is decreased for $R = 0.0$ compared with the case of $R = 0.34$. This is because the inflow from the outside of the casing through the upstream part of the slits is suppressed, which will be discussed in detail in Sec. 4.4.

Figure 9 shows the vorticity magnitude contours along 80% span ($r = (D_h + 0.8(D_f - D_h))/2$) in a developed view. The vortical structures between two adjacent blades are due to the tip vortices. The tip vortices are convected toward the trailing edge of the neighboring blade for $R = 0.34$. Downstream of the blades, wakes with vortical structures are formed, and the interaction of the wakes with the struts is observed. For $R = 0.0$, the tip vortices separated from the suction side of one blade are widespread, and the flow between the blades becomes turbulent. The tip vortices interact with the pressure side of the neighboring blade. Also, as shown in Fig. 9, the vortical structures are spread into the upstream region of the rotor, which causes the interaction of the inflow turbulence with the rotor.

4.3 Pressure and velocity fluctuations

Figure 10 shows the predicted spectra of the pressure fluctuations on the blade along 80% span on the suction side near the leading edge ($x_c/C = 0.2$) at the design flow rate ($\phi = 0.19$). The spectra were computed from the data during 25 blade-passing periods after computational flow development from the initial field. The frequency resolution of the spectra is $0.1nZ_b$. As shown in Fig. 10, the power of pressure fluctuations in the low

frequency of $f/nZ_b < 5$ is intensified for $R = 0.0$ compared with the case of $R = 0.34$, where the spectrum is approximately along the curve of $-7/3$ power [24]. This is consistent with the intensification of turbulence between the blades as discussed in Sec. 4.2. Also, some peaks in the power spectrum for $R = 0.34$ indicate the periodic interactions of the inflow through the slits and the blade, which will be discussed with the predicted flow field around the slits in Sec. 4.4.

Figure 11 shows the measured sound pressure spectra with the frequency resolution of $\Delta f/nZ_b = 0.008$ at the design flow rate ($\phi = 0.19$) for $R = 0.0, 0.17$, and 0.34 . The preliminary sound measurements with upstream ducts with different width have shown that the intense tonal sound at the frequency of $1.2nZ_b$ in Fig. 11 is related with the acoustic resonance in the upstream duct. As shown in the Fig. 11, a more intense peak appears at the frequency of $4nZ_b$ in the sound pressure spectrum for $R = 0.34$ compared with the results for smaller opening ratios of $R = 0.0$ and 0.14 . This is because the above-mentioned wakes generated downstream of the blades are convected downstream and interacted with the four struts while the wakes with intense vorticity are dispersed before the struts for the case of no slits as shown in Fig. 9. Also, Fig. 11 shows that the sound pressure level in the low-frequency range around nZ_b ($f/nZ_b < 1.5$) becomes intensified for smaller opening ratio. This is possibly due to the intensification of the above-mentioned interaction between the inflow turbulence and the rotating blade, as shown in Fig. 9, where the power of pressure fluctuations on the blade surface is also intensified particularly in the low-frequency range, as shown in Fig. 10. The measured overall sound pressure level in the range of 20–20,000 Hz (0.08 – $79.4nZ_b$) for $R = 0.0, 0.17$, and 0.34

was 65.7, 62.8, and 62.7 dB, respectively. This indicates that the suppression of the flow through the slits causes the intensification of noise.

Figure 12 shows the predicted and measured power spectra of velocity fluctuations 3 mm downstream of the blades (3 mm upstream of the struts) along the radial position of $r/(D_f/2) = 0.92$, where the velocity value measured by a hot-wire anemometer was evaluated as $u_h = (u^2 + w^2)^{0.5}$ in the computational results. The predicted spectra for both conditions of $R = 0.0$ and 0.34 are in good agreement with the experimental results. Also, the measured and predicted results show that the power in the low-frequency range is increased for the case of no slits compared with the case of $R = 0.34$. This indicates that the wake flow becomes more turbulent, which leads to the intensification of energy loss. As a result, aerodynamic performance deteriorates at the design flow rate, as shown in Fig. 6.

4.4 Flow around slits

Figure 13 shows the velocity vectors color-coded by the value of the fluctuation pressure with the ambient pressure subtracted for the low flow rate and design flow rate. For the casing with the slits, the inflow from the outside of the casing toward the fan appears in the upstream part of the slits for both the low and design flow rate conditions. The periodic interactions of this inflow with the blade causes some peaks in the power spectra of the pressure fluctuations on the blade as mentioned in Sec. 4.3.

As shown in Fig. 13, the high-pressure air flows outward through the downstream part of the slits for the case with slits, particularly at the low flow rate. This means that the fluid pressurized by the fan leaks out of the casing. Therefore, the aerodynamic

performance becomes higher at low flow rates around the maximum pressure by closing slits, as shown in Fig. 6.

4.5 Pressure on the blade surface

Figure 14 shows the predicted instantaneous pressure distributions and limited streamlines on the pressure side of the fan surface at the design flow rate ($\phi = 0.19$). The flow is approximately along the curve of the blade tip for $R = 0.34$. For $R = 0.0$, flow toward the tip of the blade is observed, where the surface pressure is increased because of the effects of coarsening of the spacing between the streamlines. This is because the inflow through the upstream part of the slits for the case with slits is suppressed by closing the slits, as shown in Fig. 13.

Figure 15 shows the variation of the torque and efficiency with the flow rate; the torque was calculated by the integration of the surface pressure on the fan blades. The torque is larger for $R = 0.0$ than for $R = 0.34$ at the design flow rate of $\phi = 0.19$. This is because the pressure rising on the pressure side occurs, as shown in Fig. 14. Therefore, the efficiency, which was defined as the ratio of the hydraulic power generated by the fan to the mechanical power, $Q\Delta p/2\pi nT$, was found to decrease at the design flow rate ($\phi = 0.19$) with the case of no slits.

5. Conclusions

The flow field and radiated sound were investigated in experiments and direct aeroacoustic simulations for a small axial-flow fan in a casing with slits. The effects of the opening ratio of the casing were focused on by adjusting the number of the slits

considering the possible installation conditions that the slits are partially or completely closed by other components.

The aerodynamic performance deteriorates at the design flow rate and becomes higher at low flow rates by completely closing the slits in the casing. The aerodynamic performance is slightly higher for the opening ratio of 17% than for that of 34%.

For the casing with slits, structured tip vortices are convected downstream at the design flow rate. Also, for the casing without slits, the active vortical structures due to tip vortices are widespread between the blades with the decrease in the axial velocity near the tip. As a result, the power of pressure fluctuations on the blade surface and that of velocity fluctuations in the wake are intensified particularly in the low-frequency range. As a result, the level of radiating noise is increased.

The aerodynamic performance deteriorates due to the intensification of turbulence around the fan at the design flow rate for the casing without slits. Also, the outflow of the fluid pressurized by the fan to the outside of the casing at the lower flow rates is suppressed, which leads to the enhancement of the aerodynamic performance.

The predicted limited streamlines on the pressure side of the blade for the casing without slits show that air flows toward the blade tip at the design flow rate, which causes the pressure to rise on the blade surface. As a result, the torque exerted on the fan is increased for the case without slits and, therefore, the efficiency of the fan becomes lower.

The obtained knowledge related with the flow around the casing slits is expected to contribute to the development of fans for enhancement of aerodynamic and aeroacoustic performances considering the various installation conditions.

ACKNOWLEDGMENT

This work was supported by JP17K06153 by the Ministry of Education, Culture, Sports, Science, and Technology of Japan (MEXT).

NOMENCLATURE

A	flow passage area, m^2
C	chord length, mm
D_f	fan diameter, mm
D_h	hub diameter, mm
D_s	inside diameter of casing, mm
E, F, G	inviscid flux vectors
E_v, F_v, G_v	viscous flux vectors
L	side length of casing, mm
M	Mach number
N	rotational speed, s^{-1}
P	Mean pressure, Pa
P	Pressure, Pa
Q	flow rate, m^3/s
Q_c	vector of conserved variable
R	opening ratio of casing
R	radial position, mm
Re	Reynolds number

T	torque, N m
U	axial velocity, m/s
V	volume penalization term
V_{tip}	tip velocity, m/s
W	velocity in z direction, m/s
X	axial coordinate, m
x_c	coordinate along blade chord, m
Y	vertical coordinate, m
Z	coordinate normal to x, y -axes, m
Z_b	number of blades
Z_s	number of struts

Greek Letters

Δ	difference
θ	angle, deg
θ_b	angle of blade surface, deg
N	kinematic viscosity, m^2/s
P	fluid density, kg/m^3
γ	parameter of volume penalization method
ϕ	flow coefficient

X mask function

ψ static pressure coefficient

APPENDIX A: Uncertainty of Measurement Equipment

The sources of the uncertainty of flow coefficient was the uncertainties of a differential pressure gauge for an orifice flowmeter and the evaluation of the ambient density using a barometer and a thermometer. The uncertainties of the pressure gauge, barometer, and thermometer were $\pm 0.4\%$ FS, 1% of the atmospheric pressure, and ± 1 K, respectively. These lead to the evaluation uncertainty of ± 0.005 kg/m³ for the ambient density and that of ± 0.0002 m³/s for the flow rate. The uncertainty of the rotation speed is 0.15 s⁻¹, which corresponds to the uncertainty of 0.017 m/s in the tip velocity.

APPENDIX B: Large Eddy Simulations

To derive the governing equations for large-eddy simulations (LES), any variable ψ is decomposed into its grid-scale (GS) $\bar{\psi}$ and subgrid-scale (SGS) ψ'' components,

$$\psi = \bar{\psi} + \psi'' . \quad (\text{B1})$$

In a compressible flow, the grid-scale quantity is recast in terms of a Favre-filtered variable:

$$\tilde{\psi} = \frac{\overline{\rho\psi}}{\bar{\rho}} . \quad (\text{B2})$$

The GS components of the conserved variables and the fluxes are given as

$$\mathbf{Q} = \frac{1}{J} (\bar{\rho}, \bar{\rho}\tilde{u}_1, \bar{\rho}\tilde{u}_2, \bar{\rho}\tilde{u}_3, \bar{\rho}\tilde{e})^t, \quad (\text{B3})$$

$$\mathbf{F}_k = \frac{1}{J} \begin{pmatrix} \bar{\rho}\tilde{u}_k \\ \bar{\rho}\tilde{u}_1\tilde{u}_k + \bar{p} \\ \bar{\rho}\tilde{u}_2\tilde{u}_k + \bar{p} \\ \bar{\rho}\tilde{u}_3\tilde{u}_k + \bar{p} \\ (\bar{\rho}\tilde{e} + \bar{p})\tilde{u}_k \end{pmatrix}, \quad (\text{B4})$$

$$\mathbf{F}_{vk} = \frac{1}{J} \begin{pmatrix} 0 \\ \tilde{\sigma}_{k1} \\ \tilde{\sigma}_{k2} \\ \tilde{\sigma}_{k3} \\ \tilde{u}_j\tilde{\sigma}_{kj} + \kappa \frac{\partial \tilde{T}}{\partial x_k} \end{pmatrix}, \quad (\text{B5})$$

$$\tilde{\sigma}_{kl} = \mu(\tilde{T}) \left(\frac{\partial \tilde{u}_k}{\partial x_l} + \frac{\partial \tilde{u}_l}{\partial x_k} - \frac{2}{3} \delta_{kl} \frac{\partial \tilde{u}_n}{\partial x_n} \right), \quad (\text{B6})$$

$$\kappa = \mu(\tilde{T}) C / \text{Pr}, \quad (\text{B7})$$

where σ_{kl} is the viscous stress tensor, κ is the thermal conductivity, and δ_{kl} is the Kronecker's delta function. No explicit SGS model is used here: the turbulent energy in the GS that should be transferred to SGS eddies is dissipated by the 10th-order spatial filter, as described below. It was shown by many researchers [25-27] that the above-

mentioned method combining the low-dissipation discretization schemes and the explicit filtering correctly reproduces turbulent flows such as a cavity flow and a round jet.

$$\alpha_f \hat{\psi}_{i-1} + \hat{\psi}_i + \alpha_f \hat{\psi}_{i+1} = \sum_{n=0}^5 \frac{a_n}{2} (\psi_{i+n} + \psi_{i-n}), \quad (\text{B8})$$

where ψ is a conserved quantity and $\hat{\psi}$ is the filtered quantity.

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Table Caption List

Table 1 Operating and flow conditions

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- Fig. 13 Predicted velocity vectors color-coded by the value of pressure with ambient pressure subtracted. (a) $R = 0.0$ and low flow rate ($\phi = 0.08$). (b) $R = 0.34$ and low flow rate ($\phi = 0.06$). (c) $R = 0.0$ and design flow rate ($\phi = 0.19$). (d) $R = 0.34$ and design flow rate ($\phi = 0.19$).
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Fig. 15 Predicted variation of torque and efficiency with flow rate coefficient

Table 1 Operating and flow conditions

	Exp.	Comp.
Flow coefficient ϕ	0.0 - 0.31	0.08 ($R = 0.0$), 0.06 ($R = 0.34$) (around maximum pressure) 0.19 ($R = 0.0, 0.34$, design flow rate) 0.21 ($R = 0.0, 0.34$) 0.31 ($R = 0.0$), 0.30 ($R = 0.34$) (maximum flow rate)
Opening ratio of casing	0.0, 0.17, 0.34	0.0, 0.34
Flow passage area A [m ²]	0.001	
Rotation speed n [s ⁻¹]	50.4	
Rotor tip velocity V_{tip} [m/s]	5.86	
Mach number M	0.017	
Reynolds number Re ($= V_{\text{tip}}C/\nu$)	8600	

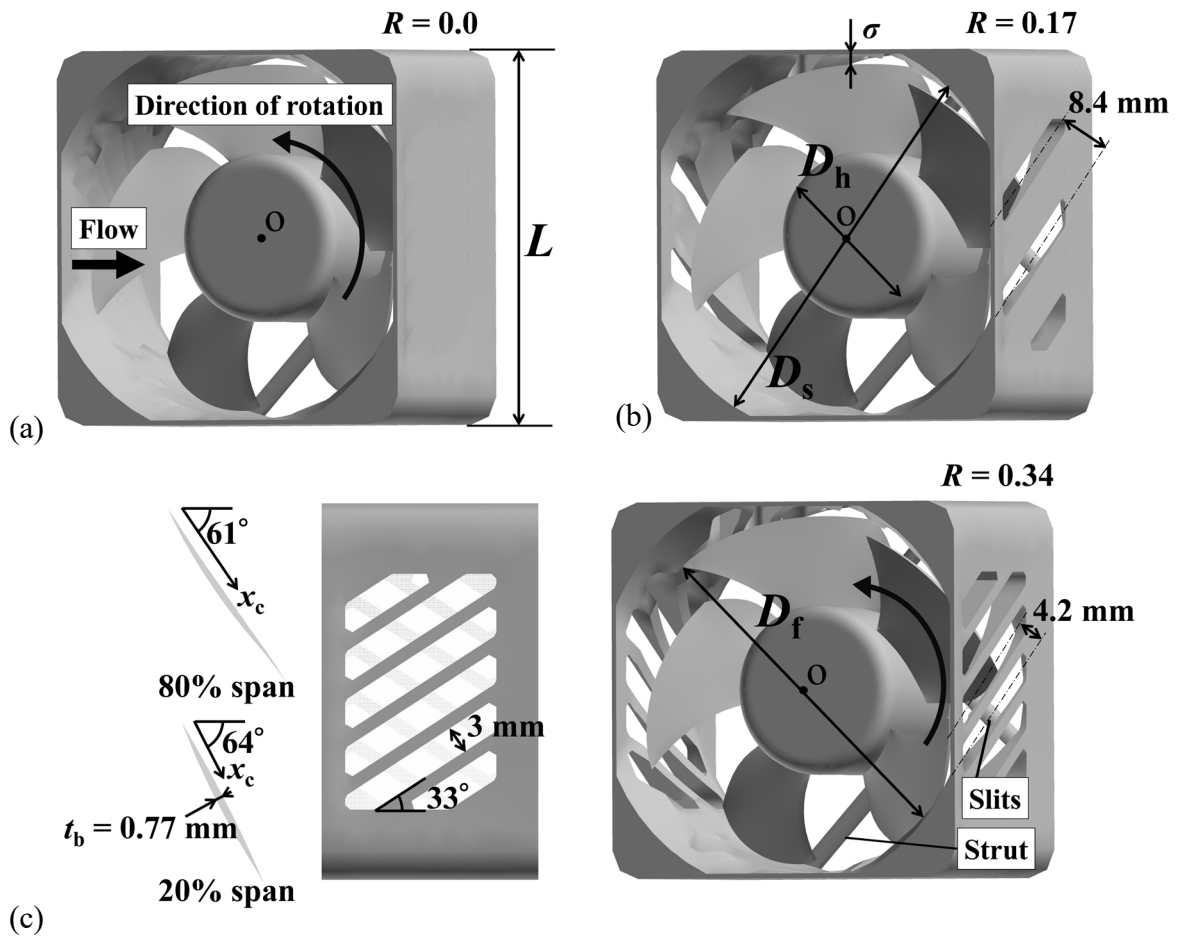


Fig. 1 Fan in a casing. (a) $R = 0.0$. (b) $R = 0.17$. (c) $R = 0.34$.

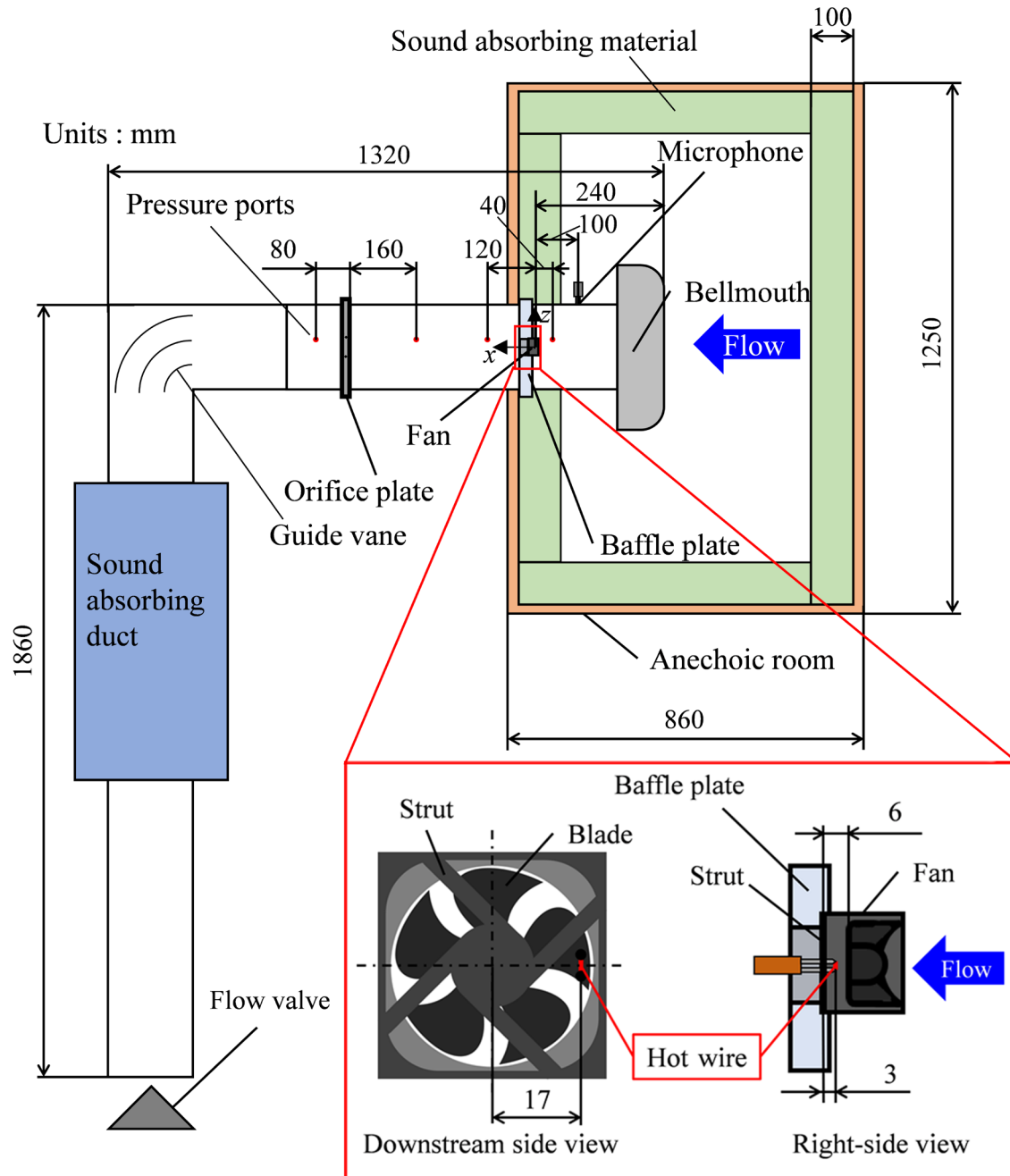


Fig. 2 Experimental setup

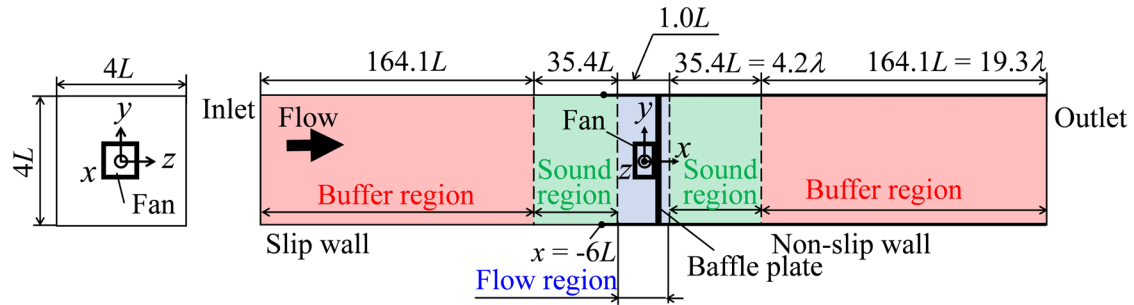


Fig. 3 Front and side views of the computational domain

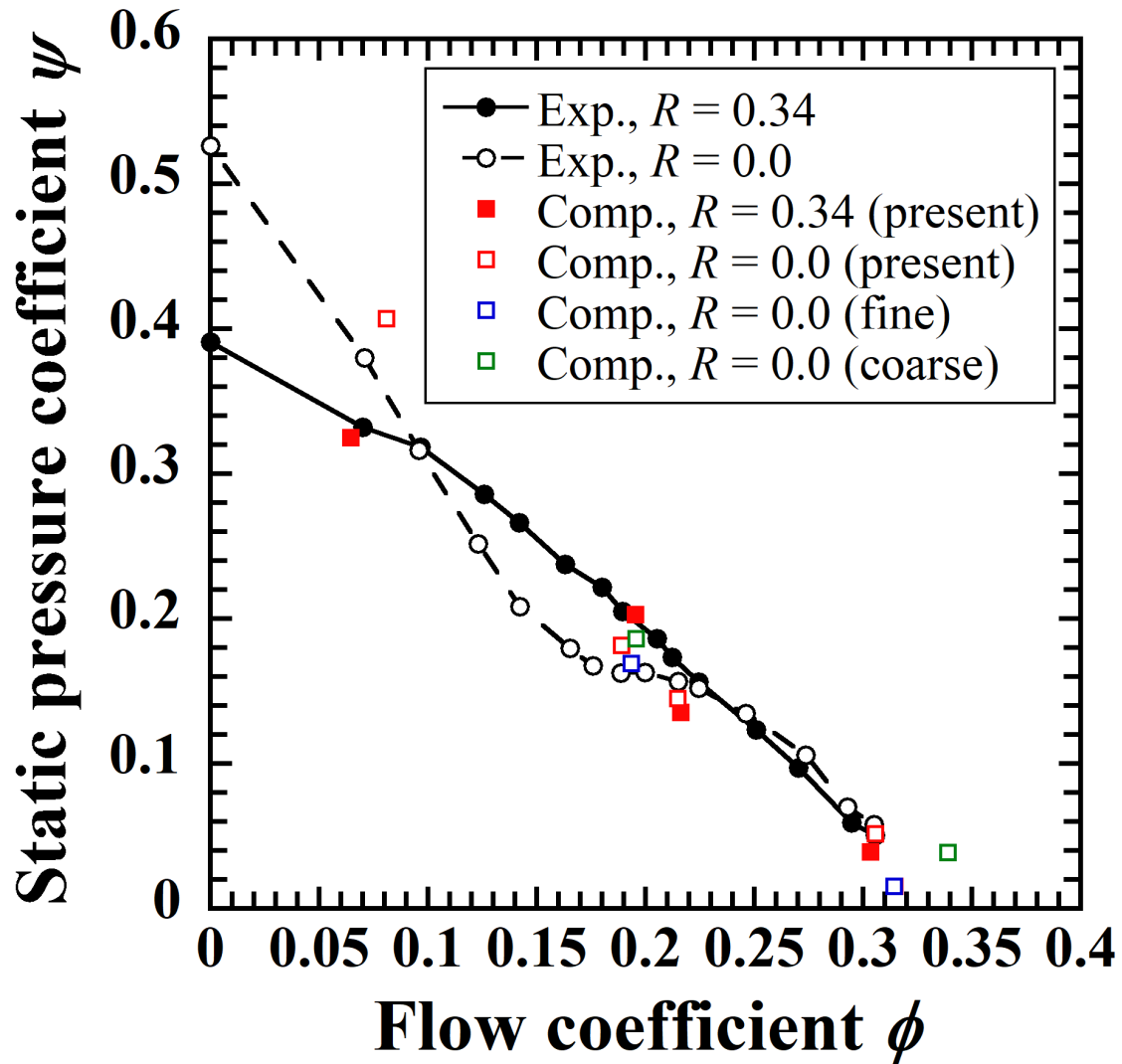


Fig. 4 Predicted variation of static pressure coefficient with the present, coarse, and fine computational grids for opening ratio of $R = 0.0$ and 0.34 with the experimental results

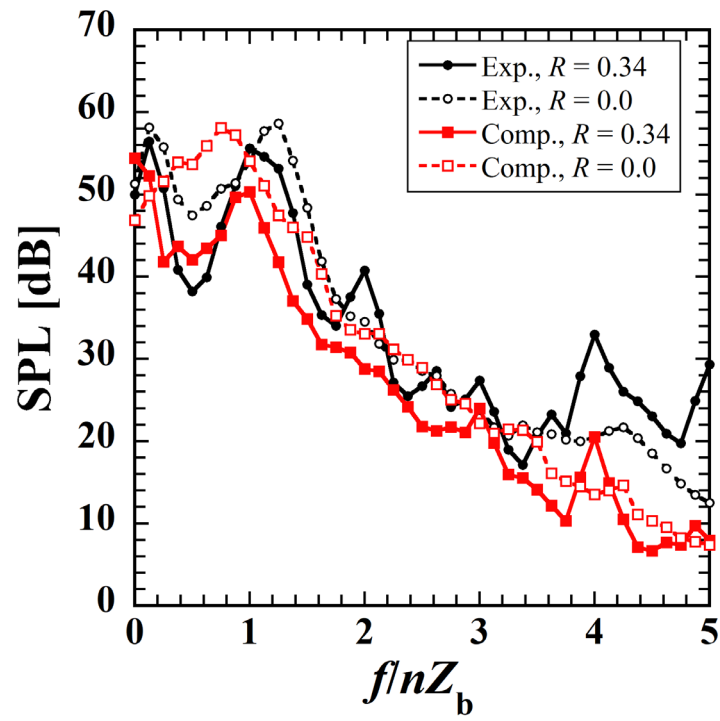


Fig. 5 Predicted and measured sound pressure spectra at position $2.5L$ upstream of fan

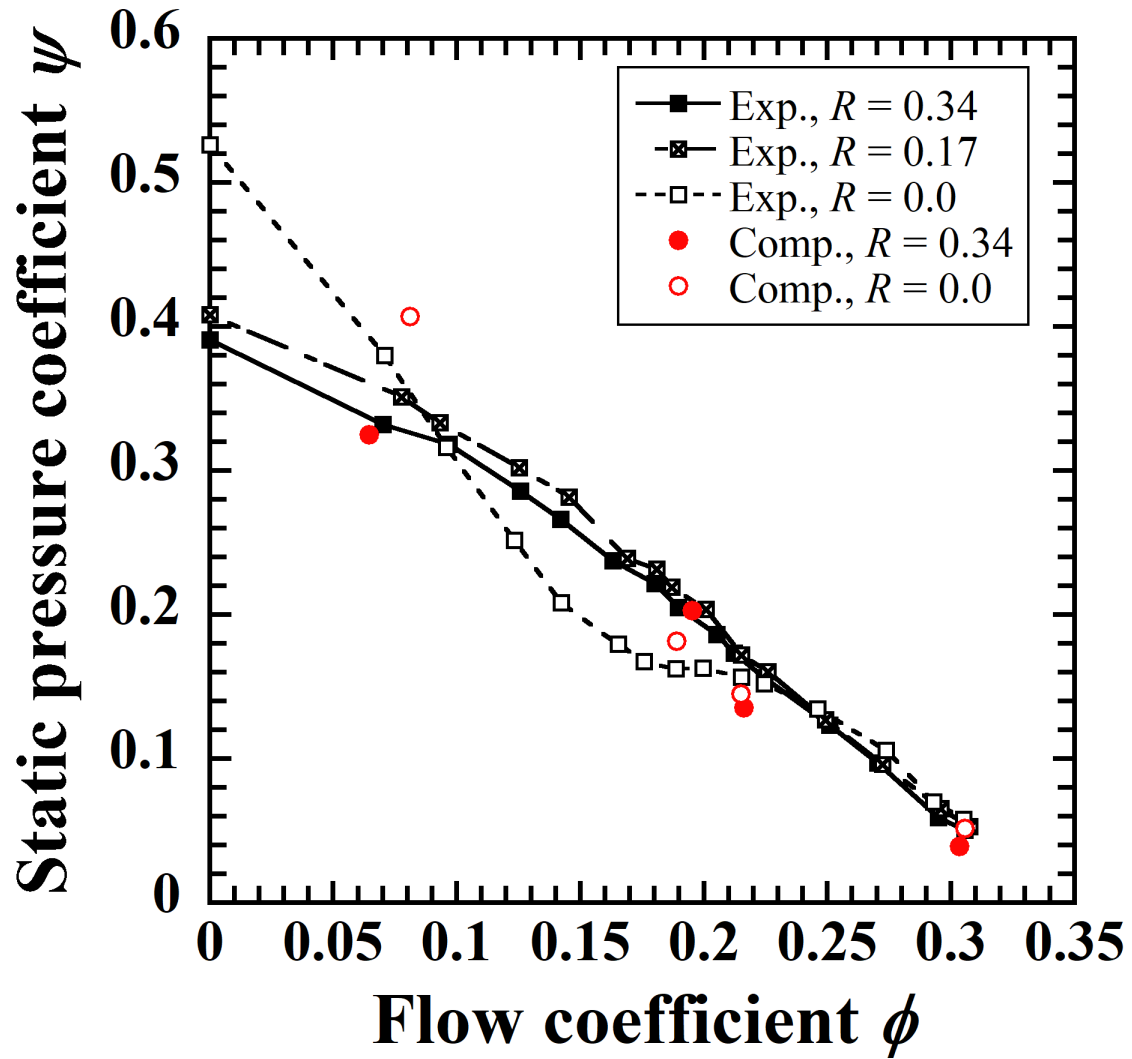


Fig. 6 Predicted and measured variation of static pressure coefficient with different opening ratio

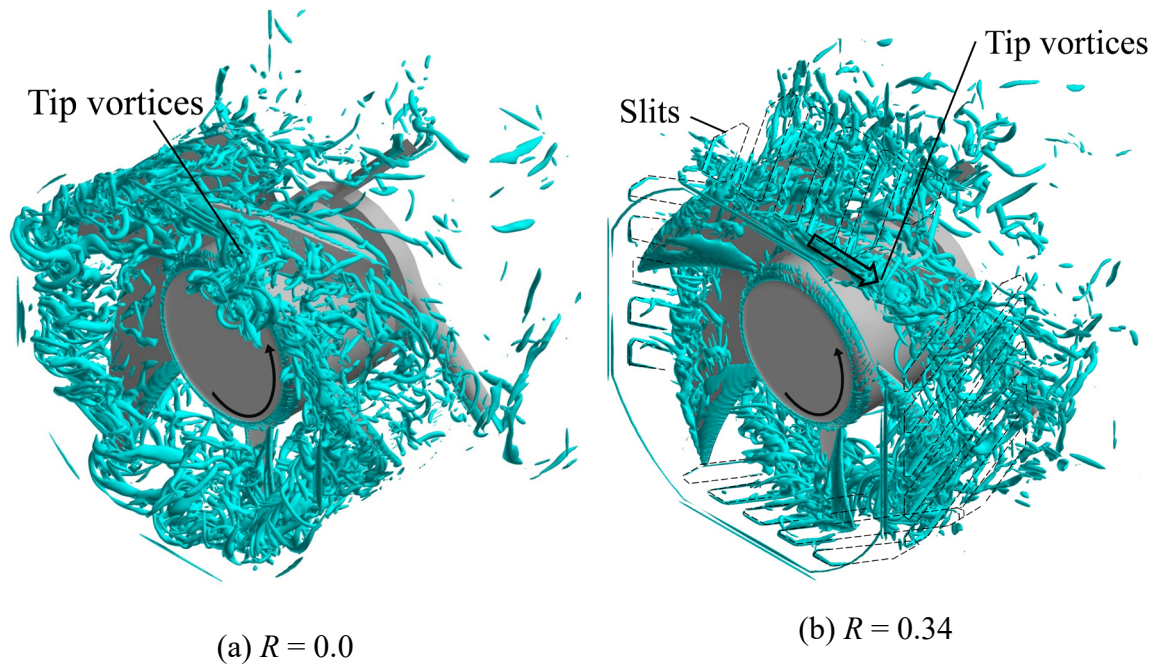


Fig. 7 Predicted iso-surfaces of the second invariant ($q/(V_{\text{tip}}/C)^2 = 50$) at design flow rate ($\phi = 0.19$). (a) $R = 0.0$. (b) $R = 0.34$.

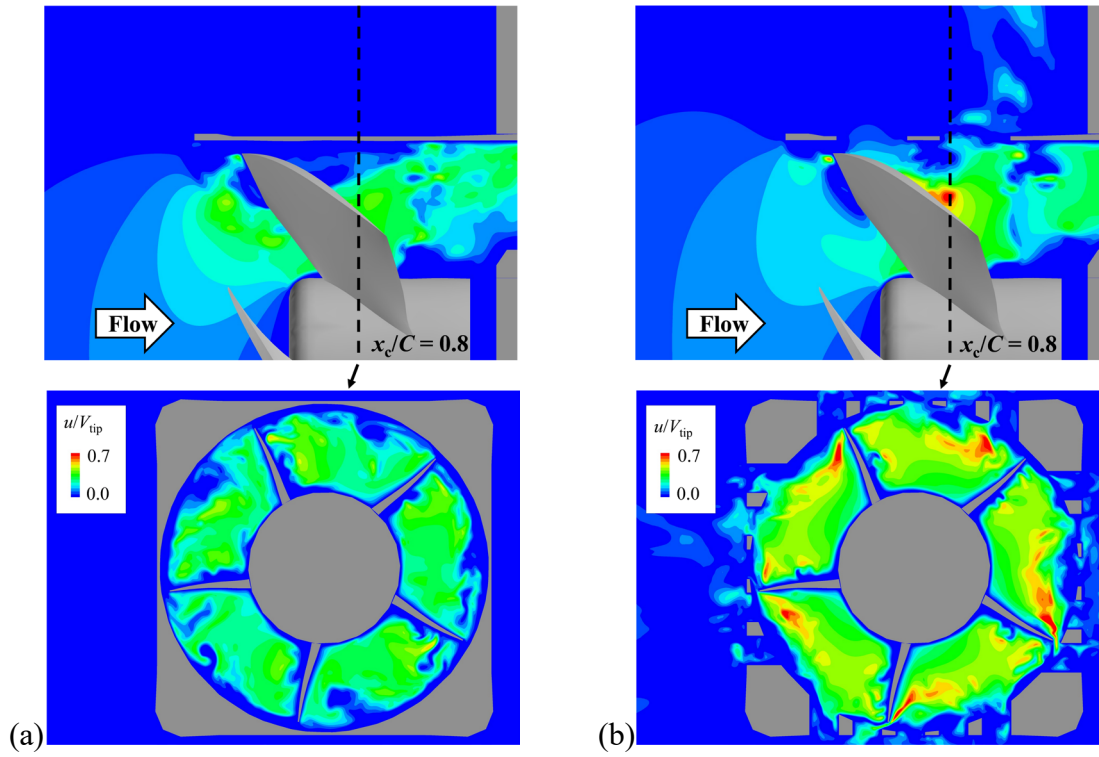


Fig. 8 Predicted contours of axial velocity on the plane of $z = 0$ (top) and those of $x_c/C = 0.8$ (bottom) at design flow rate ($\phi = 0.19$). (a) $R = 0.0$. (b) $R = 0.34$.

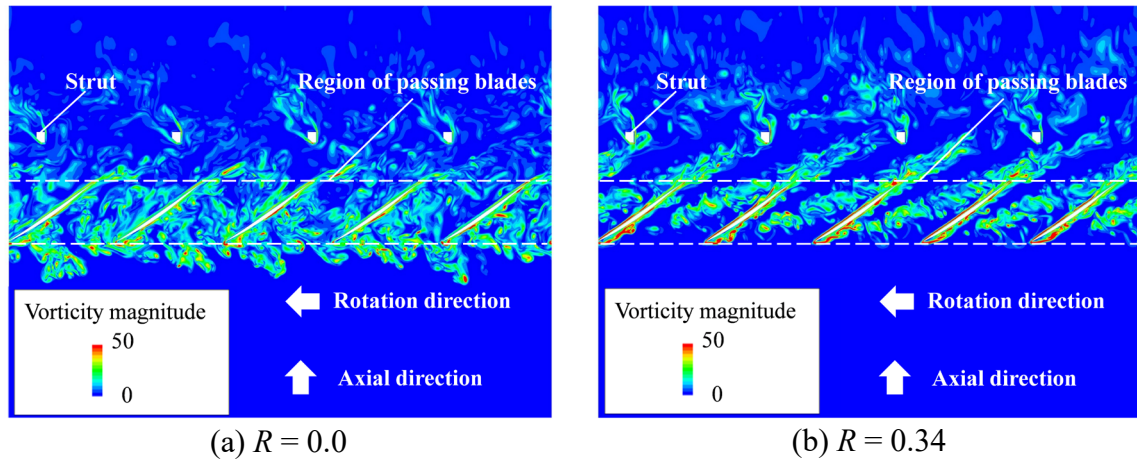
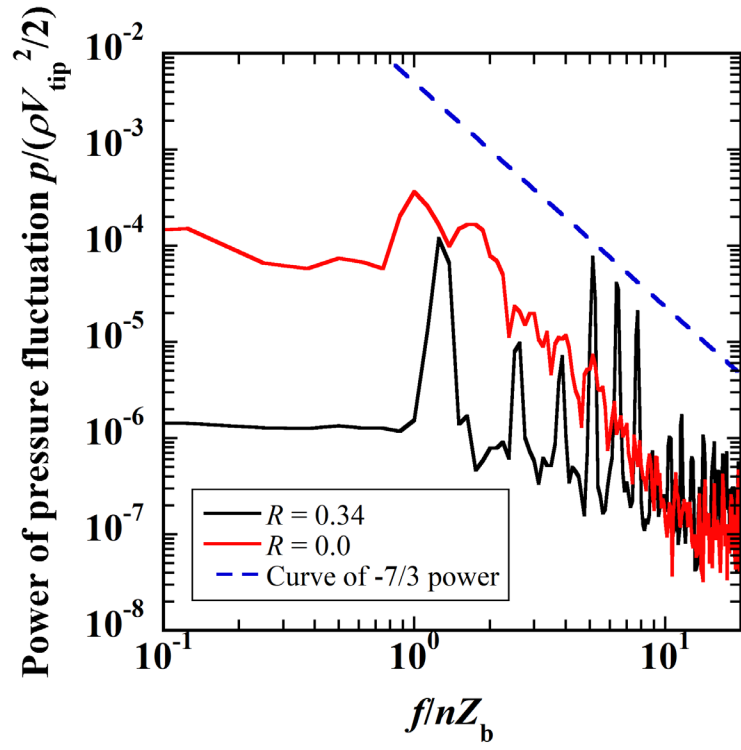


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(b)

Fig. 10 Predicted power spectra of pressure fluctuations along 80% span on pressure side at $x_c/C = 0.2$ with opening ratio of $R = 0.0$ and 0.34 ($\phi = 0.19$)

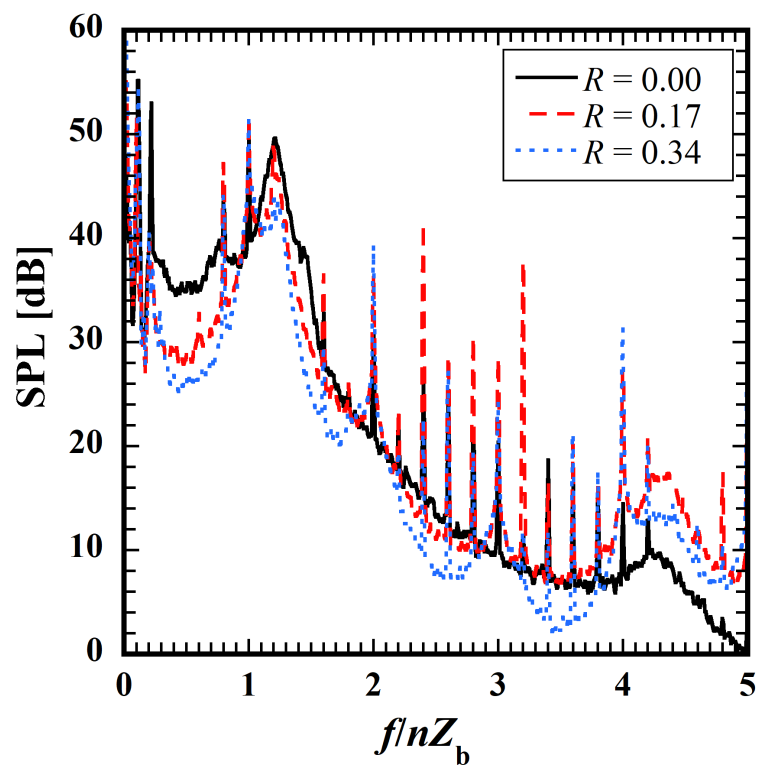


Fig. 11 Measured sound pressure spectra at position $2.5L$ upstream of fan for $R = 0.0$, 0.14 , and 0.34 ($\phi = 0.19$)

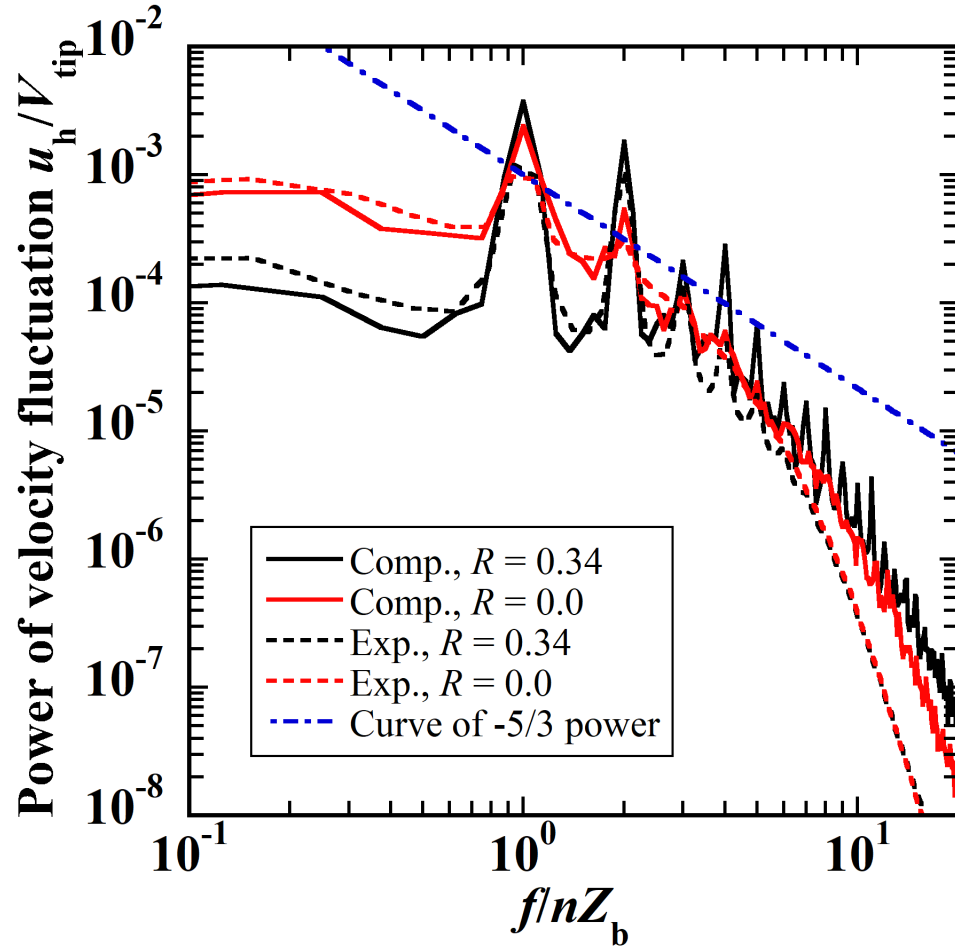


Fig. 12 Predicted and measured power spectra of velocity fluctuations at $r/(D_t/2) = 0.92$ ($z = 0$) for $R = 0.0$ and 0.34 at design flow rate ($\phi = 0.19$)

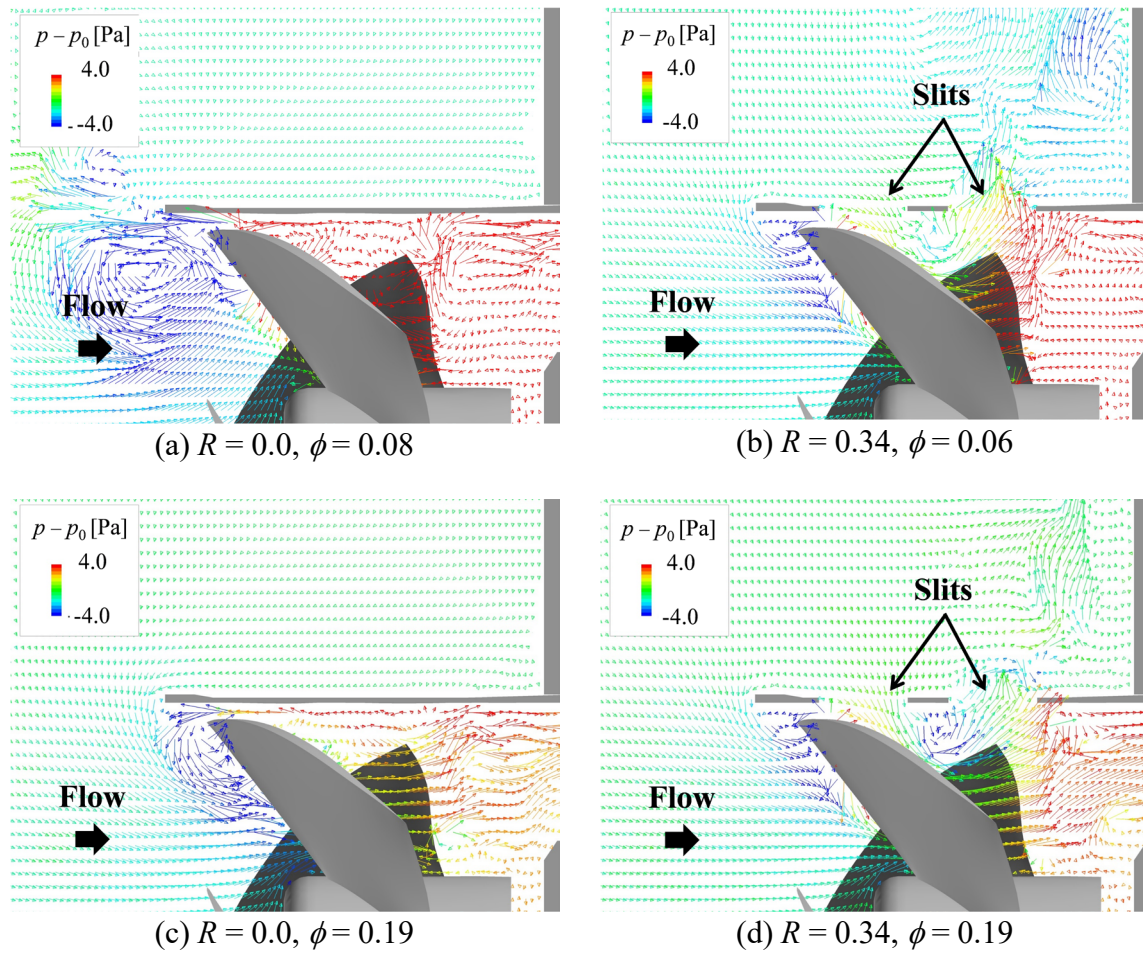


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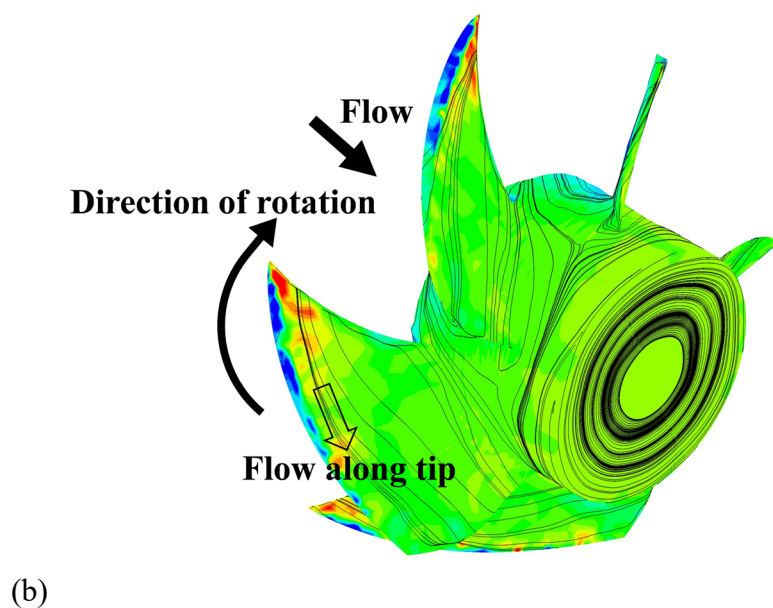
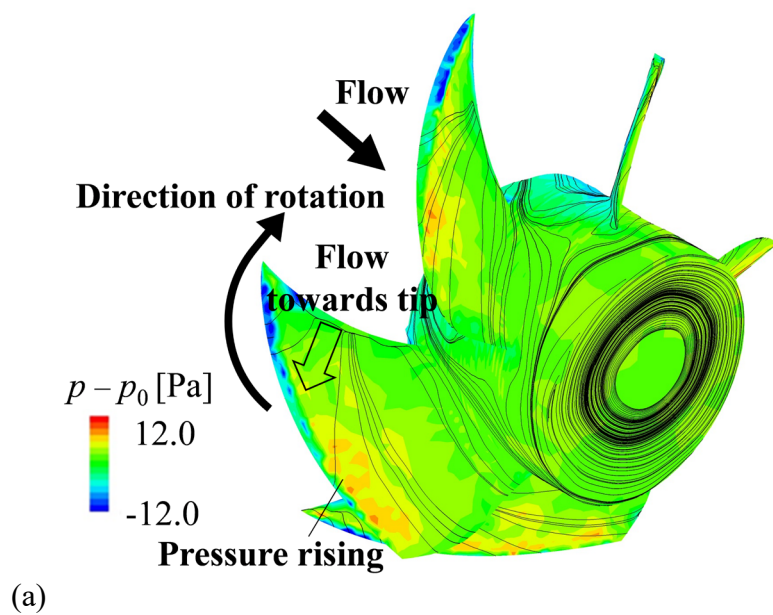


Fig. 14 Predicted pressure distribution and limited streamline on fan surface for $\phi = 0.19$.
 (a) $R = 0.0$. (b) $R = 0.34$.

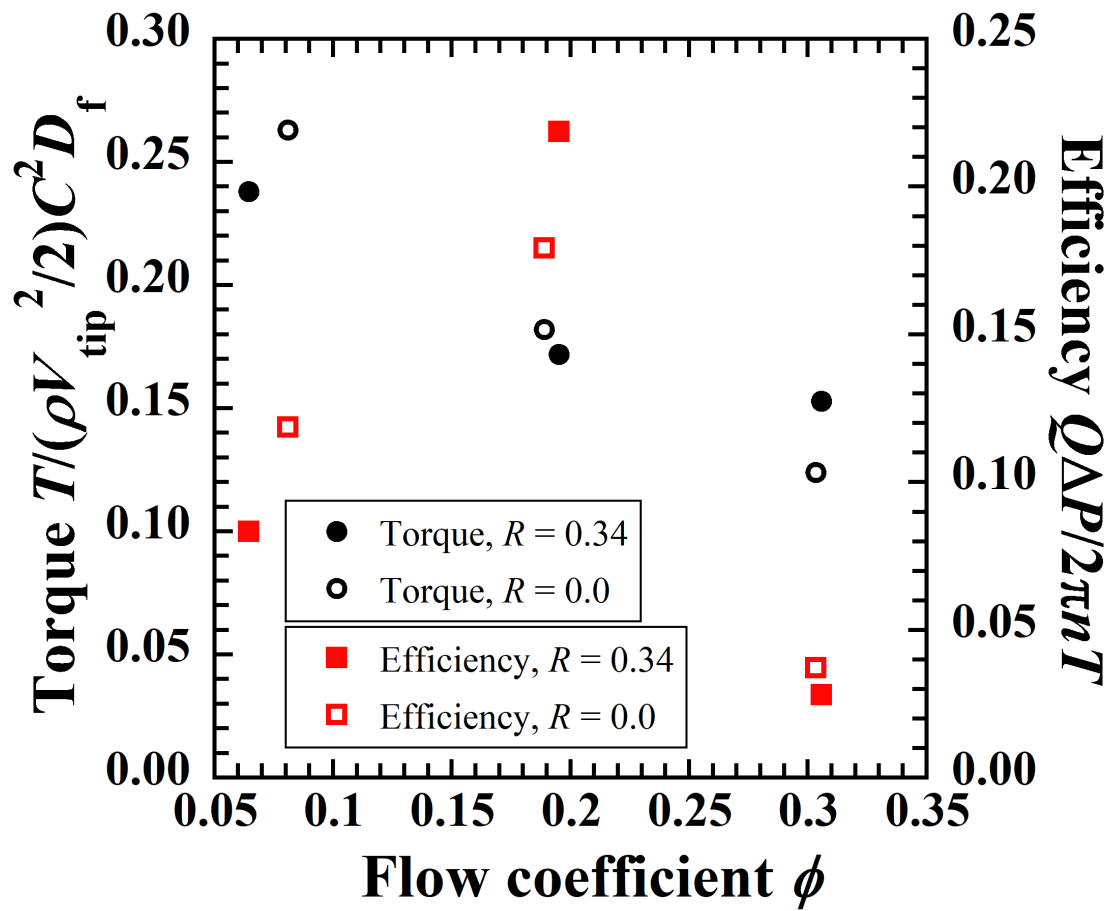


Fig. 15 Predicted variation of torque and efficiency with flow rate coefficient