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NUMERICAL AND EXPERIMENTAL INVESTIGATION OF FLOW BEHAVIOUR AND AERODYNAMIC NOISE IN AXIAL FLOW FAN OF AIR-CONDITIONER

Xifeng Zhao[1]; Jinju Sun[1] *; Zhi Zhang[2]

[1]School of Energy and Power Engineering, Xi'an Jiaotong Univ.,
28 West Xianning Road, Xi'an, China, 710049

[2] The Research Center, Midea Air-conditioning and Refrigeration Group Co. Ltd.,
Fo'shan, China, 528311

ABSTRACT

To improve aerodynamic performance and reduce noise for a split air conditioner outdoor unit fan, a hybrid method is developed, which combines the Computational Fluid Dynamics (CFD) flow simulation with Computational Aero Acoustics (CAA) noise analysis, where Large Eddy Simulation (LES) model and Ffowcs Williams-Hawkings (FW-H) acoustic analogy model are solved respectively for the unsteady flow characteristics and far field noise solutions. Experimental tests are conducted respectively for fan aerodynamic performance and acoustic behavior, with the aerodynamic performance test rig and semi-anechoic room.

Numerical results demonstrates that the main dipole sound sources are located mainly on the blade trailing edge and tip surface, and shroud and casing inner surface, such a distribution is caused by fluctuations in instantaneous pressure associated with rotor-stator interaction. The casing dipole sources contribute largely to the total noise of the fan, and are the main causes for fan noise. It is demonstrated both numerically and experimentally that modifications of impeller blade geometry are effective to reduce the fan noise. Two redesign schemes, the concaved-trailing edge and flanging outer-edge blades, are used, but the latter is more effective in reducing the fan noise as well as shaft power simultaneously. The predicted SPL agrees well with the measured results at the fundamental frequency of the highest intensity, and the hybrid method used in the present study is justified.

1 INTRODUCTION

Noise nuisance and overall performance level of the split type air conditioner outdoor unit are major concerns of air conditioner consumers while they choose the products. This largely necessitates the development of high performance low-noise axial flow cooling fan, which is one of the main noise sources in the air conditioner outdoor units. Much effort has been made by previous investigators on the noise reduction and performance improvement of the fan and some representative work is reviewed below.

Blade and shroud geometry modification are found effective to reduce improve fan performance and reduce flow-induced noise through both experimental and numerical study. Akaike and Kikuyama [1] have reported studies of the flow around the fan using 2D numerical simulations and Laser Doppler Anemometer (LDA) measurement. It is demonstrated that the flow characteristics near the blade tips can be improved and noise level reduced by changing the configuration of the shroud, and it is also clear that the vortical flow field near the rotor tip in the fan affects their aerodynamic performance and noise characteristics. Jang et al. [2,3] investigated the 3D vortical flow field in a propeller fan with a shroud in the air-conditioner outdoor unit using Laser Doppler Velocimetry (LDV) and Large Eddy Simulation (LES) to evaluate unsteady behaviour of vortex structures. In the work reported by Fukano and Jang [4], it is demonstrated that the noise increase at low rate associated with tip clearance flow is caused by high velocity fluctuation of the vortical flow and the interference between tip leakage vortex and adjacent pressure surface of the axial flow fan. Hu and Ding [5, 6] has investigated the influence of the air outlet louver and deflecting

* Contact author: Dr Jinju SUN, jjsun@mail.xjtu.edu.cn

ring on the noise in the outdoor unit of a split-unit air conditioner. It is found that a use of air outlet louver with different diameters of steel wires and deflecting ring with double contoured duct can improve aerodynamic performance and reduce noise for the outdoor unit. In all the above mentioned work, CFD simulations are used to predict the flow behavior and aerodynamic performance, while experimental approaches are used to study the causes and reduction of noise, but no attempt has been made in numerical analysis and prediction of noise.

With the advances in Computational Aero Acoustics (CAA), numerical approaches for aerodynamic noise prediction in turbomachines have produced promising results. Maaloum et al. [7] predict the tonal noise radiated by the fan in far field using the Ffowcs Williams and Hawkings (FW-H) equation according to the fluctuating forces on the blade surface. In Moroianu's work [8], the influence of the angle between the fan and ground on the total sound spectrum, and the noise generation and propagation are investigated by incorporating LES (Large Eddy Simulation) for the flow field and acoustic analogy method for the sound prediction. It is found that the dominant values of the resource terms are at the tip of the blades, which decrease from shroud to hub. Liu et al. [9] studied numerically the aerodynamic noise of a centrifugal fan using the generalized Lighthill's theory, and have found that for subsonic centrifugal fans, aerodynamic dipole sources, especially those near the volute tongue, contribute mainly to the aeroacoustic noise. Some investigations have focused on the air conditioner outdoor unit. Jiang et al. [10] has studied the noise of axial flow fan in outdoor unit of split-type air conditioner and predicted the overall sound pressure level of broadband noise by using the Fukano's model. It is demonstrated that the distance to the blade trailing edge is an important parameter for Fukano's model and derivation of noise measurement and prediction is within 3.4 dB, i.e. less than 5.5%. Tian [11] has conducted the work on the aerodynamic and aeroacoustic performance of outdoor unit with two different types of grille. It is showed that the inlet turbulence noise is generated from the interaction between the blade wake and grille, and it plays less important role in the outdoor unit configuration. Quite recently，Rumpfkeil and Zingg [12] have developed the novel hybrid NS (Navier-Stokes) and FW-H optimization method and it is used successfully to a blunt trailing edge airfoil example, which demonstrates that it is possible to minimize the pressure fluctuations at a given far-field observer position and the hybrid NS/FW–H optimization algorithm is an efficient and practical design tool for reducing aerodynamically generated noise. Liu et al. and Orselli et al. respectively reported the work on prediction of the flow-induced noise by incorporating the CFD and acoustic analogy theory [13, 14].

The Lighthill's acoustic analogy theory provides a platform for investigating the aerodynamic noise generation and propagation mechanism. The Ffowcs-Williams and Hawkings (FW-H) method [15] extends the analogy theory to suit the cases where the stationary solid boundaries and rotating blade surfaces are the sound sources, and it has been the most complete formulation of the acoustic analogy to date. Nowadays, the aerodynamic noise issues for engineering applications can be well tackled by a hybrid method of CFD and FW-H method, and the CFD simulation must capture the flow physics, especially evolution of eddies associated closely with noise generation. Clearly DNS (Direct Numerical Simulation) is the best suited approach for this purpose solely, but at present it is not very practical to be used due to the requirement of powerful computational hardware and cost. LES (Large Eddy Simulation) uses the subgrid-scale model (SGS) and thus large-scale eddies can be treated well. It is generally acknowledged that large scale eddies are largely responsible for noise generation. LES is a suitable alternative to flow modeling in the combined simulation with a moderate requirement on computation resources and cost, and it is used in the present study for flow simulation in the outdoor unit fan.

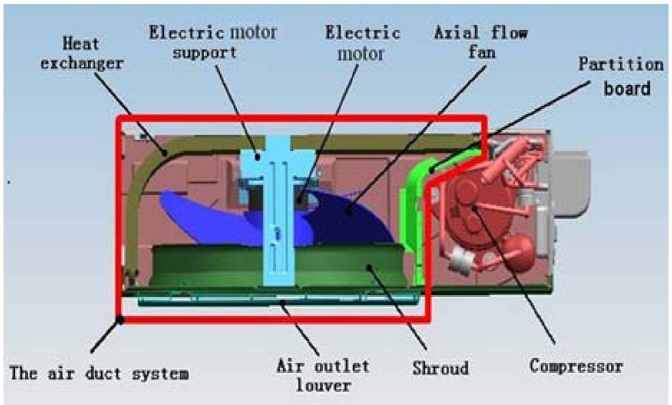
This present study focuses on the aerodynamic and aeroacoustic performance of the axial flow fan, which is used in the air-conditioner outdoor unit. A hybrid computational method of the LES model and FW-H acoustic analogy model are used and the unsteady flow characteristics and flow-induced noise are investigated. Experimental test of fan performance and noise is conducted respectively, and the hybrid method is validated based on the comparisons of the obtained numerical and experimental results.

2 EXPERIMENTAL RIG AND TEST

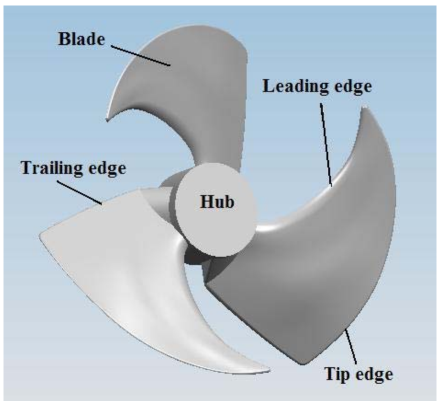
2.1 Layout and specifications

A household split-type air conditioner of 1.8 Kw is used in the present study. Overall dimension of the outdoor unit is 780 ×240×530 mm, as shown in figure 1(a), the test rig mainly includes a compressor, heat exchanger, electric motor support, electric motor, axial flow fan, partition board, shroud and air outlet louver. The axial flow fan has three forward swept blades, figure 1(b). The diameter of fan rotor is 401mm and the axial dimension is 119 mm. The blade hub to tip ratio is 0.27 and it is covered partially by the shroud over 21% percentage of the axial chord measured from the blade trailing edge with a tip clearance of 10mm (i.e. 2.4 % of the rotor diameter). The design speed and flow of fan is respectively 860 rpm and 1860 m^3/h.

The noise of air system of air conditioner outdoor unit essentially consists of the aerodynamic noise and vibration noise, and latter being mainly caused by the compressor. In present study, focus is placed on the aeroacoustic noise associated with the fan, thus the compressor is shut down during the experimental test to eliminate the main vibration cause.



(a) Outdoor unit of air conditioner



(b) Axial flow fan

Figure 1 Layout of test rig and fan

2.2 Fan Performance Test

To evaluate performance of the original fan and redesigned ones for noise reduction, experimental test is performed for the fans, and the obtained experimental data are also used to validate the numerical approaches.

As shown in figure 2, the test rig mainly consists of the pressure orifices for measuring static pressure and flow rate, thermometer for measuring the fan outlet temperature, multiple nozzles for varying the flow rate, grids used to comb the air flow, and auxiliary fan for air circulation. To achieve a real environment test, the outdoor unit and fan is fitted to the rig as a whole.

In pressure measurement, the fan is operated at design speed, and the outlet static pressure and temperature of fan are measured at different flow rates. Flow rate of the tested fan is changed through adjusting the nozzles and varying rotating speed of the auxiliary fan. Experimental data is collected by the data acquisition system and processed in the terminal equipments.

In addition, setup of the test complies with Chinese National Standard GB/T 7725-2004, GB1236-2000 and GB 2634-1993. The measurement error is within a range of ±1%.

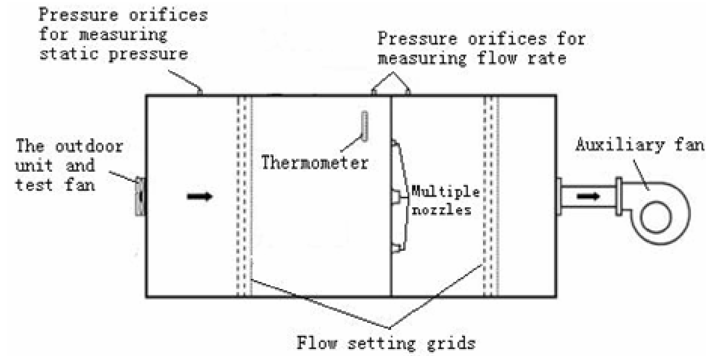


Figure 2 Fan performance test rig

2.3 Aeroacoustic Test

Noise measurement is conducted in a semi-anechoic room, figure 3, its dimension is 5.6×4.8×3.6 m and background noise is 18 dBA at operation condition (including mainly the motor noise and outdoor unit casing vibration noise). The outdoor unit is lifted to keep the centerline at a level of 1m high. The prepolarized free-field microphone is placed horizontally 1m away from the outdoor unit casing and 1.035m from the fan. The distance between the microphone location and fan is about 2.58 times of the fan diameter, which is much larger than the suggested twofold by the Chinese air conditioner noise test standard.

A wind shield (i.e. spherical sponge cover) is used to cover the microphone and reduce the direct flow effect. The fan noise is tested in the semi-anechoic room. The outdoor unit is placed on the iron platform, which is covered with a 5 mm thick rubber blanket on the top, which is expected to reduce the influence of the ground to a minimum.

For the noise data processing, the B&K Four-channel Plus 3560 type analyzer with the prepolarized free-field microphone and analog artificial sound quality system are used.

The overall A-weighted sound power level, sound pressure level (SPL) and octave band spectrum are measured. The above aeroacoustic measurement error is within ±0.2 dBA. The aeroacoustic test rig is constructed complies to the Chinese standard of GB/T 7725-2004.

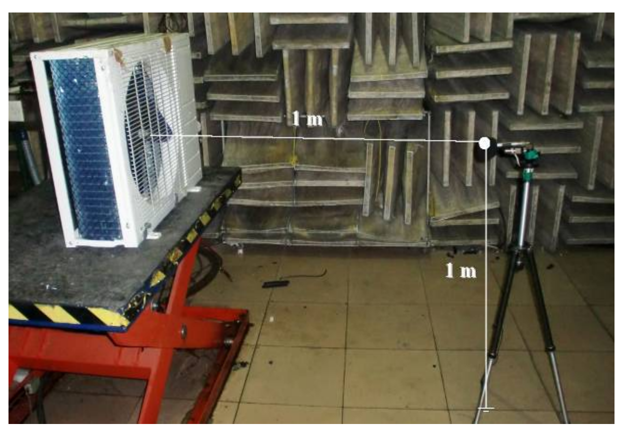


Figure 3 Aeroacoustic measurement

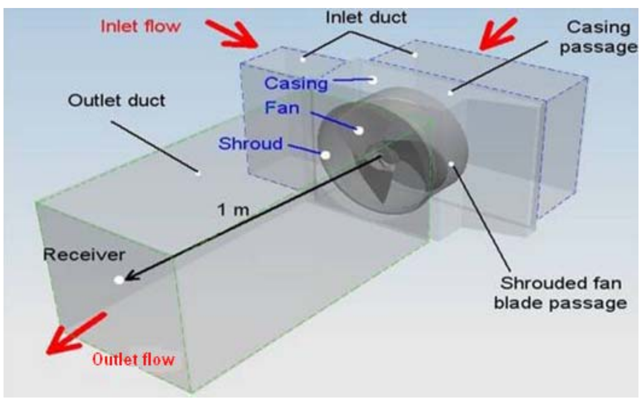
3 NUMERICAL METHOD

In the present study, the LES and FW-H acoustic analogy are incorporated to simulate the unsteady flows and flow-induced noise in the outdoor unit of the air conditioner and numerical simulations are conducted with Fluent 6.3.

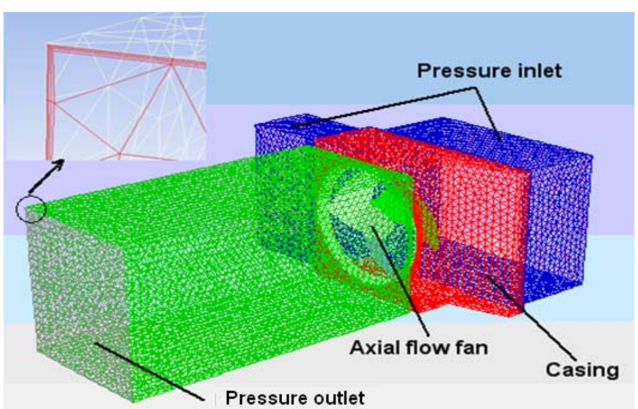
3.1 CFD Method

Physical model. The flow domain for CFD simulation consists of the inlet duct, shrouded fan blade passage, casing passage, and outlet duct. It is noted that for simulation purpose, the lengths of inlet and outlet duct are extended, as shown figure 4(a).

Grid. Gambit software is used to generate the grid for the physical model, figure 4(b). The unstructured grids are used for the flow domain. To capture the unsteady flow physics caused by the rotor-stator interaction, grid refinement at near wall regions is conducted in the conjunct region of impeller and shroud, as shown by the zoomed view in figure 4(b). Total grid size used in the simulation is 1,236,458, which is justified by the grid-dependency study.



(a) Physical Model



(b) Mesh

Figure 4 Physical model and mesh

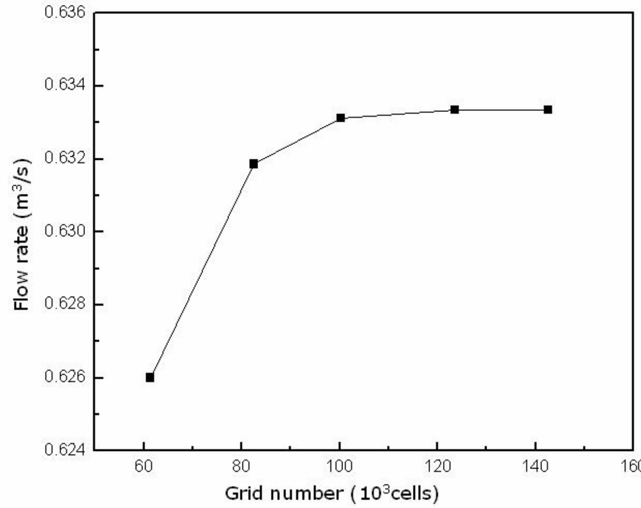


Figure 5 Grid sensitivity test result

To verify whether the solution is grid independent or not, large grids are used in flow simulation based on the LES computation, as shown in figure 5, a grid larger than 1,236,458 does not produce visible difference in the converged flow rate.

Solver. LES method is used to predict the completely unsteady flow solving the whole computational domain, the unsteady Navier-Stokes equations are solved with Smagorinsky-Lilly subgrid-scale model, where the Samgorinsky takes a value of 0.1 for the present simulation. The time-dependent term is discretized by the second-order implicit scheme and second order bounded central-differencing scheme is used for convection and diffusion terms. The pressure-velocity coupling is calculated through the PISO algorithm. To accelerate the convergence, the steady flow simulation is carried out first and then the obtained converged results are used as the initial field to start the unsteady flow simulation by means of LES approach.

Time step for the unsteady calculation must sustain the time resolution for noise prediction and analysis. It is closely related to the fan rotating speed, n , and given by

$$\Delta t = 60/360n \approx 1.94 \times 10^{-4} \text{ (s)} \qquad (1)$$

It is chosen in such a way that one complete revolution can be performed after each 360 time steps. Number of iteration is adjusted to reduce the residuals below an acceptable value in each time step. In particular, the ratio between the sum of residuals and sum of the fluxes for a given variable in all the cells is reduced to the value of 10^{-5} for computational convergence.

Boundary condition. Total pressure and total temperature are imposed at the inlet and at the outlet the static pressure is used. At the solid surface, no-slip conditions are used. At the rotor-stator interfaces of fan rotor and casing, and rotor and outlet duct, the moving mesh approaches are applied to consider the influence of rotor-stator interaction.

3.2 Aeroacoustic Noise Prediction Method

In noise prediction, it is much essential to identify the source responsible for noise generation. At present, the FW-H method is the most commonly used acoustic analogy based approach. A sound review on fan noise is given in [16], and a useful conclusion has arisen that the dipole sources are main noise sources on a rigid surface for the small and medium-sized fans, being distributed on the surfaces of moving blade, stator and casing. Previous work reported provides a good reference for noise prediction.

The present study focuses on the air-conditioner outdoor unit noise generation mechanism and reduction. In the fan of outdoor unit, the dipole sources residing on the fan blade, shroud and casing surface are main noise sources, being caused by the fluctuating forces acting on these solid walls. The approach of splitting and solving the flow and sound fields from can be simplified further if the receiver has a straight, unobstructed view of each individual point, that is a source of noise. Sound propagation from a point source to a receiver, figure 4(a), can be computed by a simple analytical formulation. The reduced FW-H equation with a moving dipole source assumption is considered for calculating sound pressure on the observer point (in the far field) [17] and written as

$$p'_d(\mathbf{x},t)=\frac{1}{4\pi c_0}\int_{f=0}\left[\frac{r_i}{r^2(1-M_r)^2}\times\left\{\frac{\partial F_i}{\partial t}+\frac{F_i}{(1-M_r)}\frac{\partial M_r}{\partial t}\right\}\right]dS \tag{2}$$

Where $\mathbf{x}$ is the spatial vector from source point to observer point, c_0 is sound speed, r distance between observer point and source point($r=|x_i-y_i|$), r_i distance between observer point and source point in the *i* direction ($r_i=x_i-y_i$, x_i and y_i are respectively observer point coordinate and source point coordinate), M_r is mach number in the radiation direction, F_i is the source point impulse force to fluid reaction in the *i* direction, and the quantity in the brace is calculated at the retarded time $\tau=t-r/c_0$. The sound pressure of the moving point force is calculated using the time variation of force and acceleration. By applying the equation to blades, shroud and casing, the sound pressure in the far field can be calculated. It is noted that only the noise source behavior and its radiation to the free field are included and the reflection effect of casing and blade surfaces are not involved.

To measure quantitatively sound sources, the root mean square value of the time-derivative of static pressure is to used to describe the dipole source intensity, written as

$$I_{ds}=\sqrt{\frac{1}{N}\sum_{i=1}^{N}\left(\left.\frac{\partial p_s}{\partial t}\right|_i\right)^2} \tag{3}$$

Where *i* is the total number of time steps, $\partial p_s/\partial t$ is the time history of static pressure fluctuation at any grid nodal point on the wall surfaces. With the dipole source intensity, the location of main dipole source can be readily determined; clearly the one with large intensity is identified as the main dipole source.

Sound pressure level (SPL) is used in the present study and defined as

$$SPL=10\log_{10}(p/p_{ref})^2 \tag{4}$$

Where p_{ref} is the standard reference pressure and takes a value of $2\times10^{-5}\,Pa$. The SPL spectrum is obtained by the Fast Fourier Transform (FFT) algorithm.

The sound pressure is calculated with an assumption that the free field is 1 m away from the outlet of the outdoor unit casing, figure 4. This is consistent with the experimental setup of aeroacoustic measurement, as shown in figure 3. Flow boundary condition and non-reflecting boundary condition on the inner wall are used as the acoustic boundary conditions.

The blade passage frequency (BPF) is a suitable reference for noise analysis, and given by

$$f=nzI/60 \tag{5}$$

where z represents the number of blade (being 3 in the present study) and I the sequence number of harmonious wave, taking an integer value of 1, 2, 3…… .

To analyze the obtained numerical and experimental acoustic results, the transient data are processed by a FFT algorithm.

4 NUMERIAL AND EXPERIMENTAL RESULTS

Three different fans (the original and two redesigned fans) are studied both numerically and experimentally, and some results are presented with comparison and analysis.

4.1 Original Fan

Performance and flow behavior

LES approach as described in section 3 is used to predict the flow and performance for the original fan at design speed of 860 rpm, and some obtained results are presented below.

Figure 6 presents the predicted fan performance together with the measured data. The predicted performance curve has the same tendency as the experimental one. At the near design flow rates, discrepancy between the numerical and experimental is much small. At large flow rates, the difference is evident in both the outlet static pressure and flow range. This needs further refinement in numerical modeling, for example, influence of the outlet louver, electric motor support, and heat exchanger on the flow has been considered in the present study, which can be the main causes for deviation of the predicted performance from the experimental. But such work is left for future consideration.

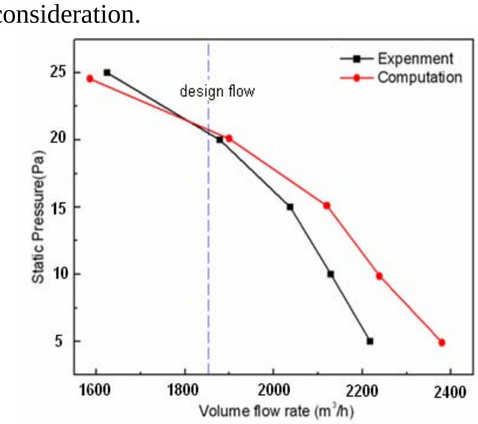


Figure 6 Original fan outlet static pressure versus flow rate

Figure 7 presents the predicted instantaneous static pressure contours on the fan blade and casing surface at three different instants. Clearly the distribution of static pressure of both rotor and stationary casing varies with the movement of the blades, which illustrates that the unsteady pressure characteristics are largely due to the rotor-stator interaction.

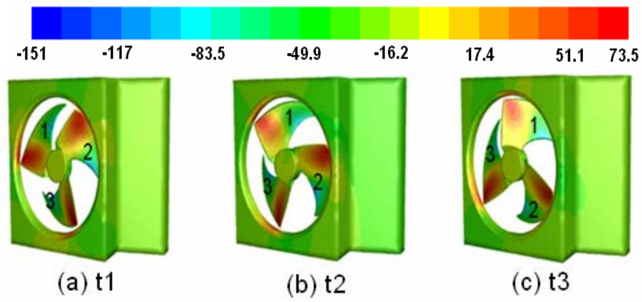


Figure 7 Static pressure distribution at different time instant

Noise

Noise at a given far-field position is predicted using the acoustic analogy method described in section 3.2, which 1 m away from the outlet of the outdoor unit casing, figure 4. The noise measurement is also conducted at this position and such a consideration is convenient for comparison between the experimental and numerical results. Some measures are taken by the authors to avoid the near-field noise measurement. The test position (where the microphone was placed) is 2.58 times of fan diameter distant from the air conditioner fan; the microphone is covered with a wind shield to reduce direct flow effect of the fan.

Variations of the surface dipoles are of great importance for noise generation. Spatial distribution of the dipole source intensity is predicted and shown in figure 8. It is visible that the main dipole sources are the located at the blade trailing edge and tip, shroud inner surface and casing inner surface.

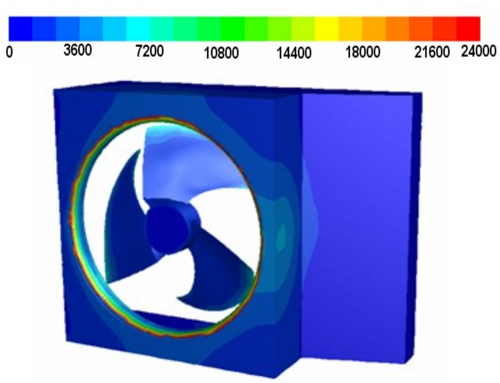


Figure 8 Distribution of the dipole sound source intensity

Figure 9 presents the predicted transient sound pressure produced by individual sound sources at the given far-field position, and clearly all the curves are periodic. The sound pressure fluctuation generated by the sources on the stationary casing is much larger than that on the rotating blades. This is resulted from the strong interactions of the rotor and casing, which is also the main cause of the discrete noise of fan. In addition, the instantaneous pressure is constituted by the periodic time-mean component plus a random fluctuating component, thus it has discrete and broadband characteristics. It can be expected that the resulted sound field manifests itself as either discrete or broadband noise.

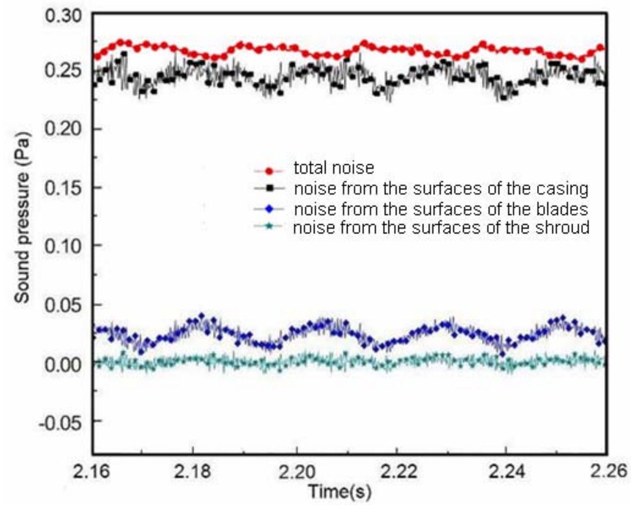


Figure 9 Sound pressure generated by different sources

To clarify further the causes of noise generated by the fan, sound pressure level spectrums are the calculated based on the obtained numerical and experimental transient data of sound pressure with a FFT algorithm and presented in figure 10. The predicted fundamental frequency of the highest intensity agrees well with the experimental one, which is coincident with the first-order of BPF of 43 Hz. It is demonstrated that the rotor-stator interaction contributes largely to noise generation in the fan, and it is the major cause for fan noise. The intensity decreases gradually for higher order harmonic frequencies. The measurements are higher than the calculated values for frequencies exceeding 200Hz, which may be caused by direct flow effect, and investigation of more suitable microphone locations are required to eliminate the flow effect. But this will be left for our near-future work.

On the other hand, for frequencies below about 200Hz, the calculated values are significantly higher than the measurement by up to 15dB at BPF (43Hz). One possible reason may be that the measurements are taken for the whole outdoor unit, whereas the physical model of flow simulation only includes the fan duct system, but excludes the other parts of the air-conditioner, such as the heat exchanger, electric motor support, and air outlet louver, which can cause some deviations from the real environment flow solution and may contributes to the larger calculated values. Moreover, as mentioned in section 3.2, the scattering and reflection effect of casing and blade surfaces are not included in the present noise prediction method, and this may be one of the causes for the deviation of numerical results from the experimental ones. All the potential causes will be identified through our near-future in-depth investigation.

To look at the relation between noise generation and flow rate of the fan, the A-weighted (A) overall sound pressure level (OASPL) is used, and its value is obtained using the A-filter, which is based on the principle of the frequency dependence on the sensitivity of human hearing, and given by

$$OASPL_A = 10\, log_{10} \sum_{i=1}^{N} 10^{(SPL_{Ai})/10} \qquad (8)$$

Table 1 $OASPL_A$ calculated results

Sound source	Fan source	Shroud source	Casing source	Total source
$OASPL_A$ (dBA)	46.3	43.7	53.6	53.9

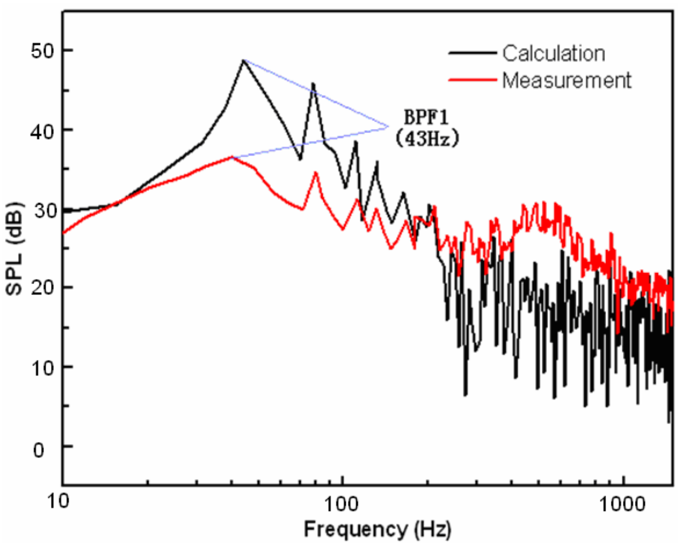


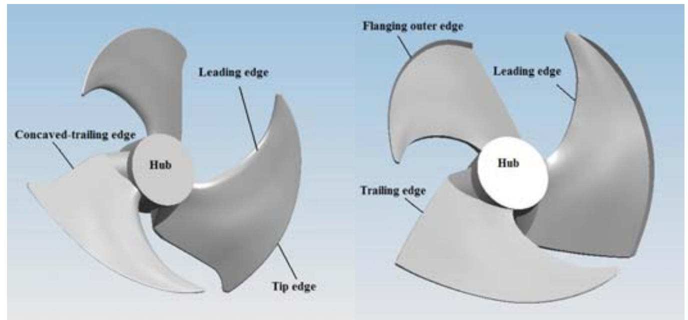
Figure 10 SPL spectral analysis of the fan

The calculated results of overall sound pressure level are given in the table 1.

4.2 Redesigned Fan

For the purpose of noise reduction, the fan is redesigned, based on the obtained flow and noise results, especially the noise generation mechanism as described in the preceding sections. The major cause for fan noise is fluctuation of static pressure in the interface region of fan rotor and casing, generated by the rotor-stator interaction, and it must be designed on the basis of improving the rotor-stator interaction to reduce effectively the noise. It is well established that the blade shape geometry plays major role in rotor-stator interaction in the fans, and an alternative to noise reduction is to improve the blade design of the fan.

In the present study, two schemes, the concaved-trailing edge blade and flanging outer-edge blade, are used to redesign the fan of the outdoor unit, as outlined in figure 11.



(a)Concaved-trailing edge blade; (b) Flanging outer-edge blade
Figure 11 Two redesigned fans

To evaluate the performance of redesigned fan, numerical simulation method as described in section 3 is used to predict the flow, performance, and noise of fan. The redesigned fan exhibits visible improvement in both flow and acoustic behavior, which is also validated by the experimental test. The three fan tests are conducted under the similar atmospheric conditions.

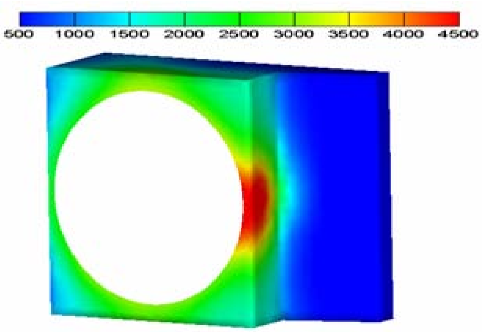
For the purpose of comparison, numerical prediction and experimental measurement are conducted respectively for the redesigned fan of two different schemes near the design flow rate of 1860 m^3/h. As well known, it is very difficult for the flow simulation converge exactly to a given flow rate due to the inherent numerical errors, and the same for the test because of measurement errors. Consequently in each test and simulation of the fans, the flow rate inevitably has somewhat deviation from the design flow. Table 2 presents the flow rates, at which numerical prediction and experimental measurement are conducted for the original and two redesigned fans, and but the deviation in flow rates is small, thus comparisons of the obtained numerical results are thought to be reasonable, that is influence of mass flow rate deviation can be neglected.

Table 2 Comparison of calculated and measured volume flow rate

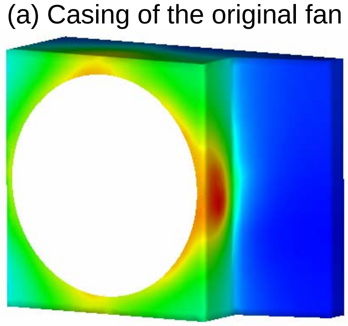
Fan \ Volume flow rate	Calculated Value (m^3/h)	Measured Value (m^3/h)	Deviation (%)
Original	1924.6	1859.6	3.8
Concaved-trailing edge	1885.1	1836.5	2.7
Flanging outer-edge	1905.5	1848.3	3.3

Figure 12 presents the predicted sound source intensity distribution of the fan casing, where the original fan result is used as a datum for comparison, figure 12(a), and results for the two redesigned fans are given respectively in figure 12(b) and (c). With the concaved-trailing edge blade, the maximum sound source intensity is reduced roughly from 4500 to 4000, figure 12(b); and it is reduced to the same level with the flanging outer-edge blade, in addition, the region of maximum intensity has shrunk largely. It is also evident that distribution of the sound source intensity is asymmetric for all the three cases, and the maximum intensity region is located at the same positions, which is caused by the asymmetric side-intake flow resulted from the asymmetric configuration of fan inlet, as shown in figure 4. As shown in figure 4(a), the fan takes in the air both axially and through the side intake duct, and this arrangement is different from the conventional fan configuration, where only an axial intake duct is often used. The side intake duct is coincidentally normal to the maximum sound intensity area, and it is thought that the influence of the side intake duct might be transmitted to the casing through interactions between the duct and rotor, and rotor and casing, and eventually increases the local intensity. However, to identify clearly the causes, some in-depth investigation is required and will be conducted in the future.

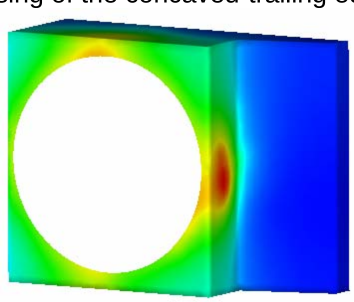
It is demonstrated that both resigned schemes of the fan is effective to reduce the sound source intensity and the flanging outer-edge scheme seems more promising.



(a) Casing of the original fan



(b) Casing of the concaved-trailing edge fan



(c) Casing of the flanging outer-edge fan

Figure 12 Comparison of the dipole sound source intensity distribution

Table 3 Comparison of measured and computed $OASPL_A$

Sound source / $OASPL_A$ (dBA)	Calculated				Measured
	Blade source	Shroud source	Casing source	Total source	
original fan	46.3	43.7	53.6	53.9	51.2
concaved-trailing edge fan	45.8	42.5	52.2	52.9	50.3
flanging outer-edge fan	45.2	42.3	51.9	52.3	49.5

Table 3 shows the calculated A-weighted OASPL of the original and redesigned fan, which compared with the experimental results. There is visible difference between the predicted and measured sound pressure levels for the three fan designs, and they are less than 2.8 dBA. Both redesigned schemes have reduced the $OASPL_A$, and the concaved-trailing edge fan produces roughly a reduction of 1 dBA and the flanging outer-edge one 1.7 dBA. The latter is more effective in the reduction of A-weighted OASPL. This conclusion is consistent with that made in the above context on the sound source intensity investigation, where the flanging outer-edge scheme seems more effective to reduce the sound source intensity.

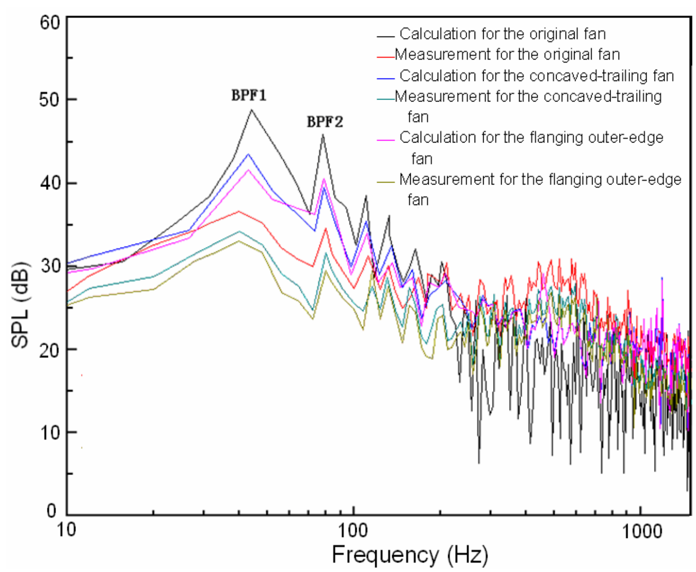


Figure 13 Comparison of SPL spectral analysis of the three fans

Figure 13 presents the measured and predicted noise spectra of the three fans. Both experimental and calculated curves for the two redesigned fans are below that of the original fan, which demonstrates that the redesign schemes are effective for the air conditioner fan noise reduction. Variations of the three fan curves versus frequency have the similar tendency; the measurement is higher than the calculated for frequency exceeding 200Hz, whereas for frequency below about 200Hz, the calculated is higher than the measurement by up to 15dB at BPF1 (43Hz).

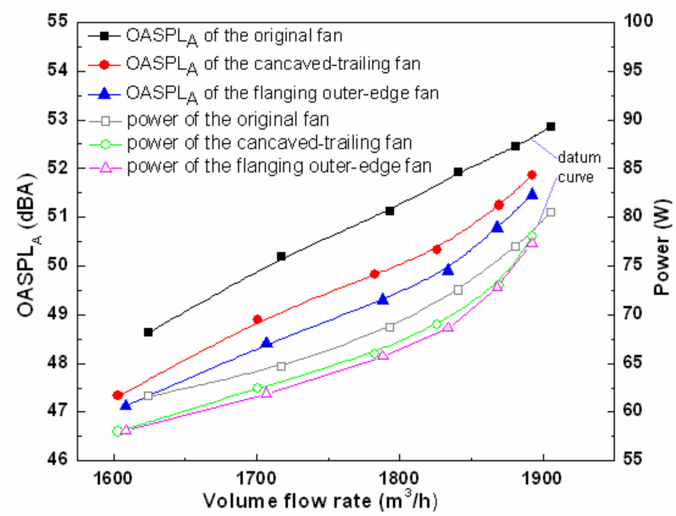


Figure 14 Comparisons of measured OASPL$_A$ and shaft power for the original and redesigned fans

Shaft power and the A-weighted OASPL are measured at different flow rates for the original and two redesigned fans, and the experimental results are represented in figure 14. It is clear that as the flow rate increase, the noise in term of OASPL$_A$ and shaft power increases for the original and also resigned two fans. In comparison with the datum curves for the original fan, the two redesigned fans have reduced the noise and shaft power simultaneously over the measured flow range. The reduction of the OASPL$_A$ is respectively about 2-2.5 dBA and 1-1.5 dBA for the flanging outer-edge fan and concaved-trailing edge fan; while the shaft power reduction is 3-5 W for the concaved-trailing edge fan and a lightly larger reduction produce by the flanging outer-edge fan.

5 CONCLUSIONS AND FUTURE WORK

A hybrid method of LES for flow simulation and acoustic analogy theory is developed and used to investigate the aerodynamic performance and noise for the air-conditioner outdoor unit fan. Performance and noise tests are also conducted for the fan, the obtained experimental results are compared with the numerical ones, and the developed hybrid method is justified. Some conclusions are arising from the present study and given below and future work is highlighted.

Conclusions

- Numerical results demonstrates that the main dipole sound sources are located mainly on the blade trailing edge and tip surface, shroud and casing inner surface, such a distribution is caused by fluctuations in instantaneous pressure associated with rotor-stator interaction.
- The casing dipole sources contribute largely to the total noise of the fan, and are the main causes for fan noise.
- It is demonstrated that modifications of impeller blade geometry are effective to reduce the fan noise. The reason is that the modified blade reduces the rotor-stator interaction and subsequent pressure fluctuations near the trailing edge casing region, and the dipole sources of casing are reduced, as a result the fan noise is reduced. Two redesign schemes, the concaved-trailing edge and flanging outer-edge blades, are used, but the latter is more effective in reducing the fan noise as well as shaft power simultaneously.
- The predicted SPL agrees well with the measured results at the fundamental frequency of the highest intensity, and the hybrid method used in the present study is justified.

Future work

In flow simulation, the causes for the deviation of the numerical results from the experimental ones will be further clarified, and some according measures will be taken to improve the simulation.

The experimental setup and measurements need to be refined in a future study to reduce the peak directivity. For example, the noise has been tested along the fan axis; this may not capture the fan peak noise, leading to a smaller measured value. A modified noise measurement is expected to be conducted within a cone around 40° - 60° (as suggested by previous investigations) to capture the peak noise.

In noise reduction, the geometry optimization method will be incorporated into the hybrid method to search for some novel configurations, which being effective for noise reduction.

6 NOMENCLATURE

BPF	blade passage frequency (Hz)
FFT	Fast Fourier Transform
FW-H	Ffowcs-Williams and Hawkings
I_{ds}	dipole source intensity
LES	Large Eddy Simulation
n	rotating speed (rpm)
p	total pressure (Pa)
p_{ref}	standard reference pressure (Pa)
p_s	static pressure (Pa)
SPL	sound pressure level (dB)
SPL_A	A-weighted sound pressure level (dBA)
OASPL	overall sound pressure level (dB)
$OASPL_A$	A-weighted overall sound pressure level (dBA)
t	time (s)

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