NOISE PREDICTION OF THE OUTDOOR UNIT OF AIR CONDITIONING WITH DIFFERENT SIZES BASED ON CFD

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SUMMARY

The investigations of the noise of two types of residential air conditioners, the up side blow-out outdoor units and the front side blow-out outdoor units are presented in this paper. Heat exchanger and associated rotor are the main parts of the studies. Aerodynamic performance is measured to evaluate the effect of changing structure size upon the flow rate and the results are used as the boundary condition for CFD simulation. Then a developing program based on vortex shedding noise model and CFD results is built up to predict the broadband noise level of outdoor unit. The relationship between the structure size and noise radiation level of outdoor unit is established. The results of the noise prediction program agree well with the experiments with small difference, and the program demonstrates of meeting the demand of engineering application for design the size of outdoor unit quickly.

INTRODUCTION

The noise produced by an outdoor air conditioning unit could be caused by a few of mechanical and aerodynamic sources, including the vibration of structures, motor noise, fan noise and fluid-solid noise, such as the interaction between flow and cooling coil, tubes and grilles. The vibration of the structures, such as compressor and shell vibration, and motor noise can be reduced to a lower level according to advanced technique of manufacture. Many investigations have shown that the main noise source of the outdoor unit is aeroacoustic induced by the air flow [1]. Recently, the market shows a demanding of quieter outdoor air conditioning unit. The stringent noise radiation requirement from the air conditioning has driven us to investigate the noise source and noise radiation regulations of the outdoor unit experimentally and numerically, and to develop an effective method in predicting aerodynamic noise of the outdoor unit. The heat exchanger and axial flow fan with partly-ducted type bellmouth are the main sources of affecting the internal flow of outdoor unit. Due to the complex internal flow of outdoor unit, the flow interaction between rotating fan and other static parts will not only affect the aerodynamic and heat transfer performance, but also increase noise radiation level. It was found that noise level correlated with internal flow, which was a function of the structure sizes of outdoor unit.

It is well known that the main aerodynamic noise sources of outdoor unit come from interactions of inlet turbulence, tip leakage vortex, vortex shedding from impeller blade and outlet grille. Crocker [2] measured the noise of a residential air conditioner unit with a two-microphone sound intensity probe to identify the noise sources from the inlet, exhaust and cabinet. Care was taken with the unit to separate the inlet and exhaust noise from the noise radiated from the cabinet. Measurements and subjective studies showed that the low frequency sound was predominantly radiated from the exhaust and inlet. Caro [3] studied the aerodynamic noise of fan system installed in an engine cooling module. It was shown that the inlet turbulence would increase discrete noise, as well as trailing edge shedding vortex increasing broadband noise. Jiang and Tian [4] investigated the relationship between air flow field and aerodynamic noise of a certain outdoor unit. Tip vortex patterns were measured by Particle Image Velocimetry. It was shown that the tip vortex was generated from blade suction surface, and the turbulence intensity would increase aerodynamic noise of outdoor unit. Zhu[5] investigated the flow field and noise performance of split-type air conditioner using Fluent® software, but the noise pressure level could not be calculated. Tian [6] has studied the possible noise source of a front side blow-out outdoor unit, including inlet turbulence noise and vortex shedding noise, and developed different noise prediction models for different noise sources.

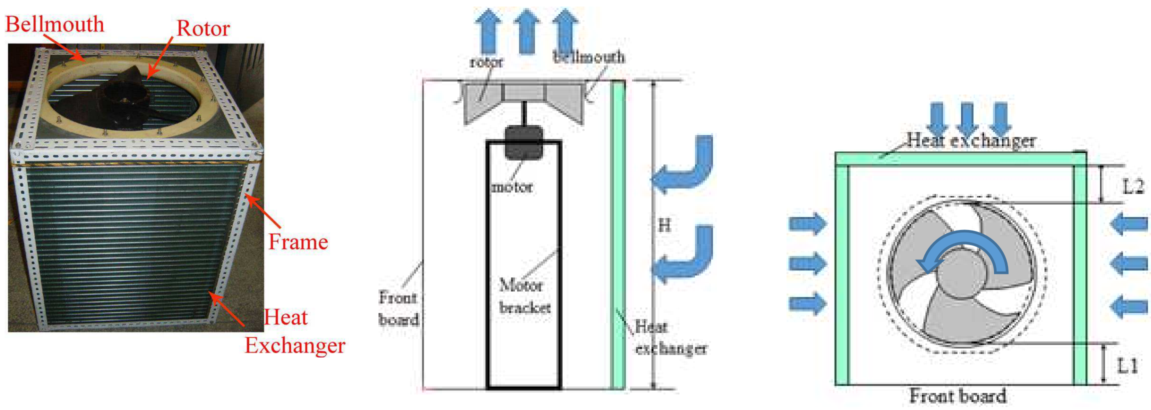
Above investigations have pointed out the main noise sources of outdoor units, and modified noise prediction model was focused on a certain outdoor unit. There are few studies on the adaptability of noise prediction model and the effect of structure sizes to the noise level of outdoor unit. Flow field of outdoor unit is a function of structure design, which would affect the aerodynamic performance of outdoor unit, as well as the overall sound pressure level. Hu [7] simulated the flow field in the outdoor unit to analyze the influence of deflecting ring to the noise generation. The relative turbulent intensities were computed to investigate the effect of the width of the ring on the noise. The computation and experiment showed that there is an optimal width of the deflecting ring corresponding to the minimum noise. There are many other parts in the outdoor unit, which affect the performance. So the study on the effect of structure sizes to the aerodynamic and aeroacoustic performance is necessary.

As for the noise generation in a low speed axial fan in outdoor unit, the force fluctuations over the rotor blade are expected to be the major sources of noise. Sharland [8] derived a rule that sound power due to the lift fluctuation of a compact plate could be expressed as a function of the correlation between the pressure fluctuations on the plate. Measurements gave a fair agreement with estimation for an isolated plate under different flow conditions. Based on Sharland’s work, Fukano [9-11] developed a very simple but physically reasonable model to estimate vortex shedding noise. In this theory, the noise source is assumed to be the lift coefficient variation produced by the local vortex shedding and the noise level varies as the sixth power of the trailing edge velocity with a length scale taken to be the thickness of the wake shed from the trailing edge. Many investigations have used this model to predict the broadband noise of rotating axial fans. The key point for this theory is the fluctuating force on the rotating blade. Lee [12] presented an analytical model for predicting the vortex shedding noise generated from the wake of axial flow fan blades. In his model, the wake model for the Kármán Vortex Street and thin airfoil theory were incorporated, and the results compared well with the experiments. Maaloum [13] brought forward an aeroacoustic approach to predict the tonal noise using the Ffowcs Williams and Hawkings (FW-H) equation. The sources of noise are located initially on the blade surface, which reduce fluctuating forces. And the forces are used to predict the tonal noise radiated by the fan in far field by means of the FW-H equation.

In present study, the noise levels of outdoor unit with fourteen different sizes have been investigated by numerical simulation. Based on noise generation mechanism, a corrected program has been developed to predict the broadband noise level using CFD results. Through the predict results, the effect of the structure sizes of outdoor unit to the noise level is obvious, and the results could be a reference for the design of outdoor units of air conditioning.

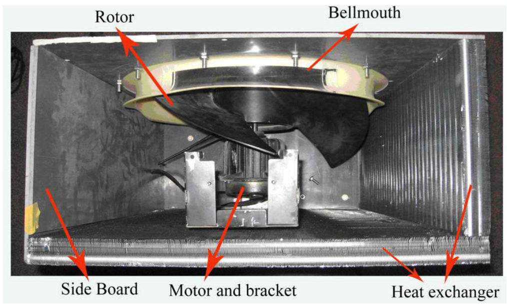
RESEARCH OBJECTS

In present study, the up side blow-out outdoor units and the front side blow-out outdoor units are investigated, both of which include a front lean axial rotor, a bellmouth, heat exchanger and shell. Different sizes of outdoor unit are built up according to changing the main parameters of the outdoor units, and the main parameters are signed by H, L1 to L5, as shown in Fig. 1 and Fig. 2. Through changing these parameters, seven different sizes are obtained for each outdoor unit, as shown in Table 1 and Table 2.

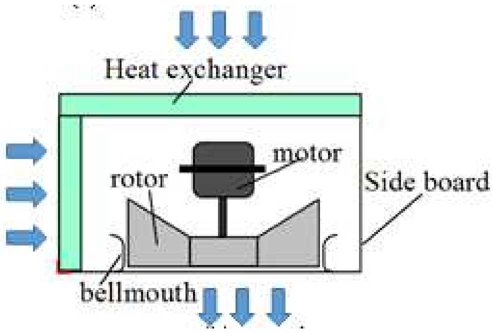


(a) up side blow-out outdoor unit (b) front view (c) top view

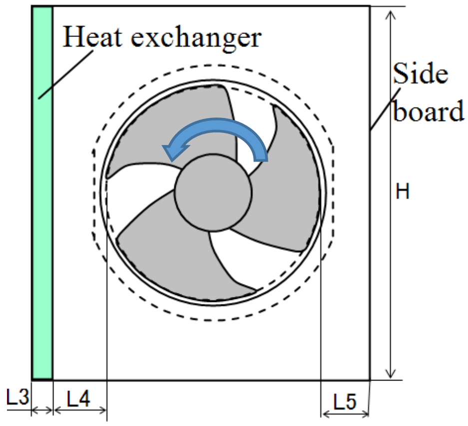
Fig. 1 Up side blow-out outdoor unit model



(a) front side blow-out outdoor unit



(b) top view



(c) front view

Fig. 2 Front side blow-out outdoor unit model

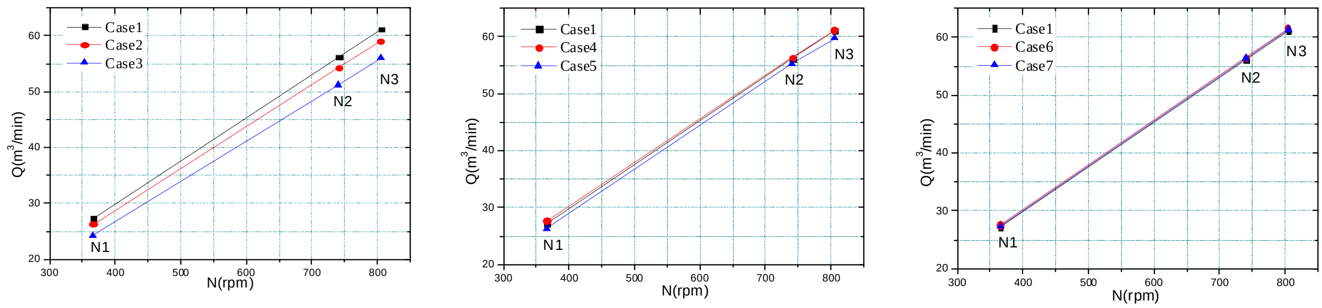
Table 1 Size of different cases of up side blow-out outdoor units

Cases		Parameters		
Base Model	Case1	H	L1	L2
Change height H	Case2	80 %H	L1	L2
	Case3	60 %H	L1	L2
Change distance L1	Case4	H	61.5 %L1	L2
	Case5	H	23.1 %L1	L2
Change distance L2	Case6	H	L1	61.5%L2
	Case7	H	L1	23.1%L2

Table 2 Size of different cases of front side blow-out outdoor units

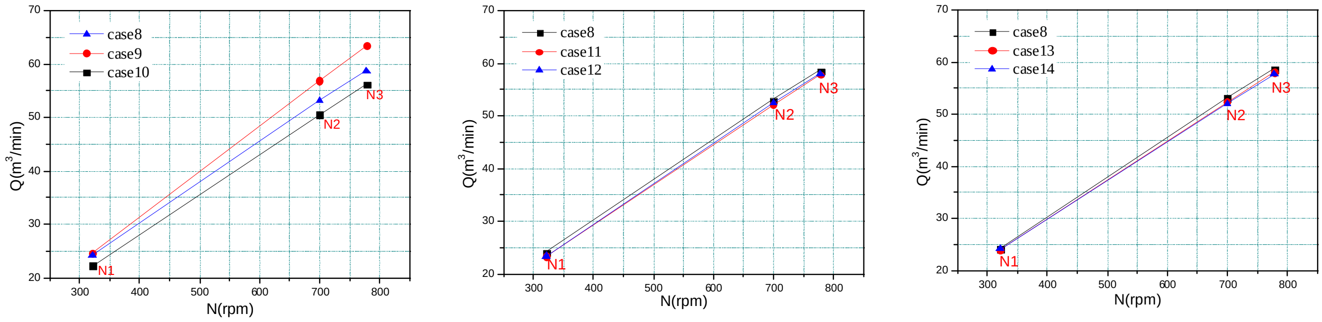
Cases		Parameters		
Base Model	Case8	L3	L4	L5
Change thickness L3	Case9	50 %L3	L4	L5
	Case10	150 %L3	L4	L5
Change distance L4	Case11	L3	61.5 %L4	L5
	Case12	L3	23.1 %L4	L5
Change distance L5	Case13	L3	L4	61.5%L5
	Case14	L3	L4	23.1%L5

Aerodynamic experiments were carried out and the measured volume flow rates vs. rotor speed curves were adopted in CFD simulation as boundary condition, as shown in Fig. 3 and Fig. 4. For up side blow-out outdoor units, with decrease of the height H of outdoor units there was apparent decrease of flux rates by 3.4 % and 8.2 % for Case 2 and Case 3 compared to Case 1 respectively. With decrease of space L1 and L2, flux rates increased first and then decreased for Case4 to Case7. However, the difference compared to Case 1 was about 1~2 %, that is to say there is little change on flow rate with decrease of L1 and L2. For front side blow-out outdoor units, it was found that a large difference exists in changing of thickness L3 of heat exchanger. Flow rate increased 8.1 % by reducing the thickness L3 of heat exchanger. On the contrary, flow rate decreased 4.1 % by increasing the thickness L3. Little change of flow rate was observed with the decrease of space of L4 and L5.



(a) Changing height H (b) Changing space of L1 (c) Changing space of L2

Fig. 3 Aerodynamic performance of up side blow-out outdoor units



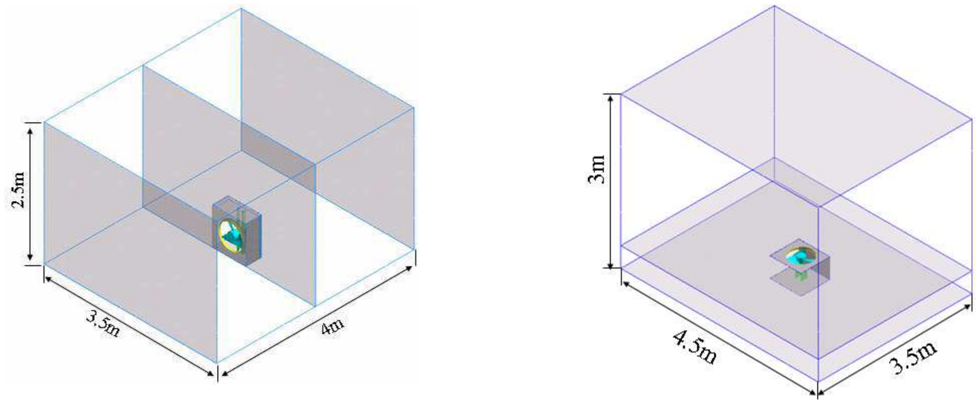
(a) Change thickness L3 (b) Change distance L4 (c) Change distance L5

Fig. 4 Aerodynamic performance of front side blow-out outdoor units

CFD SIMULATIONS

The hybrid method, i.e. CFD plus acoustic analogy theory, is used to calculate the aerodynamic noise. The noise source would be calculated on CFD platform, and then the results were introduced to Lighhill's acoustic analogy theory to compute the acoustic far field. In this investigation, commercial CFD solver Fluent® is used to simulate the incompressible viscous turbulence flow field of outdoor units. The second-order accurate upwind differencing format for the convection terms is adopted, as well as the second-order accuracy for the viscous terms. Standard k-ε turbulence model is used. The measured volume flow rate is specified for the inlet, and turbulence intensity and the characteristic length scale measured by hot wire anemometer are set for the inlet. At the outlet, a zero normal gradient for all flow variables except the pressure is applied. The outlet velocity is corrected to conserve the overall mass balance. No-slip condition is applied on the solid wall. For the heat exchanger, the porous medium and pressure drop models obtained from experimental results are used. The convergence criterion requires that the scaled residuals decrease to 10^{-5} for all equations.

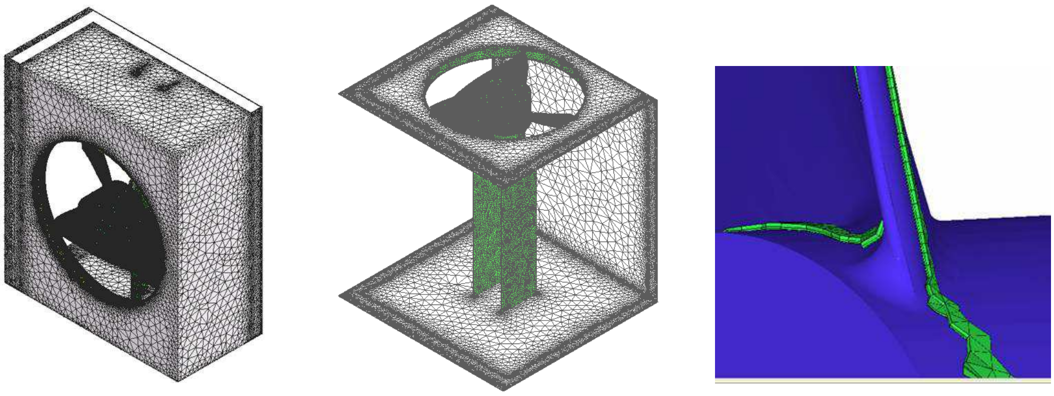
The whole computational domain of outdoor unit is shown in Fig.5, which is divided into three parts: inlet region, outlet region and outdoor unit including rotor, heat exchanger and shell. In order to achieve reasonable numerical accuracy, The inlet and outlet region have been extended by 4 m*3.5 m*2.5 m in length, width and height for the front side blow-out outdoor unit, and 4.5 m*3.5 m*3 m for the up side blow-out outdoor unit.



front side blow-out outdoor unit up side blow-out outdoor unit

Fig. 5 Whole computational domain

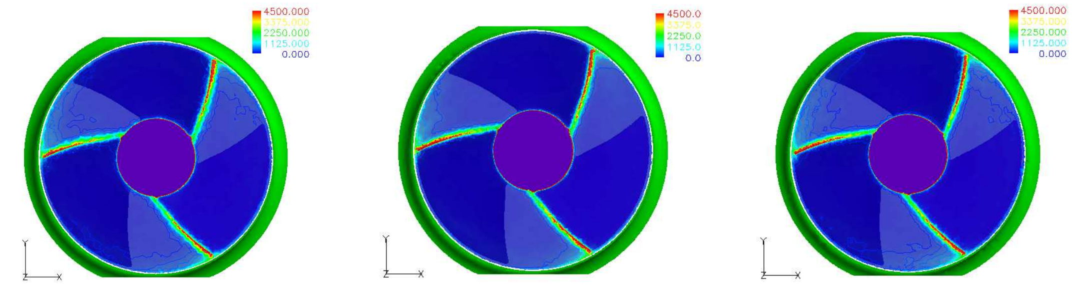
Unstructured tetrahedral grids are adopted for the full field of outdoor unit. The whole grid numbers of front side blow-out outdoor unit and up side blow-out outdoor unit are more than 4 million. In order to simulate wall boundary layers more accurately, three layers of prism grids are generated at the surfaces of axial fan blades, the mesh numbers of prism cells are about 200,000, as shown in Fig.6. The computational grid systems are achieved by commercial Ansys ICEM® software.



front side blow-out outdoor unit up side blow-out outdoor unit prism on the rotor blade

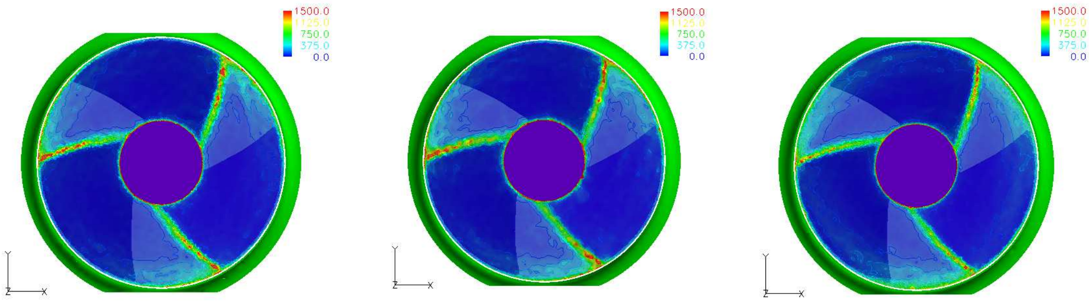
Fig. 6 CFD mesh of outdoor units

Fig. 7 and Fig. 8 show the CFD results of two types of outdoor units. From the vorticity distribution, there is strong vorticity intensity at the trailing edge of rotor. According to Powell vortex sound theory, noise is produced whenever vortex lines are stretched, dissipated and breakdown relative to aeroacoustic medium. The formulation of a region of vortex motion as an aeroacoustic source is a major step towards a physical understanding of aerodynamic noise. Powell established the correlation between vortex motion and sound generation [14]. So the main noise source is located on the trailing edge of rotor blade.



Case1 Case2 Case3

Fig. 7 Vorticity of trailing edge of rotor for front side blow-out outdoor units



Case8 Case9 Case10

Fig. 8 Vorticity of trailing edge of rotor for up side blow-out outdoor units

As shown in Fig. 9, it is found that most of the acoustic source is fastened at blade trailing edge, blade leading edge, bell mouth. The acoustic source term in leading edge may be regarded as inlet turbulence noise, and trailing edge for the vortex shedding noise. From the strength, the vortex shedding noise is the main part.

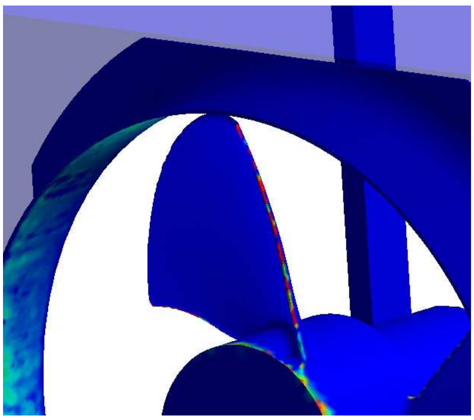


Fig.9 Acoustic source term analysis (the noise source term of Powell Vortex Sound theory is $\nabla\cdot(\vec{\Omega}\times\vec{W})$, and W is relative velocity, Ω is vorticity)

NOISE PREDICTION MODEL

According to the analysis above, the main noise source of present outdoor units is vortex shedding noise at trailing edge of rotor blades. Pressure fluctuations and turbulence shedding eddies on blade trailing edge and its development downstream are the reasons of vortex shedding noise, which affect the lift fluctuations on the blade and noise generation. According to Fukano’s model, the blade self-noise of a low-pressure axial flow fan is a function of the wake at the blade trailing edge.

A dimensional analysis then yields a noise level varying as the sixth power of the trailing edge velocity and a length scale taken to be the thickness of the wake shed from the trailing edge.

$$E = \frac{B\pi\rho_0}{1200 c_0^{\,3}} \int_{span} D(r) W(r)^6 \, dr \tag{1}$$

Where, E is the total sound power, B the number of blades, c_0 the speed of sound, D the wake width and W the relative velocity downstream the trailing edge. However, there is a presupposition that all the blades are axisymetric, so the number of blades B can be multiplied directly in Formula 1. In present study, the values of D and W are not equal with each other, so the new formula can be rewritten as bellows:

$$E = \sum_{i=1}^{B} \frac{\pi\rho_0}{1200 c_0^{\,3}} \int_{span} D_i(r) W_i(r)^6 \, dr \tag{2}$$

After the wake width Di and relative velocity Wi are obtained, the sound pressure level could be calculated according to Formula 2. In practice, the stagger angle of blade can affect the wake width. In order to solve this problem, a new correctional method considering the effects of blade stagger angle is introduced in present study. In general, the wake width should be defined as Ds in Fig. 10. while Ds is difficult to obtain, Dr is easily obtained from CFD results. The Dr and Ds have the following relationship

$$Ds = Dr \times \sin\beta \tag{3}$$

Where β is blade stagger angle.

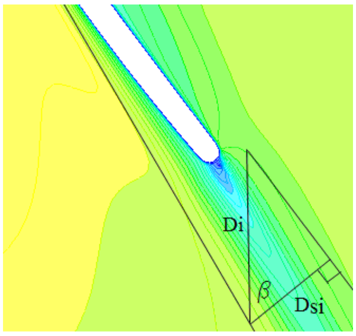


Fig. 10 Effect of stagger angles

Further, since the noise source is equivalent to dipole distribution, and also it is assumed that one-half of the radiated acoustic energy is transferred to one side of the fan, the mean square sound pressure $\overline{p^2}$ can be expressed in terms of the total sound power E by the relation

$$\frac{E}{2} = \frac{4\pi}{3} \frac{R^2}{\rho_0 c_0} \overline{p^2} \tag{4}$$

Where, R is the distance of the measuring point to the rotor on the axis of the rotor. The sound pressure level SPL is defined as

$$SPL = 10 \log\left(\overline{p^2} / p_0^{\,2}\right) \tag{5}$$

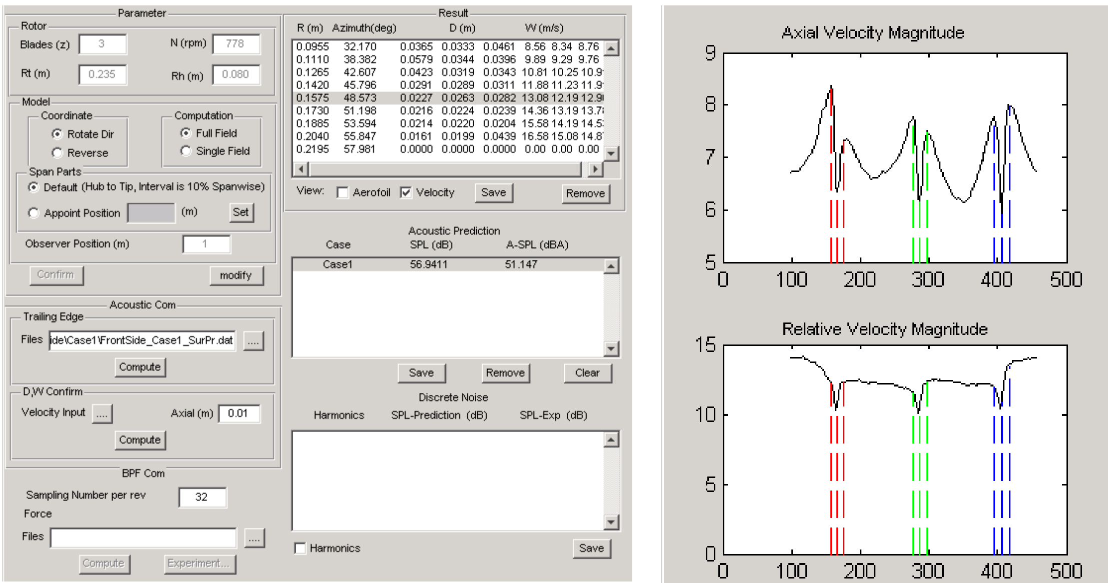
Where $p_0 = 2.10^{-5}$ Pa.

In most engineering research field, we care more about the A-weighted SPL than the linear SPL. So it is also important to get the A-weighted SPL of fourteen different outdoor units. The central frequency of vortex shedding could be calculated in semi-experience formula, as bellows [15].

$$f = 0.1 * \frac{V_{tip}}{h_m} \tag{6}$$

Where, $Vtip$ is the tip speed, $Vtip = \omega r$, and h_m is maximum thickness of the blade, which is different from hub to tip. After the central frequency of vortex shedding was determined by Formula 6, the A-weighting attenuation value could be calculated according to the A-weighted characteristic curve. Finally, the predicted A-weighting sound pressure level is achieved.

In present study, a program developed on Matlab® platform is utilized to calculate the SPL of fourteen cases. After steady CFD simulations the pressure distributions on the surface of blade at different span are extracted to compute the azimuth angle of trialing edge. Then the axial velocity and relative velocity at 10mm downstream distance to trailing edge at different span are obtained. Based on the azimuth angle, the velocity, wake width and relative velocity of three blades are obtained through the program and the results are shown in Fig.11. In Fig. 11 red, green and blue lines represent confirmed wake width of different blade respectively. A-weighing attenuated curve is inserted in program. With different central frequency is computed by Formula 6, A-weighing attenuated value can be obtained and total A-weighing *SPL* can be obtained directly from the program.



(a) main interface (b) wake width determination

Fig. 11 Interface of noise prediction program

After that, the predicted A-weighing total *SPL* could be calculated according to prediction program. The calculated results are shown in Fig. 12. It is clearly shown that the trend of prediction results is similar to the experiment results, and the maximum error is 2.09 dB(A), the minimum error is 0.32 dB(A). The results are good and suitable to the engineering application.

For the up side blow-out outdoor unit, the decrease of height H would increase noise level and decrease the flow rate. With the decrease of distance L1, the noise level of outdoor unit first reduced and then increased. So the noise level of Case 4 is the minimum. From the predict model above, the value of wake width for different cases is close to each other, the relative velocity downstream the trailing edge *W* is the most important key, Fig.13 shows the relative velocity distribution in spanwise, the value of Case 4 is smaller than that of Case 1 at most of the span. The decrease of distance L2 of upside blow-out outdoor unit can reduce the noise level, and it is not as effective as the decrease of distance L1. So for this type of outdoor unit, the distance from side board to the rotor tip must be suitable.

For the front side blow-out outdoor unit, the increasing and decrease of heat exchanger thickness L3 could reduce the noise level, but flow rate increased by reducing the thickness L3 and decreases by increasing the thickness L3 of heat exchanger. The decrease of distance L4 had little effect to the noise level, while the decrease of distance L5 could increase the noise level. The distance L5 is from side board to the rotor tip, the decrease of that could reduce the internal space of the outdoor unit, then the flow field could become complex.

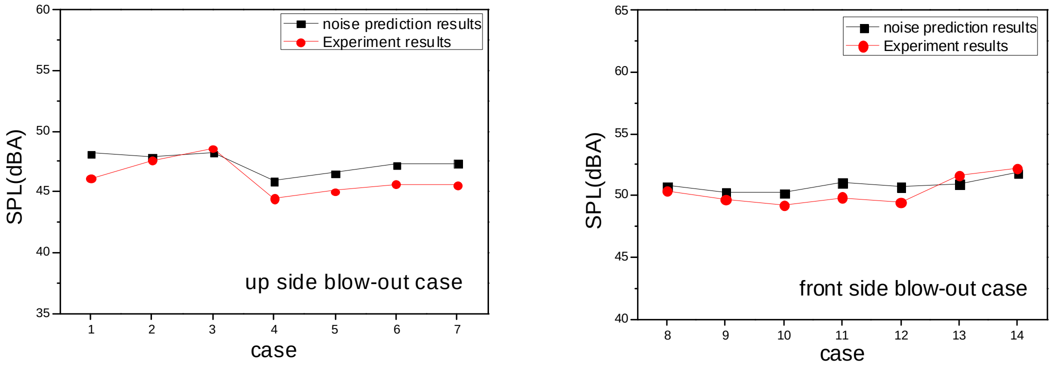


Fig. 12 Comparison between experiment and predicted broadband noise level of two types of outdoor units

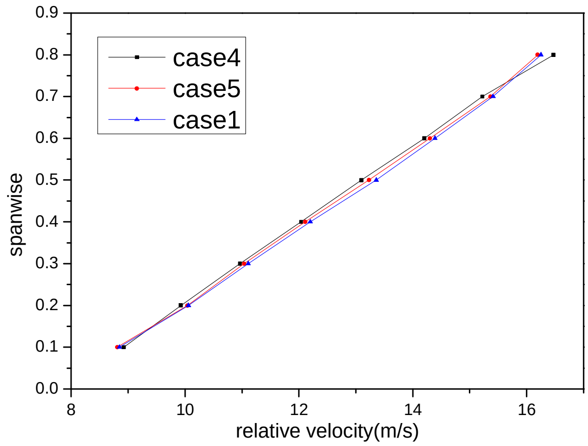


Fig. 13 Relative velocity distribution of Case1, Case4 and Case5 along spanwise

CONCLUSIONS

In present study, aerodynamic noise prediction model has been validated by outdoor unit with fourteen different cases, including up side blow-out and front-side blow-out cases. Main achievements and conclusions can be drawn as follows.

A developed Fukano's vortex shedding noise model is presented, and the noise prediction program has been set up, with correctional of stagger angle, as well as A-weighted SPL has been used in this program. The noise prediction accuracy of two types of outdoor units is quite well, with difference between measured and predicted noise level less than 2.09 dB(A). The vortex shedding model and its central frequency formula are suitable in present study. The achievement in present study could be valuable for engineers to estimate noise level at product preliminary design process.

The size of outdoor unit will affect the flow rate, noise radiation. For the up side blow-out outdoor unit, the decrease of height of heat exchanger could increase noise level and decrease flow rate. The distance from side board to the rotor tip must be suitable; a well-designed distance could make the noise level down. For the front side blow-out outdoor unit, the thickness of heat exchanger has great effect to the flow rate and noise radiation. The distance from side board to the rotor tip must not be too short, otherwise the internal space could become small and the flow field could become complex.

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