TONAL NOISE PREDICTION OF UNEVEN-SPACED BLADES AXIAL FANS BASED ON BLADE FORCE MODEL

Yadong WU[1*], Dinghao PAN[2], Zhigang PENG[1], Hua OUYANG[1]

1 School of Mechanical Engineering, Shanghai Jiao Tong University, 200240, Shanghai, China

2 School of Aeronautics and Astronautics, Shanghai Jiao Tong University, 200240, Shanghai, China

SUMMARY

To obtain the blade force, CFD modeling of three axial low speed fans, including uneven-spaced and even-spaced fans. Discrepancies and characteristics of blade forces within uneven-spaced fans were identified and thus modeled. The blade segmentation strategy was used to abstract the blade force from unsteady simulation, propagated into the far field using point force model derived from Lowson model. Then the noise modulation relation was set up based on the blade spacing angle and blade force. Verifications are done with comparing results with experimental data, as well as the prediction done by applying Lowson model directly to the calculated forced items obtained from CFD. The comparison results show that the modified noise modulation prediction model agreed well with the experimental results.

INTRODUCTION

Within an automotive engine cooling fan, there are two types of noises radiating from the flow, tonal noise and broadband noise. Usually tonal noise brings about the most discomfort to our feelings, rather than broadband noise [1][2]. This is because tonal noise has a frequency spectrum with a few discrete peaks located at particular frequencies, which cause annoyance and is highly distinctive unlike broadband noise. Many methods have been developed to reduce the tonal noise level, including swept and leaned fan, as well as uneven-spaced fan. Research has proven that if the rotor blades are spaced unevenly in the circumferential direction, the frequency features of flow patterns can be modified and the tonal noise level can thus be reduced[3][4]. Also, experiments have proved that an alteration to the spacing scheme of blades does little influence on the fan's overall aerodynamic performance, which make uneven-spaced fans a promising method for reducing tonal noise within automotive engine cooling fan modules.

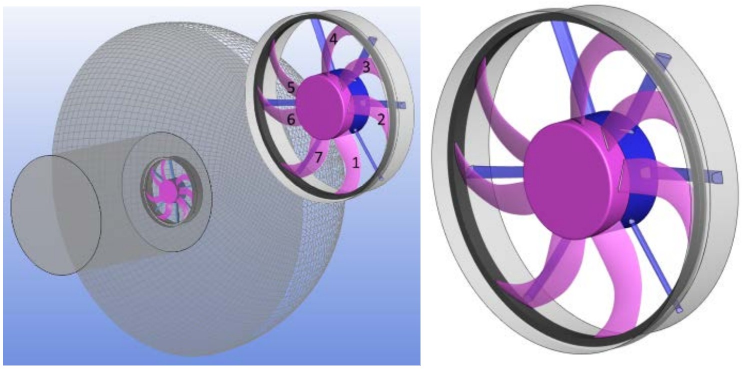
* Corresponding author, Associate Prof. Dr. WU, email: yadongwu@sjtu.edu.cn

In most cases, the spacing of an uneven-spaced fan is designed according to a Modulated Blade Spacing (MBS) method developed by D.Ewald[5]. The one-freedom-degree method is attractive in that rotors with MBS are readily balanced and modulations of the noise's frequency spectrum can be deduced with an explicit sinusoidal form[6]. However, the method is developed in a mathematic vision and no detailed study of unsteady aerodynamics is included. Lewy focused on rotors spaced in a more random manner, but still unsteady forces exerted on blades are not investigated [7]. In fact both series of study take an assumption that the aerodynamic forces on the blades are different only in their temporary phase angle. This assumption is reasonable for some fans, especially the fans of which the rotors are spaced at small irregular intervals or of which the fan blades are relatively small in span direction. However whether the assumption still holds true for all kinds of fans remains unproved. Since the appropriateness of the assumption affects directly the accurateness of tonal noises prediction, the paper is carried out primarily to investigate the characteristics of the blade forces. Transient CFD results are analyzed and blade-segmentation strategy is used to get the characteristics of the blade forces. Based on the blade force characteristics, a prediction method for tonal noise reduction is then developed with applications of Lowson model [8]. And later experiments show that the method is effective for the prediction of tonal noise reductions.

RESEARCH OBJECTS AND METHODS

Research objects

A particular automotive cooling fan was investigated in this research. Three spacing patterns of 7 fan blades are investigated in all. One is with even-spaced fan blades and it is thus noted as Fan-e, the other two are uneven and are noted as Fan-a and Fan-b. Detailed spacing angles are shown in Table 1. The model is given in Fig.1. The right side is the configuration of a fan equipped with uneven-spaced rotors. As it shows, the rotor blade (purple) is surrounded with a shrouded structure (black) which is designed due to acoustic and structural concerns. For the simulation domain, in the axial direction there are four parts, that is the cylinder-shaped inlet domain, the rotor domain, the strut domain, and a large hemisphere as the outlet domain.



(a) *(b)*

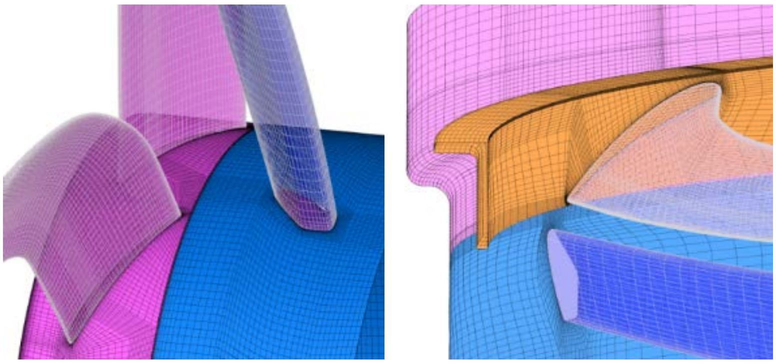
Fig.1 CFD configuration (a: fluid domain; b: fan module)

Table 1 Two uneven-spaced fans (°)

Spacing	Blade1	Blade2	Blade3	Blade4	Blade5	Blade6	Blade7
a	0	64	105.4	145	220	257	310
b	0	56	109.6	145.6	213.7	260	307

CFD methods

Structured topologies and meshes are set up using Ansys ICEM®. The grid independence is carried out first, and a set of mesh of about 3.7 million units is adopted. Each blade is surrounded with O grid, the first mesh layer normal to the wall is very small to obtain suitable y+, Fig.2 shows the mesh near the hub or the shroud, in the right picture, the orange solid part is the shroud of the blade.



(a) *(b)*

Fig.2 Mesh details (-a: near the hub; -b: near the shroud)

The CFD simulations are carried out in Ansys CFX® platform. For the boundary setting, the hemisphere outlet of the domain is set to an opening boundary close to the atmosphere. For the interface between rotor and strut, a frozen rotor mixing model is firstly adopted for steady state simulations, and transient rotor-stator model is then adopted for transient CFD simulations.

For turbulence model, four models are chosen. In order to validate the CFD results, performance curve of Fan-a is given, as shows in Fig.3. And the CFD results are from transient simulation. The flow coefficient φ and the pressure coefficient Ψ are defined with the following formula, where U_t is the rotating speed of the blade tip of rotors, Q is the volume flow, D is the diameter of the rotor.

$$\varphi = \frac{4Q}{\pi D^2 U_t^2} \quad (1)$$

$$\Psi = \frac{2\Delta p}{\rho U_t^2} \quad (2)$$

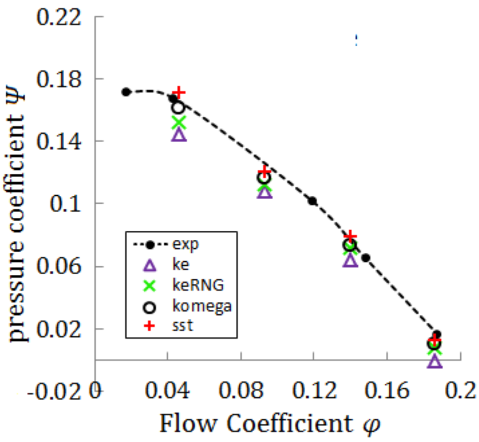


Fig. 3 Performance curve uneven-spaced Fan-a

In Fig. 3, the results show that the turbulence models have a little effect to the performance curves, By comparison, the $k-\omega$ model agrees well with the experimental results, so in this simulation, $k-\omega$ model is chosen. For the transient simulation, the time step setup is also a key factor, especially in this research due to uneven-spaced blade. First the time step independence is checked, and time step is thereby set to be 1/96 of the fan's rotating cycle. This choice of time step also assures that at least 10 time steps are needed for the narrowest flow passage to pass through.

Point-force model

Lowson model is used here to develop a tonal noise prediction method based on the blade force. The model serves well in predicting the free-field noise harmonics radiating from pointed forces which move at low speed [8]. In this paper, the noise radiated from the rotor can be taken as a compact source of sound due to the relationship between interested frequency and wavelength of this frequency range. Lowson model starts with the sound pressure formula of a moving point-force, which goes like

$$\mathrm{p(t)} = \left[\frac{y_i - x_i}{a_0 r(1-M_r)} \frac{\partial}{\partial t}\left(\frac{F_i}{4\pi r(1-M_r)}\right)\right] \qquad (3)$$

where y_i stands for the location of the monitor, which does not move like the pointed forces do. Also in the Formula (3), x_i is for the transient position of the pointed force and F_i is for the force vector of the pointed force. a_0 is the local speed of sound. $|r|$is the absolute value of $\vec{r}$, with $\vec{r} = y_i - x_i$. M_r is referred to as the Mach number of the pointed force's movement in the direction of $\vec{r}$.Harmonics of sounds then can be derived from Formula(3) to the form below.

$$c_n = -\frac{\omega}{4\pi^2}\int_0^{2\pi/\omega}\left(\frac{in\omega F_r}{a_0 r} + \frac{F_i}{r(1-M_r)}\left(-\frac{M_i}{r} + \frac{M_r r_i}{r^2}\right)\right)e^{in\omega\left(\tau+\frac{r}{a_0}\right)}d\tau \qquad (4)$$

where $F_r = \frac{(y_i - x_i)F_i}{r}$, ω is the fan's rotating speed and M_i is the Mach number of the pointed force's speed. The SPLs of sound harmonics at different frequencies are then obtained with SPL=20lg(p_A /p_0) where $\mathrm{p_{A,n}} = 2|\mathrm{c_n}|$ and p_0 is the reference pressure.

So for the Lowson model, the blade force is the key parameter. Through transient CFD simulation, the blade surface force can be abstracted, and the force can be divided into time-averaged and fluctuation items, that is $\mathrm{F} = \overline{\mathrm{F}} + \widetilde{\mathrm{F}}$. Then substitute $\overline{\mathrm{F}}$ and $\widetilde{\mathrm{F}}$ respectively into Formula(3), The harmonics cause by $\overline{\mathrm{F}}$ and $\widetilde{\mathrm{F}}$ are noted as $c_{n,\mathrm{base},\overline{\mathrm{F}}}$ and $c_{n,\mathrm{base},\widetilde{\mathrm{F}}}$. Through CFD results, the relationship between blade force and phase angle can be obtained through Least Squares Methods, and which will be discussed later. Then multiply the $c_{n,\mathrm{base},\overline{\mathrm{F}}}$ with the normalized forces. Thereby we get the harmonics radiated from the j_{th} blade as follows.

$$c_{\mathrm{n,j}} = \left(c_{n,\mathrm{base},\overline{\mathrm{F}}} \cdot \overline{[\mathrm{F}]}_j + c_{n,\mathrm{base},\widetilde{\mathrm{F}}}\right) \cdot e^{ik\theta_j} \qquad (5)$$

Then calculate the SPL spectrum of the tonal noise of the $\mathrm{j_{th}}$ blade with

$$\mathrm{p_n} = 2|\mathrm{c_n}| \qquad (6)$$

$$\mathrm{SPL_n} = 20\,\lg\left(\frac{2\left|\Sigma\left(c_{n,j} \cdot e^{ik\theta_j}\right)\right|}{p_0}\right) \qquad (7)$$

In order to obtain the blade forces from CFD results, all blades are divided into 10 radial segmentations. The strips are numbered as #1~#10, as shown in Fig.4. Every single rotating strip is assumed to radiate exactly the same noise as a single point-force does if the point-force equals to the composite force exerted on the blade and locates exactly on the center of the blades. These point-forces which are equal to the composition forces exerted on blade strips are referred to as 'blade forces'. The directions of the 'blade force' on one strip are defined within the reference frame bound to that single strip, as shown in Fig.5.

Fig.4 Blade strips

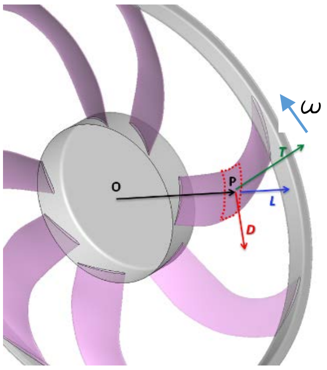


Fig.5 Direction of a point- force

In Fig.5, the red dot area represents a strip that is rotating in the ω direction. Blade force exerted on the strip is defined with three scalar components: T, D, and L. T is for the axial component, while D and L represent respectively for the drag component in tangent direction and the centrifugal component in radial direction.

RESULTS AND DISCUSSIONS

Flow field

Figure 6 shows the transient pressure on the rotor-stator interface for Fan-e and Fan-a. The black arrows in the figure point to the single blade which goes through a whole cycle. As Fig.6 shows, the high pressure zone at the pressure side of every blades are different from each other, no matter the fan is even or not. This is because of a set of strut configurations that locate downstream of the fan. However, when the blades of a fan are spaced unevenly, the discrepancy between every blade's pressure zone are much more serious than in the case of an even-spaced fan. This difference between Fan-e and Fan-a means that aerodynamic forces exert on the blades are different between even-spaced and uneven-spaced fans, which calls for the need of investigating the blade forces' direction characteristics for uneven-spaced fans.

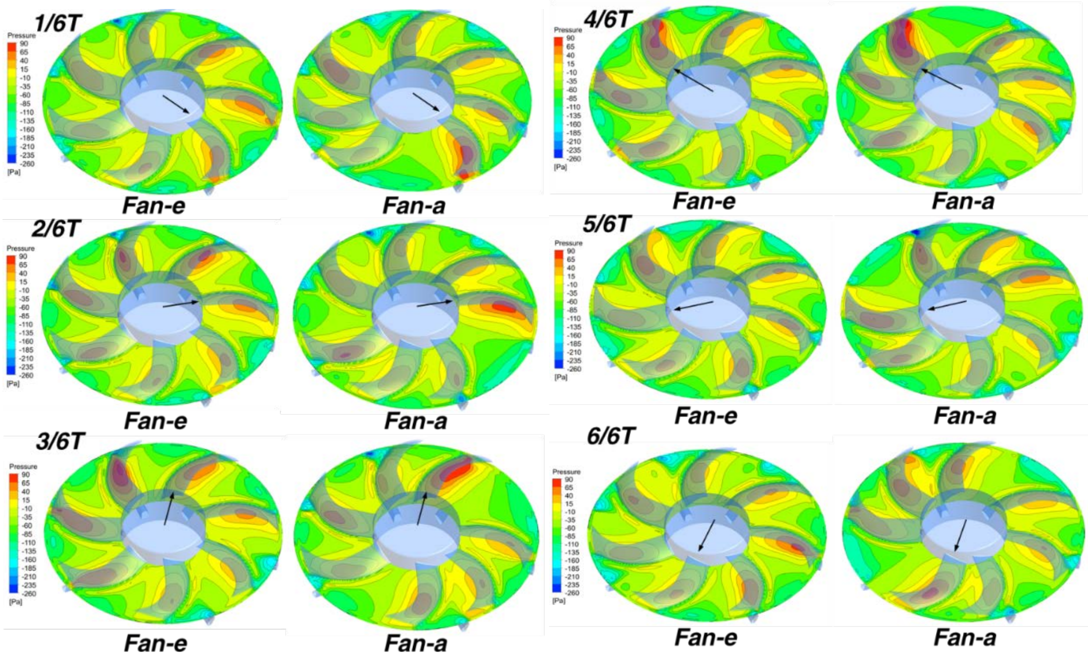


Fig.6 Transient pressure @R-S interface

Blade force characteristic

Directions of blade forces

Time average values of the blade forces exerted on the 10 strips on blade 1 of Fan-a are given in Fig.7. T is the main force compared to the other two forces D and L. The variation of T and D is smaller than that of L, that means the direction of blade force of blades slightly changes for different blade strips. The force L is increasing with the radius of blade.

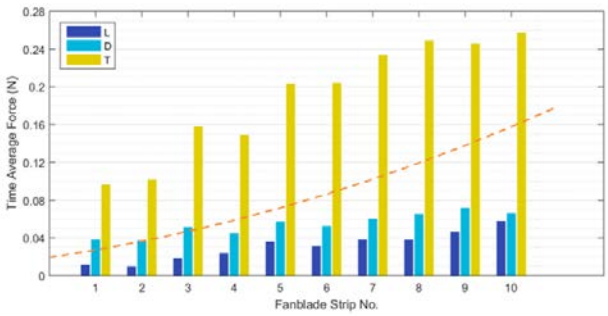


Fig.7 Time averaged blade force components on Blade 1

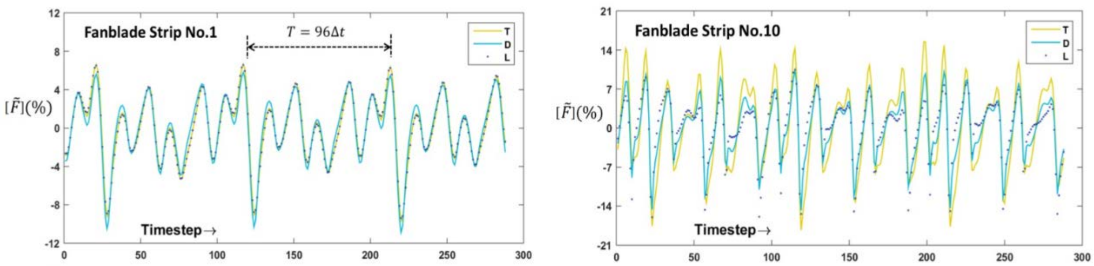


Fig.8 Normalized Fluctuation of blade force components of Fan-a

Fluctuation of the blade forces are shown in Fig.8. $\tilde{F} = (F - \bar{F})/\bar{F}$ is the dimensionless force, where F represents one scalar component of blade forces and $\bar{F}$ represent the time average value of a blade force components). As Fig.8 shows, force components fluctuate with the same cycle and the fluctuating patterns of the force components are close to each other, that means the blade force exerted on the strip keeps ‘locked’ to a particular direction in the strip-bound reference frame and do not change the direction even when the blade is rotating. Since the 'locked' direction is close to the axial direction, it is reasonable to take the force T for an investigation of the blade forces' characteristics, rather than investigate all of the three blade force components since the other two components are almost proportional to T when the blade is rotating.

Time-averaged blade forces

The colored points in Fig.9 represent time average value of the T components exerted on 70 different strips after being processed with normalization. The symbols b1 to b7 stand for balde1 to blade7. The normalization is done with the following formula

$$[T]=T/T_{base} \quad (8)$$

where T_{base} represents the arithmetic average of T on 7 blades within Fan-e and it varies as strip number changes.

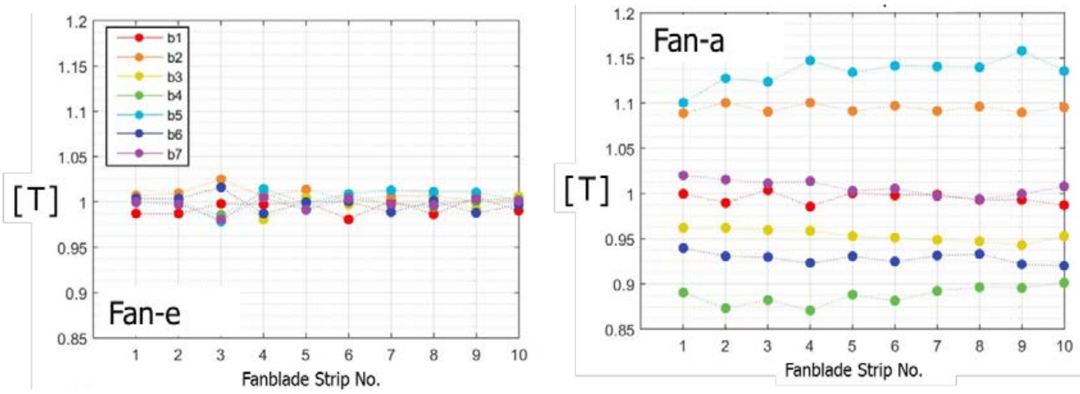


Fig.9 Normalized Time-averaged T

As Fig. 9 shows, for the even-spaced fan Fan-e, the [T] of different blades are almost the same, just a little difference due to the uneven struts downstream, which can be attributed to the circumferentially symmetric feature of it. While for uneven-spaced fan, [T] of different blades are very different from each other in the case of Fan-a. To check the mechanism of the time-averaged blade forces' difference among fan blades, contours of relative pressure are checked on the 75% radial cylinder cross-section of the uneven-spaced Fan-a, as Fig.10 shows. The pressure on the suction side of the blade are close to each other. While on the pressure side of the blade, the pressure contours vary, especially with the interaction of struts downstream, which cause the blade force magnitude change.

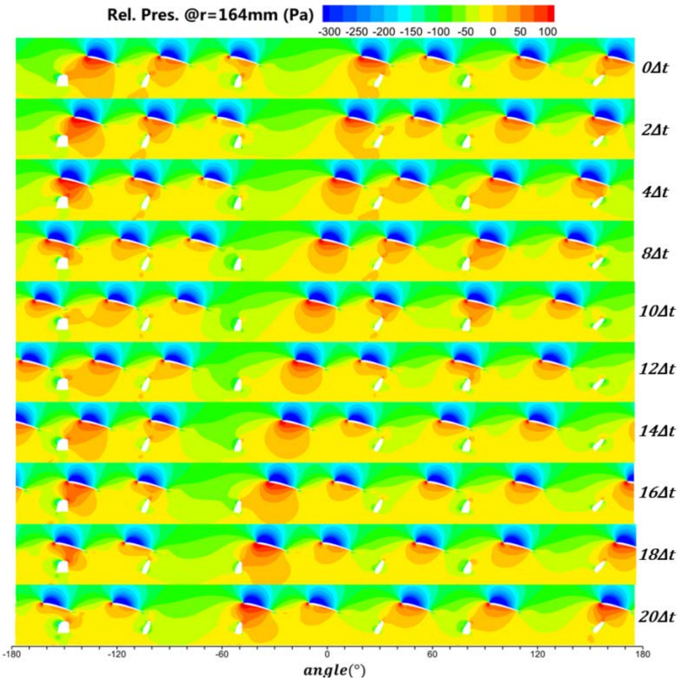


Fig. 10 Pressure Contour @ 75% radial Cylinder Cross-section

From above, the two features [T] and θ are the key parameters for the uneven-spaced fan. The model goes like $[T]_k = f([T]_{k-1}, \theta_k)$, where $[T]_k$ is the [T] one particular blade, $[T]_{k-1}$ is the [T] of the blade upstream of this particular blade, and θ_k is the phase angle of the flow passage between the two blades. Fig.11 shows the relationship of [T] and θ from CFD results of Fan-a.

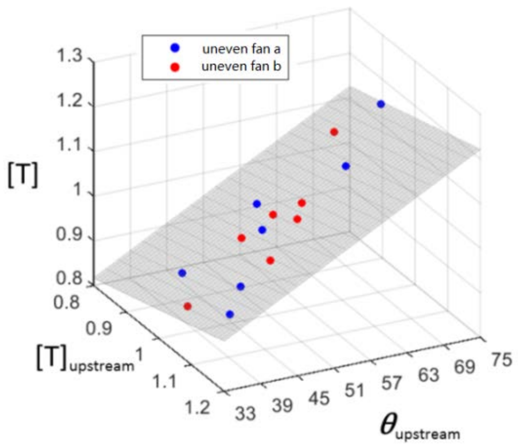


Fig.11 [T] points and the regression plane

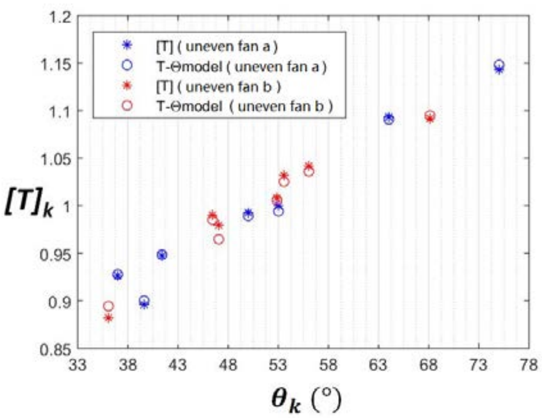


Fig.12 Prediction of the [T]=f([T]k-1,θk)

The blue points in Fig.11 can be regressed to a plane, Then these data can be processed by LSM(Least Squares Methods) to get the equation, and result goes like $[\mathrm{T}]_{\mathrm{k}} = \mathrm{A}[\mathrm{T}]_{\mathrm{k}-1} + \mathrm{B}\theta_{\mathrm{k}} + \mathrm{C}$,where A=0.239, B=0.00735, C=0.383. In fact, the fitting equation can give a prediction on the uneven-spaced fan strategy. Fig. 12 shows the predicted [T] with the equation at given blade phase angles of Fan-a. Apparently the prediction results agree well with the results obtained from CFD results. So the regressed relationship between [T] and θ can apply to an arbitrary spacing of uneven blades.

<u>Blade forces' fluctuations</u>

Fig.13 shows the axial force T fluctuations of the 7# strips of seven blades, with their temporary phases modified to be synchronous. Although the time-averaged blades forces change among blades for the uneven-spaced fan, the fluctuation of the blade forces is almost the same, with only their temporary phase not the same. And the other strips show the same trend. That means the uneven-spaced fan blades will not change the fluctuation pattern of every single blade, and all blade shares the same fluctuation pattern which is determined by the blades and struts configuration.

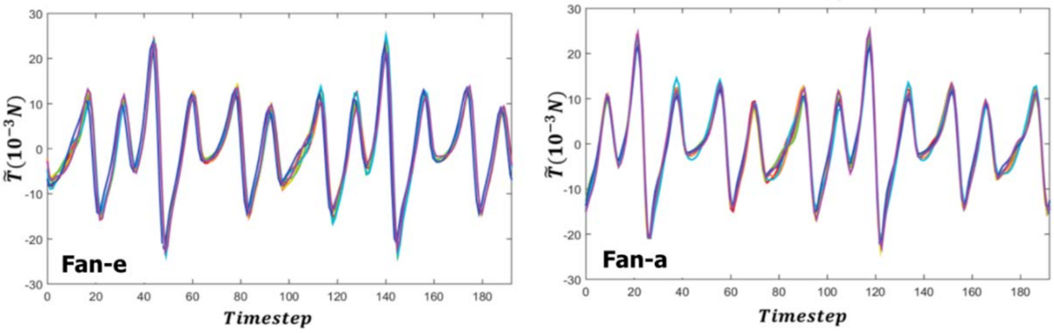


Fig.13 T Fluctuations of the 7# strips

Tonal noise prediction

When the blade forces are abstracted from CFD results, and the point-force models are set up between force and phase angle of blade spacing, the tonal noise prediction can be carried out. Fig.14 shows the predicted tonal noises at shaft frequency. For the even-spaced fan, the 1st BPF noise and 2nd BPF noise are dominant. While using the uneven-spaced blade, the 1st BPF noise and 2nd BPF can be reduced, though other tonal noise level at shaft frequency will increase. From Fig.14, the decreased SPL is 4.1dB and 15.7dB respectively using uneven-spaced Fan-a. Results of experiments show that the decrease of the tonal noise at 1st BPF and 2nd BPF compared to even-spaced fan are respectively 1.54 dB and 18.12 dB, which are close to the prediction. Fig.15 shows the prediction tonal noise level and the experiment results of two uneven-spaced fans compared to even-spaced fan. The prediction results using blade force [T]- θ model is very close to the results based on transient CFD data. That prove the blade force [T]- θ model is effective, and it is quick to get a prediction result, and it is not necessary to do the transient CFD simulation. Then the phase angel among blades can be change arbitrarily and a best distribution of phase angle of blade can be obtained.

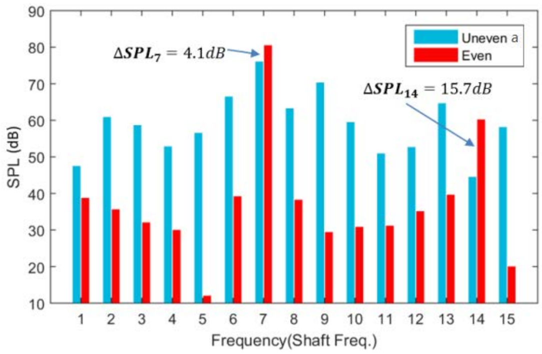


Fig.14 Predicted SPL spectrum @1m distance 45° downstream

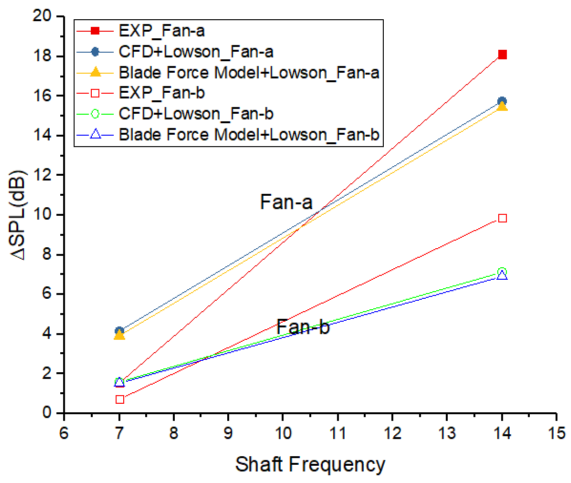


Fig.15 Noise prediction VS Experiment results

CONCLUSIONS

In this investigation, the blade forces of uneven-spaced and even-spaced fans were discussed based on CFD simulation. The blade segmentation strategy was used to abstract the blade force from unsteady simulation, propagated into the far field using point-force model derived from Lowson model. Based on the blade force characteristic, a force and phase angle [T]- θ model was brought forward, and this model can be used directly to predict the tonal noise using Lowson model. And the results show that this model is effective and efficient. In the future, more uneven-spaced fan cases must be validated to check this [T]- θ model for this kind of automotive cooling fans.

BIBLIOGRAPHY

[1] Yoshida K, Semura J, Kohri I. Reduction of the BPF Noise Radiated from an Engine Cooling Fan. SAE 2004 World Congress & Exhibition,**2004**.

[2] Becher M, Becker S. Investigation of the Applicability of Numerical Noise Prediction of an Axial Vehicle Cooling Fan. SAE 2014 World Congress & Exhibition, **2014**.

[3] R C Mellin, G Sovran. Controlling the tonal characteristics of the aerodynamic noise generated by fan rotors. Journal of basic engineering, 92(1): 143-154, **1970**.

[4] Kim Tae. Reduction of Tonal Propeller Noise by Means of Uneven Blade Spacing. Master Degree paper, University Of California, Irvine, **2016**.

[5] D Ewald, A Pavlovic, JG Bollinger. Noise Reduction by Applying Modulation Principle. Journal of the Acoustic Society of America, 49(5): 1381–1385, **1971**.

[6] PE Duncan, B Dawson. Reduction of Interaction Tones from Axial Flow Fans by Suitable Design of Rotor Configuration. Journal of Sound and Vibration, 33(2): 143–154, **1974.**

[7] S Lewy. Theoretical study of the acoustic benefit of an open rotor with uneven blade spacings. Journal of the Acoustical Society of America, 2181–2185, **1992**.

[8] M V Lowson. Theoretical analysis of compressor noise[J]. Journal of the Acoustical Society of America, 47: 371–385, **1970.**