Fan noise and Resonance Frequency Analysis in Fan-cooled Induction Motors

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In recent years, with the rise of concern in environmental problems, noise reduction in motors has become a stringent requirement. There are three kinds of motor acoustic noise: ventilation noise, electromagnetic noise and machine noise. It is assumed that ventilation noise in high speed machines consists mostly of fan noise. Fan noise depends on the rotational speed and the number of fan blades and the turbulence sound is produced by the disturbed flow of cooling air. Many studies have been done on reduction of fan blade and on turbulence noise by flow analysis based on number of fan blade and shape. However, there have been no papers describing the spatial resonance frequency of the fan spatial, which is a propagation channel of the fan noise. Thus, in this paper, the spatial resonance frequency and its resonance mode of the fan channel is analyzed by FEM and experiments. The effect of sound due to rotation and resonance mode on noise is also described.

1 Introduction

In recent years, there has been strong demand for noise reduction of induction motors due to increasing concern about environmental problems. There are three kinds of noise: ventilation noise, electromagnetism noise, and mechanical noise. Fan noise is composed of noise by rotation, expressed by the product of the number of rotations f and the number of shuttlecock sheets z, and of turbulent flow noise generated by the flow of a wind, being confused with fan noise as a typical part of ventilation noise. About rotation sound and turbulent flow sound, a lot of research (1) - (11) has been carried out on noise reduction based on examination of the number of shuttlecock sheets, and the flow analysis of shuttlecock form. Reference (11) studied a centrifugal radial fan used for an induction motor. In this study, Yamanishi et al. studied an induction motor centrifugal radial fan. They reduced the number of fan sheets from 12 to 5, and the generating frequency of sound by rotation from 720Hz to 300Hz[11] . Acoustic noise level was reduced by 2dB. However, in higher speed regions, generating frequency became higher and acoustic noise was not reduced much. Harada studied the effect of fan capacity and pressure of airflow on acoustic noise. This showed that diameter reduction of the fan results in lower noise.

These papers do not deal with spatial resonant frequency around the fan. Spatial around the fan provides a propagation channel for noise from it.

This paper describes spatial resonant modes around the fan. The following process clarifies the effect of the relation of noise and spatial resonant frequency of acoustic noise.

2 Specification of fan-cooled Induction motor for analysis

Table 1 shows the specification of the fan cool type induction motor for analysis

Figure 1 shows the motor, cooling fan and fan cover. The central section of a cylindrical fan cover inhales cooling air and the corner part inside the fan cover changes inhaled air flow from radial direction to axia direction .
The cylindrical part inside the fan cover around the motor produces axial direction flow of inhaled air around the fan of the motor. The number of fan blades is eight.

Table 1 Specification of fan-cooled Induction motor for analysis

Kind of motor	Induction Motor
Pole、Power	2 P, 5.5 kw
Rate volt	200／220 V
Frequency	0～100 Hz
Rotation	0～6000 min^{-1}($100s^{-1}$)
Fan	Radial fan
Number of blades	8
Diameter of fan cover	φ245 mm

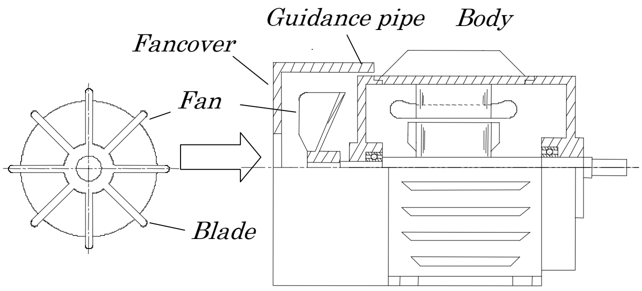


Figure1 Cross section of induction motor

3 Resonant frequency analysis by finite element method

In examining the acoustic field problem of the three-dimensional spatial of the circumference of a fan, acoustic field analysis in the spatial is carried out. Since it is important to make the acoustic characteristics clear, the finite element method calculates resonant frequency and resonance mode.

3.1 Basic Equation of Acoustic System

The finite element method to deal with the acoustic field analysis as follows.

In air, a minute volume and displacement of the element with displacement of the element $d(\xi,\eta,\zeta)$ with a size dx, dy, dz is considered. Air motion equation, will be given by a formula as (1).

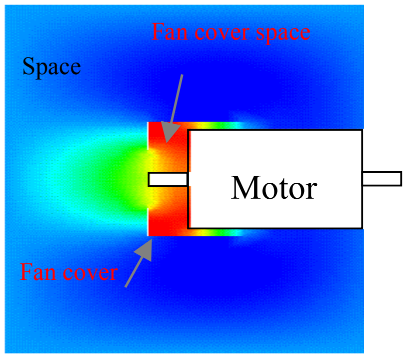
$$\rho_a \frac{\partial^2 d}{\partial t^2} = -\left(\frac{\partial}{\partial x} + \frac{\partial}{\partial y} + \frac{\partial}{\partial z}\right) p = -\nabla p \qquad (1)$$

Whire ρ_a: fluid density P: pressure on element

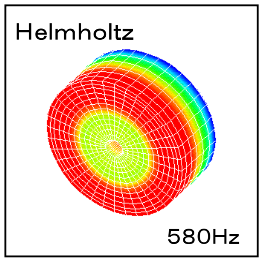
Pressure p is expressed by equation (2) with the bulk modulus of fluid:

$$p = -K\left(\frac{\partial \xi}{\partial x} + \frac{\partial \eta}{\partial y} + \frac{\partial \zeta}{\partial z}\right) = -K\nabla d \qquad (2)$$

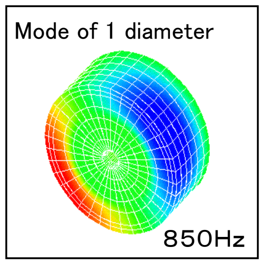
The wave equation became equation (3) from equation (1) and equation (2):



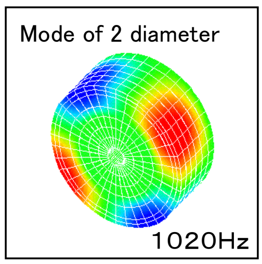
(a) Mode of Helmholtz (580Hz)



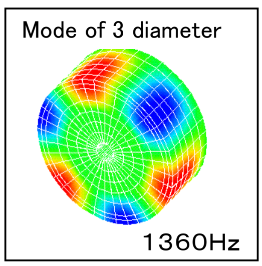
(b) Resonance mode(580Hz)



(c) Resonance mode(850Hz)



(d) Resonance mode(1020Hz)



(e) Resonance mode(1360Hz)

Figure 2 FEM model 3 dimensional spatial

$$\frac{\partial^2 p}{\partial t^2} = \frac{K}{\rho_a}\left(\frac{\partial^2 p}{\partial x^2} + \frac{\partial^2 p}{\partial y^2} + \frac{\partial^2 p}{\partial z^2}\right) = \frac{K}{\rho_a}\nabla^2 p \qquad (3)$$

To simplify the process, a 1-dimensional acoustical system is assumed. Formula (3) can be expressed as formula (4):

$$\frac{1}{K}\frac{\partial^2 p}{\partial t^2} = \frac{1}{\rho_a}\frac{\partial^2 p}{\partial x^2} \qquad (4)$$

Furthermore, if this minute volume element contacts a plane with acoustic impedance Z_n, the following boundary condition is obtained.

$$p = j\frac{1}{\rho_a \omega} Z_n \nabla p \qquad (5)$$

Where j: imaginary number nit, ω: angular frequency (rad./sec).

The general condition yields the following equation.Opening end boundary

$$Z_n = 0; \; p = 0 \qquad (6)$$

rigid wall boundary

- $Z_n = \infty; \; \nabla p = -j\rho_a \omega \xi = 0 \qquad (7)$

As shown in Figure .2, in a finite-element-method model, induction motor and a fan cover are treated as a rigid body boundary, and the sixth plane edge of the three-dimensional spatial mesh end is treated as an opening end boundary.

3.2 Finite Element Method Analysis Result

As shown in Figure 2, the FEM model expresses a 3-dimensional spatial inside the fan cover and surrounding the motor. Since it is a very thin plate mode cooling fan, this fan mode was omitted. Figure .2(a) and (b) shows calculated results of resonant frequency and resonant modes. Figure .2(b) displays the spatial inside the fan cover.

Spatial resonant frequency around the fan was found to be 580Hz. This mode at resonant frequency shows that sound pressure changes at the intake of the fan cover and that sound pressure in the fan cover becomes large.

Since the fan cover has an opening at the end of the cylinder, sound pressure changes around the motor.

As shown in Figure 2(c) and (d), mode at calculated frequencies for diameter direction in the fan cover spatial were 850Hz, 1020Hz and 1360Hz.

As these resonant modes appear in an enclosed fan cover spatial, they were considered to be standing wave depending on the diameter of the fan cover.

As shown by the results, the spatial resonance around the fan was the resonant mode, of which sound pressure changed at the intake of the fan cover.

The enclosed spatial inside the fan cover yielded the diameter mode.

4 Experiment

4.1 Experiment System

Resonant frequency and fan noise were measured in experiments.

Figure 3 shows the experimental systems. A microphone was set up 20cm away from the intake of the fan cover. An FFT analyzer analyzed the frequency of the noise from 0-100min^{-1}.

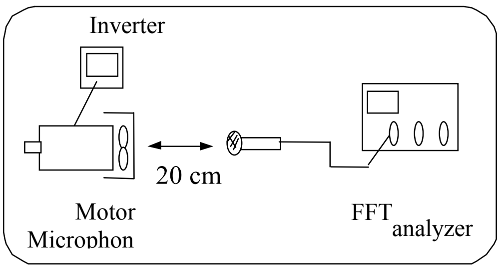


Figure 3 Experimental system

4.2 Measurement Result of Spatial Resonant Frequency in Fan Cover

A speaker is installed in the fan cover, white noise is generated, and an FFT analyzer analyzes and detects resonant frequency.

4.2.1 Noise Frequency Analysis Result of Rotation

As shown in Figure 4 (a), (b), (c) and (d) some gathered peaks existed around 580Hz and 1000Hz.

These frequencies groups did not shift when fan speed changed and neighbored FEM calculated resonance frequencies 580Hz and 1020Hz.

Therefore, the noise was regarded to be due to the resonance of flow turbulent noise.

As rotational noise frequency such as fz=560Hz approached the resonant frequency, distinguished rotational noise frequency appeared.

However, noise frequency of rotation, peak fz=720Hz was buried in the turbulence frequency peaks in all ranges.

It was estimated that the noise frequency of rotation shifted from the resonant frequency. From these results and fan experiments, noise frequency of rotation, fz constitutes a distinguished peak frequency in a group of frequency peak and provides noise level increase.

4.2.2 Measurement Result of Spatial Resonant Frequency in Fan Cover

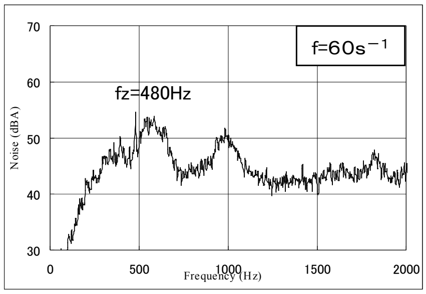
As shown in Figure 5, a speaker was installed in the fan cover, white noise was generated, and spatial resonant frequency was analyzed with the FFT analyzer.

These results were compared with FEM values in Table 2. Resonant frequency, 590 Hz , appears and is nearly the same as the value of FEM in Table 2.

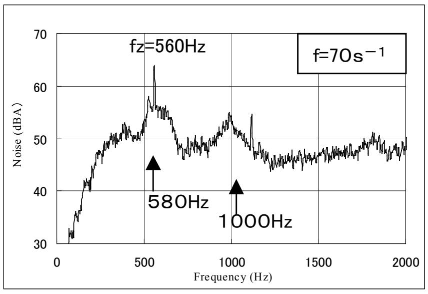
High order frequency shift from calculated values, because of the speaker inside the fan cover.

In Table 2, experimental values is mostly in good agreement with the calculated values. Resonant frequency was considered to be 590Hz.

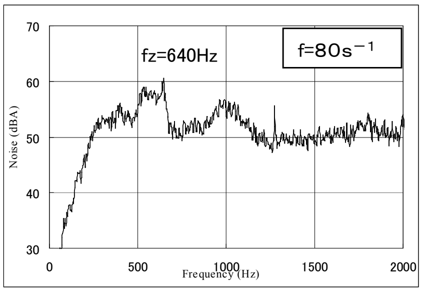
However, higher order, frequency in Table 2 was deviated, from a calculated value, since it was influenced by the speaker in the fan cover.



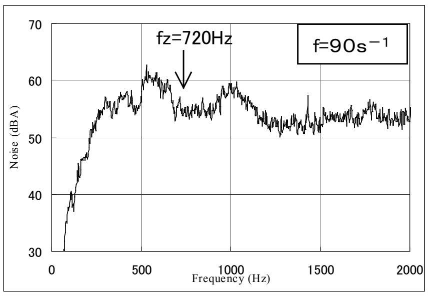
(a) driving condition :f=60s-1



(b) driving condition :f=70s-1



(c) driving condition :f=80s-1



(d) driving condition :f=90s-1

Figure 4 FFT analyzer analyzes

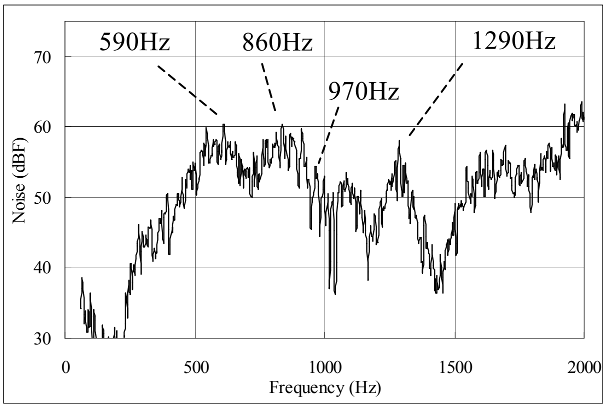


Figure 5 White noise is generated, and spatial resonant frequency

Table 2 FEM values and experimental results

Number	1 st	2 nd	3 rd	4 th
FEM values (Hz)	580	850	1020	1360
Experimental (Hz)	590	860	970	1290

5 Consideration

Fan noise is considered from the relation between resonant frequency and rotation sound frequency of rotation.

(1) Sound pressure varies at intake of fan cover and whole sound pressure becomes large at resonant frequency 580 Hz .

As shown in Figure .6, the fan cover has an intake at the end of the cylinder and sound pressure changes around the spatial near the motor. In this mode, resonant frequency is almost the same as the frequency calculated by equation (8): Helmholtz natural frequency [14].
This value is calculated from spataial volume in fan cover V, intake length, l and section area, S . The short pipe length l is modified by equation (9).

$$f_0 = \frac{c}{2\pi}\sqrt{\frac{S}{l \cdot V}} \tag{8}$$

$$l = {}^{8d}\!\big/_{3\pi} \tag{9}$$

Therefore, if the sound source fan is installed in a Helmholtz resonance vessel, the vessel will become a speaker and noise will be increased.

(2) The peak fan noise is the fan noise of rotation. The turbulent flow sound in all frequency bands is increased gradually with increase of speed , and never reduces.

(3) The sound of rotation shown in Figure . 4 (d) fz = 720Hz has disappeared. This frequency is estimated to be related to anti-resonant frequency. If the motor drives at a constant speed and is in agreement with noise of rotation and anti-resonant frequency coincides with it, no peak noise appears.

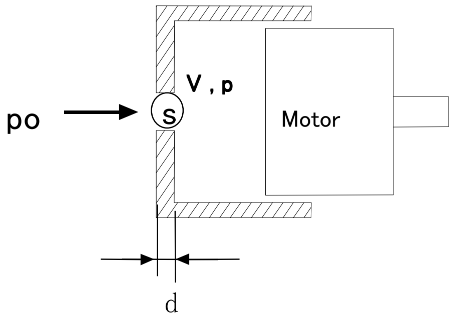


Figure 6 Model of Helmholtz

6 Conclusion

The following conclusions were obtained from analysis of spatial resonant frequency and motor fan noise.

This resonance appears inside fan cover of the motor.

(1) Since the fan cover had an opening, The spatial resonance mode around a fan was Helmholtz resonance mode.

(2) Since noise is emitted through an opening, the Helmholtz resonant frequency significantly affects noise.

(3) In the closed spatial in a fan cover, it turned out that mode exists in radial direction.

(4) The spatial resonant frequency and rotation sound is yielded as follows.

The noise frequency of rotation reaches a maximum in the Helmholtz resonant frequency zone, and disappears in the anti-resonant frequency zone.

In this paper, fan noise reduction of a fan cool type induction motor are studied.

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