A Study on Suppressing Acoustic Resonance of Interaction Tones from a Centrifugal Motor Blower*

(Modeling of the Phenomenon and a Proposal for a Method of Suppression)

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In a centrifugal blower with diffuser vanes, the noise level may increase at certain rates of revolution. In particular, the phenomenon of acoustic resonance due to tones arising from aerodynamic interaction between the impeller and the diffuser is serious. In this paper, we develop a physical model to describe this phenomenon. The model assumes that the flow paths in both the impeller and the diffuser behave as quasi-one-dimensional acoustic tubes. The model predicts the critical rates of revolution at which resonance occurs, and the prediction of the model is verified by experiments. We also propose an effective method for suppressing the resonance. This involves the use of diffuser vanes with slits. Each slit is located just behind the end of the overlapping region between neighboring vanes. The physical model determines this location. Experiments showed that the method successfully reduces sharp peaks in the noise spectrum.

Key Words: Compressor, Blower, Diffuser, Internal Flow, Noise, Interaction Tone, Acoustics, Aeroacoustics, Resonance

1 Introduction

Centrifugal blowers with diffuser vanes are generally used to provide the suction of vacuum cleaners (Fig. 1). This is because such centrifugal blowers have benefits in terms of low cost, small size and high pressure ratio. As a centrifugal blower becomes smaller, the operable rate of revolution increases and aerodynamic force is strengthened. This increases the level of aerodynamic noise as opposed to the noise from structural vibration. There are two main forms of aerodynamic noise. One is turbulent noise, which has a broad spectrum, while the other is the pure tones (interaction tones) that are generated by the unsteady fluctuation in pressure arising from aerodynamic interaction between the rotor and stator.

Figure 2 shows a typical example of a noise spectrum. We can clearly observe the dominance of the interaction tones, which consist of the blade-passing frequency (BPF) and its harmonics. Because a vacuum cleaner has a wide range of operation, the noise spectrum is also dramatically changed. The peaks that clearly indicate pure tones in Fig. 2 are particularly dominant when the mass-flow rate is low. Turbulent noise, however, becomes prevalent when the mass-flow rate becomes high. Generally speaking, turbulent noise is preferable in terms of sound quality and then pure tones can be treated as a serious problem.

It has been observed that the interaction tone is excessively maximized at certain rates of revolution, as shown in Fig. 3. At these rates, the interaction tones irritate users because of both the increase of over-all noise level and the bad sound quality. We observe a similar phenomenon with an axial flow compressor where the pattern of aerodynamic interaction between rotor and stator coincides with the pattern of acoustic modes in a duct. It is not proper to apply the same mechanism to a centrifugal blower

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Fig. 1 A motor blower with diffuser vanes

Fig. 2 Example of a noise spectrum

Fig. 3 Unexpected increase in noise level at certain rates of revolution

because there is no mechanism analogous to the duct wall. For a centrifugal fan, Kobayashi⁽¹⁾ showed that an acoustic resonance appears within the diffuser vanes when the flow path forms a one-dimensional acoustic tube with fixed pressure ends at both ends. Akaike⁽²⁾ reported on counter measures to reduce the sharp spectrum peaks by controlling the number or length of blades, or by implementing an uneven blade-pitch arrangement. Their work is helpful for designers but sometimes failed to explain the case of our blower. The major distinction lies in the distance (gap) between rotor and stator.

In this paper, we propose a new theoretical model for the acoustic resonance of interaction tones in a centrifugal blower. We limit our discussion to the case where there is very little distance between the rotor blades and diffuser vanes. This implies the dominance of potential interaction phenomenon rather than wake interaction phenomenon. We then

verified the model experimentally. We also propose a method of resonance suppression that is based on the physical principle described in the above theoretical model, and examine the effectiveness of the method experimentally.

2. Theoretical Model

2.1 Formulation of theoretical model

As the wavelength approaches the representative length of a motor blower, sound waves tend to interact with each other in both the impeller and diffuser during their propagation through the flow duct⁽³⁾. This is especially true of tonal noise with its strong wavy characteristics as compared to turbulent noise with its statistical characteristics. The sources of interaction tones are of two types, potential interaction and wake interaction. In our case, potential interaction is dominant because there is very little distance between the impeller and diffuser, i.e. (inner diameter of diffuser)/(outer diameter of impeller)=1.017.

We thought that excessively sharp spectrum peaks would appear if the potential interaction caused an acoustic resonance either in the rotor blades or stator vanes. We assume here that the noise source only exists in a spatially small area around the gap between the rotor and the stator, and that the ducted part made up by the overlapping region between neighboring blades or vanes may be thought as an acoustic resonator (acoustic tube).

Figure 4 shows an acoustic tube model of the resonance phenomenon in the case of rotor configuration (the following process of formulation can be applied to the stator configuration by changing boundary conditions). We define the overlapping region within adjacent blades (l is the length). This region is considered as a quasi-one dimensional acoustic tube with a linear variation in area and flow velocity. One end of the tube is the forced end (closed end) in terms of pressure fluctuation. The other end is the fixed end (open end; because the acoustic impedance is high enough). We add end correction δ to the fixed end (so that its total length becomes $l + \delta$). δ is calculated as follows,

$$\delta = \frac{b}{2} \left(\frac{1}{\pi} + \frac{2}{\pi} \log_e \frac{2a}{b} \right), \quad (1)$$

where a is the longer side of the rectangular cross-section of the end and b is the shorter side⁽⁴⁾.

If the wavelength is long enough in comparison with the size of the noise source (acoustically compact), sound waves propagate as plane waves through the duct. This becomes the incident wave. On the fixed end, the incident wave is reflected and becomes a reflected wave. Both waves interact in the acoustic tube. We call this phenomenon of interaction an

Fig. 4 Acoustic tube model (example of a rotor)

acoustic tube effect. Describing the sound pressure of the incident wave $p_+(x, t)$ and the reflected wave $p_-(x, t)$, we have

$$p_+(x, t) = p_{o+} \sin(\omega t - k_+ x + \varphi_{o+}), \quad (2)$$

$$p_-(x, t) = p_{o-} \sin(\omega t + k_- x + \varphi_{o-}), \quad (3)$$

p_o : amplitude

ω : angular frequency

k : wave number

φ_o : initial phase

+: indicates incident wave

-: indicates reflected wave

x : axial distance

t : time

Superimposed sound pressure $p(x, t) = p_+(x, t) + p_-(x, t)$ then becomes

$$p(x, t) = P(x) \sin(\omega t + \Phi(x)), \quad (4)$$

$$P(x)$$

$$= \sqrt{p_{o+}^2 + p_{o-}^2 + 2p_{o+}p_{o-} \cos((k_+ + k_-)x - (\varphi_{o+} - \varphi_{o-}))}, \quad (5)$$

$$\tan \Phi(x)$$

$$= \frac{-p_{o+} \sin(k_+ x - \varphi_{o+}) + p_{o-} \sin(k_- x + \varphi_{o-})}{p_{o+} \cos(k_+ x - \varphi_{o+}) + p_{o-} \cos(k_- x + \varphi_{o-})}. \quad (6)$$

Our research aims to unveil the mechanism of acoustic resonance and predict the critical rates of revolution. We don't intend to predict the noise level itself precisely. For this reason, we assume perfect reflection at the fixed end. This produces

$$p_{o+} = p_{o-} = p_o. \quad (7)$$

Applying Eq.(7) to Eqs.(4), (5), and (6), we obtain

$$p(x, t) = P(x) \sin(\omega t + \Phi(x)), \quad (8)$$

$$P(x) = \sqrt{2p_o^2 [1 + \cos \{(k_+ + k_-)x - (\varphi_{o+} - \varphi_{o-})\}]}, \quad (9)$$

$$\tan \Phi(x) = \frac{\sin \left\{ \frac{(k_- - k_+)x + (\varphi_{o+} + \varphi_{o-})}{2} \right\}}{\cos \left\{ \frac{(k_+ - k_-)x - (\varphi_{o+} + \varphi_{o-})}{2} \right\}}. \quad (10)$$

The appropriate initial phase is chosen and the superimposed wave profile is then transformed to

$$P(x) = \sqrt{2p_o^2 [1 - \cos \{(k_+ + k_-)x\}]}, \quad (11)$$

$$\tan \Phi(x) = \tan \left\{ -\frac{(k_+ - k_-)x}{2} \right\}. \quad (12)$$

In terms of rotor configuration, the inlet is a fixed end ($x=0$) and the outlet is a forced end ($x=l+\delta$). When a standing wave appears in an acoustic tube, acoustic resonance is going on. Under this condition, there is a node of the standing wave at the inlet, and there is a loop of standing wave at the outlet. This condition of resonance at the critical rates of revolution is described by $P(l+\delta) = \sqrt{4p_o^2} = 2p_o$ in Eq.(11). Alternatively we have

$$(k_+ + k_-)(l+\delta) = \pi + 2n\pi \quad (n=0, 1, 2, \dots)$$

$$\Leftrightarrow N = \left(\frac{n+1/2}{s} \right) \left(\frac{60}{Z(l+\delta)} \right) \left(\frac{c_+ c_-}{c_+ + c_-} \right). \quad (13)$$

N : critical rate of revolution [min^{-1}],

s : order,

c : speed of sound [m/s],

Z : number of blades (number of stator vanes for rotor-side effect and vice versa)

From Eq.(13), the critical rate of revolution proves to be a function of the number of the blades, the speed of sound and the length of the overlapping region. The speed of sound is expressed as

$$c_+ = a + u, \quad (14)$$

$$c_- = a - u.$$

a : average speed of sound in a stationary medium

u : average mean flow velocity in the duct

Equation (13) then becomes

$$N = \left(\frac{n+1/2}{s} \right) \left\{ \frac{60}{Z(l+\delta)} \right\} \left(\frac{a^2 - u^2}{2a} \right) \\ = \left(\frac{n+1/2}{s} \right) \left\{ \frac{30}{Z(l+\delta)} \right\} \{a(1 - M^2)\}. \quad (15)$$

M : average Mach number $\left(= \frac{u}{a} \right)$

The Mach number's effect on the critical rate of revolution is to the 2nd power. With a simple estimate of the velocity triangle and thermodynamic calculation, we can use Eq.(13) or (15) to predict several critical rates of revolution.

If the Mach number is 0, Eq.(15) is simplified to

$$N = \left(\frac{n+1/2}{s} \right) \left\{ \frac{30}{Z(l+\delta)} \right\} a. \quad (16)$$

The form of Eq.(16) is identical with that of the well-known solution to the problem of a simple acoustic tube.

We need the speed of sound (taking the mean flow velocity into account) as well as the geometrical data in order to predict the case for an actual blower. We'll now look at how to calculate the speed of sound. To start with, we assume a one-dimensional variation of area (i.e. quasi-one-dimensional acoustic tube)

$$A(x) = A_{ent} + qx. \quad (17)$$

$A(x)$: area of cross-section at x

q : variation of area per unit length along the flow path

ent : indicates the inlet of the overlapping region

From the continuity equation, $A(x)u(x) = A_{ent}u_{ent}$, we get

$$u(x) = \frac{A_{ent}u_{ent}}{A(x)}. \quad (18)$$

The speed of sound propagating in the downstream direction is

$$c_+ = \frac{1}{l+\delta} \int_0^{l+\delta} (a + u(x)) dx \\ = a + \frac{A_{ent}u_{ent}}{q(l+\delta)} \log_e \left(\frac{A_{ent} + q(l+\delta)}{A_{ent}} \right). \quad (19)$$

The speed of sound only changes by 0.6 m/s with the temperature change of one degree. On the other hand, the flow velocity changes by at least several tens of meters per second. This allows us to neglect changes in the speed of sound, so that we are able to use an averaged speed of sound which is independent of x .

Similarly, the speed of sound propagating in upstream direction becomes

$$c_- = \frac{1}{l+\delta} \int_0^{l+\delta} (a - u(x)) dx \\ = a - \frac{A_{ent}u_{ent}}{q(l+\delta)} \log_e \left(\frac{A_{ent} + q(l+\delta)}{A_{ent}} \right). \quad (20)$$

The averaged speed of sound in a stationary medium is expressed as

$$a = 331.5 + 0.6 \left(\frac{t_{ent} + t_{ext}}{2} \right). \quad (21)$$

t_{ent} : static temperature at the inlet of overlapping region

t_{ext} : static temperature at the outlet of overlapping region

At this point, we can calculate c_+ , c_- by using geometrical data and the velocity triangle.

To obtain the static temperature data required to solve Eq.(21), we carry out a calculation on the basis of the conservation law of energy; as follows.

[1] The rotor

$$C_p T_A = C_p t_{i-ent} + \frac{1}{2} u_{i-ent}^2. \quad (22)$$

$$C_p T_B = C_p t_{i-ext} + \frac{1}{2} u_{i-ext}^2. \quad (23)$$

T_A : total temperature of incoming flow [K]

T_B : total temperature of compressed flow [K]

C_p : specific heat at constant pressure [J/K]

t : static temperature [K]

u : absolute velocity [m/s]

$i-ent$: indicates the rotor's inlet

$i-ext$: indicates the rotor's outlet

Applying the conservation law of energy and taking the rotor's power balance into account, we obtain

Fig. 5 Prediction of critical rate of revolution (for the rotor)

$$C_p T_A + \frac{P}{\rho_o \left(\frac{Q}{60} \right)} = C_p T_B. \quad (24)$$

P : impeller output power [W]

ρ_o : density [kg/m³]

Q : mass-flow rate [kg/s]

From Eqs.(22), (23) and (24), the averaged static temperature is obtained as

$$\left(\frac{t_{i-ent} + t_{i-ext}}{2} \right) \\ = \frac{1}{2} \left\{ 2 T_A + \frac{60(\kappa-1)P}{\rho_o Q \kappa R} - \frac{\kappa-1}{2\kappa R} (u_{i-ent}^2 + u_{i-ext}^2) \right\}. \quad (25)$$

κ : ratio of specific heat,

R : gas constant

[2] The stator

$$C_p T_B = C_p t_{d-ent} + \frac{1}{2} u_{d-ent}^2 = C_p t_{d-ext} + \frac{1}{2} u_{d-ext}^2. \quad (26)$$

$d-ent$: indicates the inlet of the stator

$d-ext$: indicates the outlet of the stator

The averaged temperature becomes

$$\left(\frac{t_{d-ent} + t_{d-ext}}{2} \right) = T_B - \frac{\kappa-1}{4\kappa R} (u_{d-ent}^2 + u_{d-ext}^2). \quad (27)$$

2.2 Analytical result

Figure 5 shows the analytical results obtained with our theoretical model for the amplitudes of 1st (fundamental) and 2nd harmonics of the BPF. If compressibility of the fluid can be neglected, the acoustic power is known to be proportional to the 6th power of fluid velocity (i.e. the amplitude of sound pressure is proportional to the 3rd power of fluid velocity or the rate of revolution). We apply this assumption in order to calculate p_o in Eq.(11), and then we can get $P(x)$ in Eq.(11). In Fig. 5, we compare the predicted variation in noise level with the experimentally measured values. As the prediction of the noise level itself is not our goal, the horizontal

Table 1 Dimensions of the fan system

	Z	D_{in}	D_{out}	l	a	b	δ
Impeller	8	-	90	50	11.0	6.0	3.4
Diffuser	13	96.7	115	20	8.7	6.0	3.0

* unit of length [mm]

D_{in} : Inner diameter of the impeller or diffuser

D_{out} : Outer diameter of the impeller or diffuser

Fig. 6 Distribution form of $P(x)$

axis for each pair of plots is adjusted to separate the two graphs. The dots in the figure designate the critical rates of revolution predicted by Eq.(13).

Figure 5 shows that our model accurately predicts the critical rates of revolution and also qualitatively characterizes the variation in noise level. This agreement implies, in our case, the large-scale variation in noise level is mainly dominated by the impeller's geometry (and its acoustic tube effect) rather than the diffuser's geometry that is relatively smaller. Besides, if we chose an acoustic tube model with two fixed ends instead of our assumption, the width of the variation in noise level would change.

The spatial waveform of a standing wave may also be obtained from Eq.(11). For instance, Fig. 6 shows the amplitude distribution of the standing wave at $18\,500\text{ min}^{-1}$ at which there is strong acoustic resonance with the 2nd harmonic; as shown in Fig. 5.

Here is a summary of the geometrical data used in the theoretical analysis.

3. Verification of the Model

3.1 Experimental apparatus

We measured unsteady pressure on the diffuser wall to verify the concept of our theoretical model. This experiment was carried out only at low rates of revolution due to the performance limits of the experimental facility. Using a low rate of revolution is, however, suitable for verification of the model, because there is less effect from compressibility of fluid (non-linearity of acoustic wave), which is beyond our assumption.

The unsteady pressure was first measured at $11\,000\text{ min}^{-1}$, where acoustic resonance had actually been observed, as shown in Fig. 5. The theoretical model predicted the resonance of the second harmonic of the BPF in the rotor (Fig. 5) and the model predicts

Fig. 7 Experimental apparatus

simultaneously another resonance of the third harmonic of the BPF in the stator. This coincidence of resonance phenomena strongly increases the sound pressure level (see the experimental data in Fig. 5). The second measurement of the unsteady pressure was conducted at $14\,300\text{ min}^{-1}$, where no resonance happened.

Figure 7 is a schematic diagram of the experimental apparatus. The motor and blower are set apart and connected by a drive shaft. The motor is covered by an absorption material to prevent radiation of the noise. The pressure chamber used to measure the mass-flow rate and pressure ratio has sides of different lengths and contains another absorption material on the walls inside. These treatments enable to prevent a degeneration of the resonance frequencies of the chamber itself. The blower is inserted between upper wall (microphone holder) and lower wall. The upper wall has several holes to measure unsteady pressure, which is converted to electrical signals by the microphones. We used 6.35-mm (1/4-inch) microphones which have a resolution of up to 20 kHz. We checked the Helmholtz resonance frequency of the cavity between the microphone head and measurement hole in advance, and designed it to be much higher than the frequency being measured. Outputs from microphones are recorded on a data-recorder system before they are analyzed with a computer. The level of noise for the over-all level evaluation was measured at the distance of 1.5 m from the rotational axis within the plane of the impeller's rotation. The experimental apparatus measures the blower's performance to use with the theoretical model prediction.

Fig. 8 Waveforms of local pressure perturbation at 11 000 rpm

3.2 Experimental result

Figure 8 shows the result of unsteady pressure measurement at $11\,000\text{ min}^{-1}$. Measurement points A and B are schematically illustrated in Fig. 8. Point A was located on the wall just downstream of the impeller, where it is possible to observe the acoustic tube effect of the rotor. Point B was located within the overlapping region where it is possible to observe the acoustic tube effect of the stator. The waveforms in the figure designate the time variation of pressure fluctuation after averaging the results for 128 periods. This impeller model has eight blades so that the waveform graph corresponds to one rotation of the impeller.

Obviously, the 2nd harmonic of the BPF is dominant at point A. The pressure fluctuation at point A indicates the behavior of the standing wave in the rotor. Our model similarly predicts that, at the rotor's exit (point A), there is the loop of a standing wave induced by 2nd harmonic of the BPF at $11\,000\text{ min}^{-1}$. Although the result of measurement doesn't directly prove that this standing wave has a fixed end at the rotor's inlet, this situation is likely, because our assumption of a fixed end explains the distribution of the critical rates of revolution as shown in Fig. 5. If we assumed a forced end instead, the distribution would change. Therefore, we have ensured the validity of the theoretical model in terms of the rotor.

At point B, however, we see that the 2nd harmonic of the BPF is no longer dominant. Instead, the 3rd harmonic of the BPF is strong in this region. This fact indicates the existence of another standing wave in this region. Our model predicts that another standing wave will be created by the 3rd harmonic of the BPF in the stator region. This observation of the spatially localized resonance phenomena induced by different harmonics at the same time is remarkable. We have thus validated our model in terms of the

Fig. 9 Waveform of local pressure perturbation at 14 300 rpm

Fig. 10 Slit diffuser

stator.

If the rate of revolution changes, so does the waveform of pressure fluctuation, and the acoustic resonance phenomena disappear. We show an example at $14\,300\text{ min}^{-1}$ in Fig. 9. Apparently, the fundamental BPF has become prevalent (this is the normal case without resonance). Hence, we can conclude that acoustic resonance happens when a standing wave, which has both a fixed end and a forced end, is established in either the rotor's or stator's overlapping region. And this phenomenon only occurs when the impeller and stator are configured with a small gap. Our theoretical model is able to predict the critical rates of revolution at which resonance appears.

4. Method of Noise Suppression

We thought that it would be possible to suppress the interaction tone by reducing the strength of a standing wave described with our theoretical model. We utilized a special device called a slit diffuser. This is shown in Fig. 10. Slits or holes are made at the node of the standing wave. The location of the node is then decided with Eq. (1). The diffuser vanes are inclined towards each other, so the slit connects the nodal part with a non-nodal part. The result is the damping, via the slits, of the pressure fluctuation that is driven by Helmholtz resonance. The slit diffuser also weakens the propagation of sound waves from the impeller, including those that are strengthened by acoustic resonance in the region of the impeller. This is because holes or slits shift the phase of the propagating wave. Although the similar treatment can be

Fig. 11 Effect of the slit diffuser vanes

applied to the impeller, we took the loss of aerodynamic performance into account and chose the diffuser rather than the impeller.

We show the effectiveness of the slit diffuser in Fig. 11. The horizontal axis shows the rate of revolution and the vertical axis shows the over-all noise level (level of the interaction tone almost decides over-all noise level). The thin line in Fig. 11 is for the ordinary diffuser and the wide line is for the slit diffuser. The dots in Fig. 11 indicate the critical rates of revolution predicted by the theoretical model for the 1st and 2nd harmonics of the BPF only.

The noise level was dramatically reduced at all operating points, and this was particularly so at the critical rate of revolution. The reduction was up to 8 dB. The model's predictions are also in good agreement with the experimental results. The loss of aerodynamic performance was measured by total efficiency and was only between 0.2–0.5% at the design point. Because the slit is located relatively deep in the downstream region, the difference in the static pressure between both sides of the diffuser vane is not large. The result is that little performance is lost.

5. Conclusion

In a centrifugal blower with diffuser vanes, the noise level may increase at particular rates of revolution. This is especially significant when there is little distance between the impeller and the diffuser. We have obtained the following results on this phenomenon.

(1) Acoustic resonance occurs when a standing wave appears in the overlapping region in either the rotor or stator region. In terms of pressure fluctuation, the standing wave has a forced end at the region of blade interaction and a fixed end on the opposite side. This resonance causes an excessive increase in the level of the interaction tone.

(2) We formulated a theoretical model to describe the resonance phenomenon. The model is able to make good predictions of the critical rates of revolution at which resonance occurs.

(3) We proposed the concept of slit diffuser in order to suppress resonance. This treatment is able to reduce tonal noise by as much as 8 dB with a minimum of aerodynamic loss.

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