Aeroacoustic Characteristics and Noise Reduction of a Centrifugal Fan for a Vacuum Cleaner

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The aeroacoustic characteristics of a centrifugal fan for a vacuum cleaner and its noise reduction method are studied in this paper. The major noise source of a vacuum cleaner is the centrifugal fan. The impeller of the fan rotates at over 30000 rpm, and generates very high-level noise. It was revealed that the dominant noise source is the aerodynamic interaction between the rotating impeller and stationary diffuser. The directivity of acoustic pressure showed that most of the noise propagates backward direction of the fan-motor assembly. In order to reduce the high tonal sound generated from the aerodynamic interaction, unevenly pitched impeller and diffuser, and tapered impeller designs were proposed and experiments were performed. Uneven pitch design of the impeller changes the sound quality while the overall sound power level (SPL) and the performance remains similar. The effect of the tapered design of impeller was evaluated. The trailing edge of the tapered fan is inclined. This reduces the flow interaction between the rotating impeller and the stationary diffuser because of some phase shifts. The static efficiency of the new impeller design is slightly lower than the previous design. However, the overall SPL is reduced by about 4 dB(A). The SPL of the fundamental blade passing frequency (BPF) is reduced by about 6 dB(A) and the 2^{nd} BPF is reduced about 20 dB(A). The vacuum cleaner with the tapered impeller design produces lower noise level than the previous one, and the strong tonal sound was dramatically reduced.

Key Words : Vacuum Cleaner, Aeroacoustic Noise, Centrifugal Fan, Low Noise, Performance, Specific Noise Level

1. Introduction

The centrifugal fan has been widely used in home appliances due to the high static pressure and compact size. Universal type motor has been applied to the vacuum cleaner due to its low cost and compact size. Recently the requirements of the compactness, high power and light weight for the vacuum cleaner has been increased. This requires smaller fan size and higher rotating speed. The increase of the rotating speed may cause two significant engineering problems : reliability and noise. In this paper we are concerned with the noise only.

It is well known that the noise level and sound quality of the vacuum cleaner significantly depends on the design of fan-motor assembly because it is the major noise source. The fan-motor assembly consists of an impeller, a diffuser, a guide vane, two ball bearings, a stator, a rotor, a shaft and two stationary brushes. The noise sources of the fan-motor assembly can be divided into three categories : aerodynamic, vibration, and frictional ones. The aerodynamic noise source results from the unsteady flow interaction between the wake flow of the rotating impeller and the stationary diffuser, and the turbulence flow

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at the complex flow passages. The vibration is caused by the static and dynamic unbalance of the rotating parts. The friction between the brushes and commutator also generates noise. The systematic research by Toru showed that the most significant noise is the aerodynamic noise (Toru, 1989). Among the aeroacoustic sources, the flow interaction between the rotating impeller and the stationary diffuser vane plays a major role in generating the strong tonal noise. The flow separation over the impeller and diffuser blades and inflow turbulence contribute to the broadband noise. Due to the small gap between the impeller and diffuser, the sound pressure levels at BPF and its higher harmonic frequencies are dominant in the noise spectrum of the vacuum cleaner's fan.

As the low noise is an essential requirement for the fan design, various noise reduction methods have been studied for last three decades (Neise, 1976 ; 1982 ; Sugimura and Watanabe, 2000 ; Lauchle and Brungart, 2000). Earlier works mostly focused on identifying the dominant noise generation mechanism for the centrifugal fan and suppressing the generated noise (Neise, 1976 ; 1982). Neise summarized the efforts to reduce the blade passage tone by improving the geometry of the impeller and the cut-off (Neise, 1976 ; 1982). They inclined the cut-off and the trailing edge of impeller and increased the radius of cut-off. Sugimura and Watanabe found that the overall noise level abruptly increased in their experiments, and discussed the resonance phenomena at the impeller flow passage (Sugimura and Watanabe, 2000). They suggested useful formula, which predicts the resonant rotating speed, and showed that holes at the impeller blades could reduce the noise from the acoustic resonance at the flow passage. Lauchle and Brungart studied the unevenly pitched impeller configuration to reduce the tonal noise (Lauchle and Brungart, 2000). But, this method did not give any clue for dramatic noise level reduction while it improves the sound quality. Using two dimensional vortex element method for the simulation of unsteady flow and acoustic analogy for acoustic prediction, numerical study was performed by one of authors (Jeon et al., 2003), where the tonal noise was successfully predicted while some discrepancy between the experiment and numerical simulation was found for the broadband noise. Jeon et al., also found that the tonal noise results from the interaction between the rotating impeller and the stationary diffuser.

In this paper, we identify the acoustic characteristics of the centrifugal fan and suggest some methods to reduce the tonal noise. The suggested methods, which are the unevenly pitched impeller and diffuser, and tapered impeller, show that the strong tonal sound can be significantly reduced. The tapered impeller suppress the aeroacoustic noise source and, therefore, reduces the radiated noise by about 4dB(A).

2. Experiments

2.1 Vacuum cleaner

The vacuum cleaner used in this study does not have a dust bag. The cyclone system is applied for dust collection, as shown Fig. 1. The dusty air sucked from outside goes into the dust tank. In the dust tank, the dust is collected at the bottom dust collector due to the centrifugal and gravitational forces, and then the air is cleaned. The cleaned air goes to the centrifugal fan and motor. Finally the air is discharged to the environment. Conventional vacuum cleaner with the dust bag radiates lower noise because the dust bag, which is located in front of the impeller, absorbs and reflects acoustic pressure. But the noise in this case propagates through the flow path and radia-

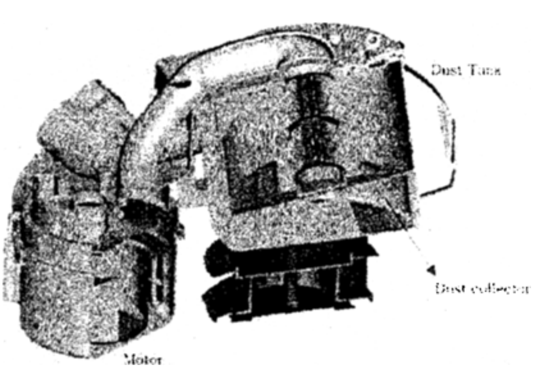


Fig. 1 The flow path of present vacuum cleaner

tes to the environment without any absorption and attenuation. Therefore, low-noise fan design becomes more important for the vacuum cleaner without the dust bag.

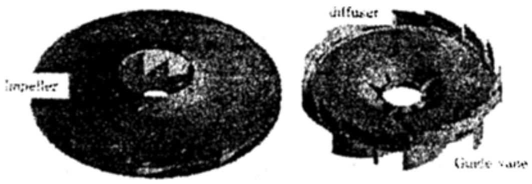
2.2 Fan-Motor assembly

The fan has a rotating impeller and a stationary diffuser, which have 9 and 16 blades, respectively. The impeller is centrifugal and backward type. The outer diameter of the impeller is 0.095(m) and the inner diameter of the diffuser is 0.0965 (m). The outlet blade angle with respect to the circumferential direction is 25° and the position of the maximum camber is 45% chord length from the leading edge of the blade. The inlet and outlet angles of the diffuser vane are 4° and 14.5°, respectively. Schematic of the impeller and diffuser are shown in Fig. 2.

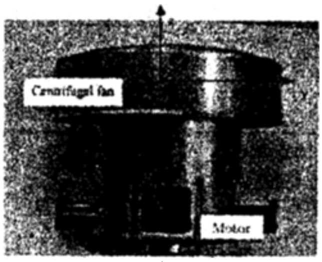
Sound generated from the vacuum cleaner is mostly aeroacoustic as calculated by Jeon et al. (2003). Generated sound is closely related to the high rotating speed of the impeller. The small clearance between the impeller tip and diffuser vane, which is an important design parameter for the performance, makes the tonal sound dominant in the centrifugal fan. The clearance between the impeller tip and diffuser vane is about 1.5 mm.

2.3 Measurement

The performance data of the fan-motor assembly was acquired using the test box (Bleier, 1998). The dimension of the box was 0.5×0.5×0.5(m) as shown in Fig. 3. Dimensions of the test box followed DIN-44956. The static pressures and flow rate curves were obtained by changing the orifice diameter from 0 to 40(mm). The static pressures were measured by YOKOGAWA 2655 with 0.05 mmAq resolution. The noise was measured in the anechoic chamber which has the dimensions of 4.8 m(Width) ×3.8 m(Height) × 4.8 m(Length). The cut-off frequency is 100 Hz and the background noise level was 11 dB(A). The sound pressures were measured by B&K 3550 system and 1/2 inch microphone. SPL of the fan-motor assembly was measured in the center of anechoic chamber. Loudness was measured by MTS Sound quality system.



(a) Configuration of the impeller and diffuser



(b) Fan and motor assembly

Fig. 2 Overview of the centrifugal fan of a vacuum cleaner

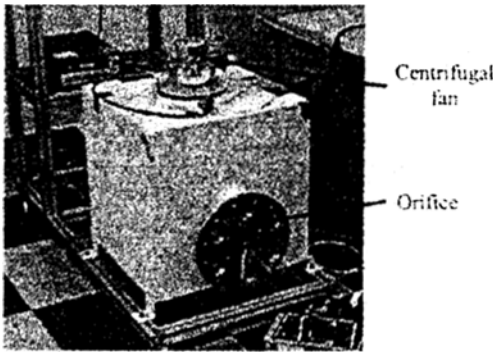


Fig. 3 Fan performance measurement utility

3. Aeroacoustic Characteristics of Fan

The acoustic pressure spectrum of the previous fan-motor assembly at 26760 rpm, is shown in Fig. 4. One microphone was located 1(m) apart from the center of the centrifugal fan unit in the x-direction (side direction at Fig, 4). Another microphone was located 1(m) apart in the y-direction (upside direction at Fig. 4). The peaks at BPF (4014 Hz) and its harmonics (8028, 12042 Hz) are shown clearly. The sound pressure level of the tonal noise was 25 dB(A) higher than that of the broadband noise. From the measured spectrum, it should be noted that the high level peaks are so serious that they significantly affect the

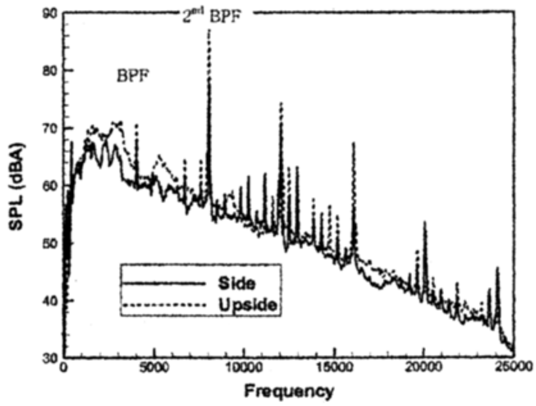


Fig. 4 Measured sound pressure at side and upside direction

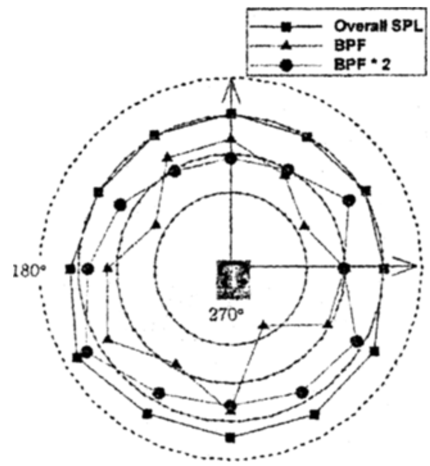


Fig. 5 Directivity pattern of centrifugal fan

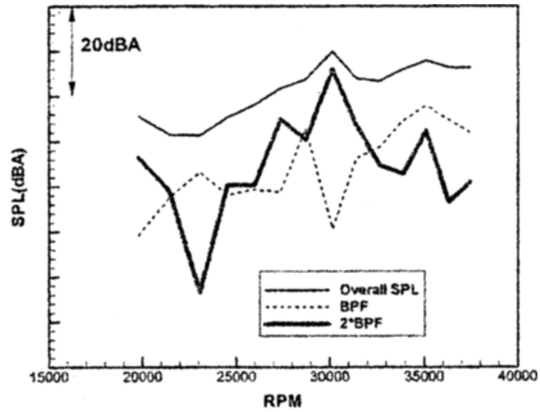


Fig. 6 SPL variation with rotating speed

sound quality of the fan-motor assembly.

The directivity pattern of the radiated noise is shown in Fig. 5, where the tonal noise at BPF and the 2^{nd} BPF dominate the overall SPL and sound quality. Especially, the sound level at the 2^{nd} BPF is higher than the one at the fundamental BPF and governs the overall SPL. The generated sound propagates to the downstream of the fan (270° direction in Fig. 5).

In order to check the dependency of the aeroacoustic pressure on the impeller tip velocity, we measured the overall SPL by changing rotational speed from 19,000 to 38,000 rpm. The result was plotted in Fig. 6. We can see resonance phenomena at around 30,000 rpm. It is well known that high rotating speed generates high level noise. However, the overall sound level at 30,000 rpm increases abnormally. The possibilityof the resonance at the impeller flow passage was carefully reviewed. The resonance frequency was calculated using Sugimura and Watanabe's formula (Sugimura and Watanabe, 2000) in Eq. (1)

$$N=\left(\frac{n+1/2}{s}\right)\left(\frac{30}{Z(l+\delta)}\right)(a_o+(1+M^2)) \quad (1)$$

where, N, l and δ are the rpm, length of the passage and end correction of passage, respectively. Z, M and a_o are the number of blade, Mach number and speed of sound, respectively. The calculated resonance rotating speed of this fan is 29800 rpm and similar to the measured one. It is believed that the abrupt change of overall SPL measured at 30,000 rpm is due to the resonance in the impeller blade flow path. But more study is necessary to understand the abrupt change at the first and second BPF level.

4. Noise Reduction Method

4.1 Unevenly pitched impeller and diffuser

The unevenly pitched impeller and diffuser, which is one of the well known methods to reduce the strong tonal noise as shown in Fig. 4, was applied in this study. Using the genetic algorithm and random modulation, uneven pitch angles of blades were obtained. The objective function of genetic algorithm is the unbalance force of the impeller. The magnitude of mass unbalance was very small and nearly the same with the evenly pitched impeller. Each pitch angles between two blades is illustrated in Fig. 7. As shown in Fig. 8, the performance remains similar to the evenly pitched impeller. However, the SPL is changed as shown in Table 1. The SPLs at peak frequencies are reduced by about 1~4 dB(A). The SPL at the 2nd BPF peak is reduced by about 3.9 dB(A), but at BPF peak it remains almost same. Unevenly pitched impeller makes new small tonal peaks between BPF and its harmonic frequencies as shown in Fig. 9. This phenomenon was also shown in Hayashi et al. (1996). But they used the sinusoidal modulation. Therefore, we believe that the random modulation method used in this paper does not affect the noise level.

The unevenly pitched diffuser does not affect the acoustic pressure spectrum significantly. The uneven pitch angles of the diffuser are also calculated using genetic algorithm. It reduces SPLs at BPF and its harmonic frequencies by small levels as shown in Fig. 11.

Table 1 SPL of peak frequencies for original and uneven pitched impellers

Frequency	Original Impeller	Uneven pitched Impeller
BPF	69.7 dB(A)	69.5 dB(A)
2nd BPF	84.7 dB(A)	80.8 dB(A)
3rd BPF	72.2 dB(A)	71.9 dB(A)

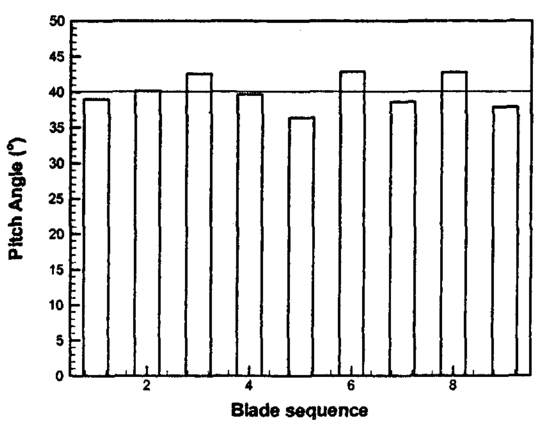


Fig. 7 Uneven pitch angle variation of impeller blade

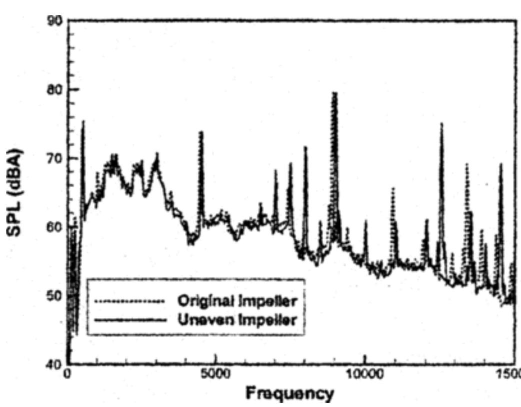


Fig. 9 Measured SPL for original and uneven pitched impellers

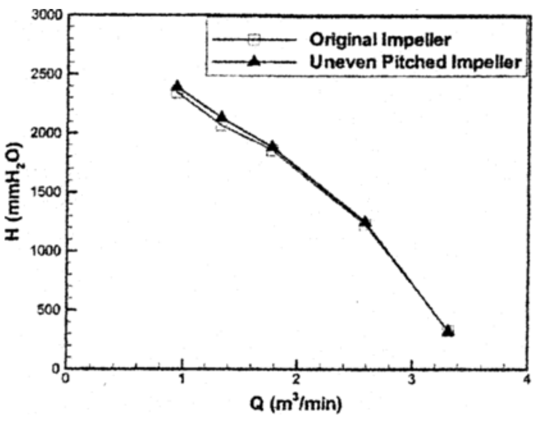


Fig. 8 Performance for original and uneven pitched impeller

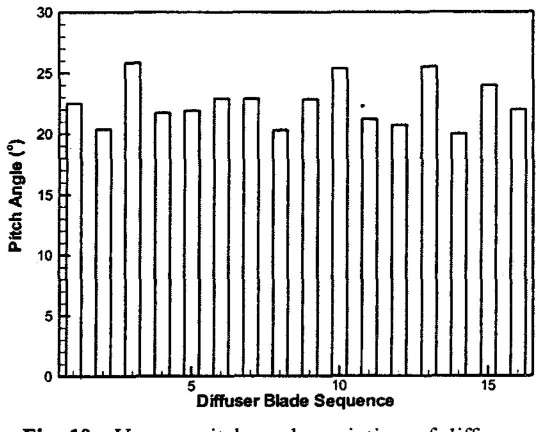


Fig. 10 Uenven pitch angle variation of diffuser blade

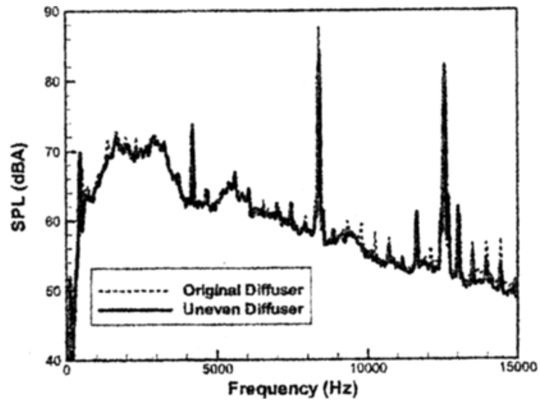


Fig. 11 Measured SPL for original and uneven pitched diffusers. (SPLs at BPF, 2nd and 3rd harmonic frequencies of original diffuser and uneven diffuser are 70.2, 87.3, 81.8, 73.3, 83.7, 81.2 dB(A), respectively)

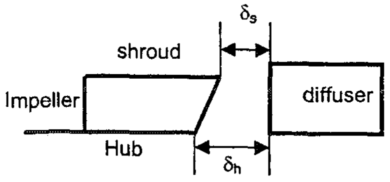
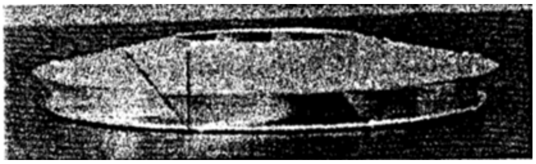


Fig. 12 Schematic diagram of new tapered impeller

(a) Top view of the designed impeller



(b) Side view of the designed impeller

Fig. 13 New designed impeller for low noise centrifugal fan

4.2 Application of tapered impeller

As mentioned earlier, the tonal noise generated by the strong interaction between the impeller tip and the diffuser blade is dominant in the centrifugal fan noise. Especially, SPL at the 2nd BPF is very high, and therefore it is necessary to control this peak in order to reduce the total noise level and improve the sound quality. Therefore, we modified the impeller shape to reduce the strong aerodynamic interaction and suppress the strong peak. For this purpose, the diameter of impeller hub was slightly reduced from δ_s to δ_h as shown in Fig. 12. This makes the gap between the trailing edge of impeller and the leading edge of diffuser increase and therefore, reduce the interaction strength. The value of δ_s is 0.0015 m and δ_h is 0.0065 m. The effect is equivalent to reduction of the diameter of impeller by 2.6%, and the performance becomes worse. The trailing edge of impeller was inclined against to the rotating axis as shown in Fig 13. The inclination of trailing edge makes the pressure fluctuation around the impeller and diffuser blades have some phase shift, consequently reduces the strength of fluctuating pressure. These phase shifts and the reduced strength of pressure fluctuations suppress the strong peak sound at BPF.

The performance and the noise were measured to evaluate the effect of the inclination angle. The performance curves of the two impellers are shown in Fig. 14. Performance of the tapered impeller was slightly lower than that of the original impeller. The overall noise level was reduced by about 3.64 dB(A) while the maximum static efficiency decreased by 4%. In Fig. 15, the measured noise spectra of the original and tapered impellers are compared. The SPLs at peak frequencies are remarkably reduced as shown in Fig. 15. The sound level at BPF and the 2nd BPF are reduced by about 6 dB(A) and 20 dB(A), respectively. Additionally, very sharp and piercing sound was removed from the noise signal. Regarding the effect of the inclination on the sound quality, Zwicker's Loudness is reduced by about 6 phon. The acoustic spectrum of the vacuum cleaner with the tapered impeller is shown in Fig. 16. The overall SPL is reduced by about 3.5 dB(A). So we can reduce the noise level of the

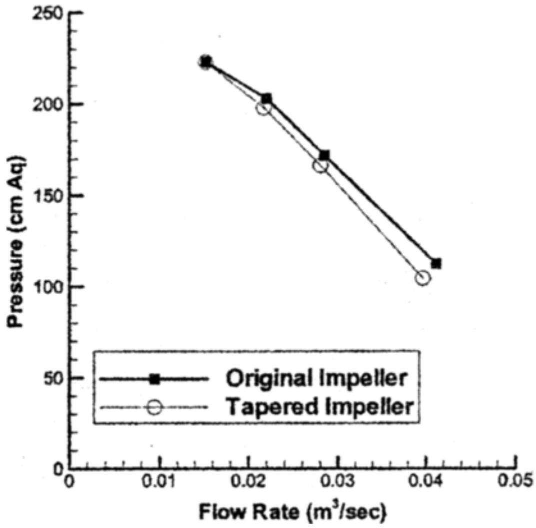


Fig. 14 Performance curves for original and tapered impeller

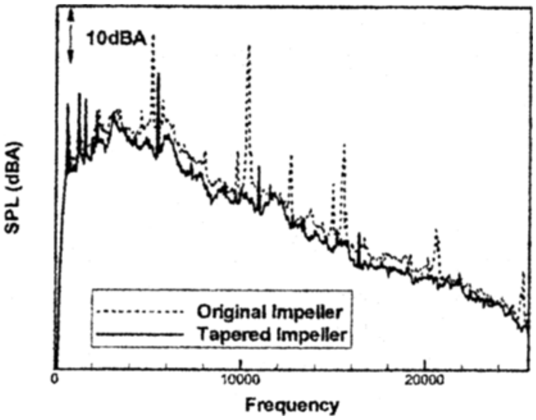


Fig. 15 Comparison of the measured SPL for the original and tapered impeller

vacuum cleaner by using the new tapered impeller, and the piercing tone can be effectively removed. When the diameter of shroud is reduced instead of hub, it was found that the reduction of peak level is smaller than the previous tapered impeller. So, we only consider the impeller of small hub diameter case.

As mentioned previously, SPL was measured in the center of the anechoic chamber without any system. Figure 15 displays the sound pressure spectrum at free condition. Because the primary purpose of this study is the noise reduction of the vacuum cleaner, it is necessary to evaluate

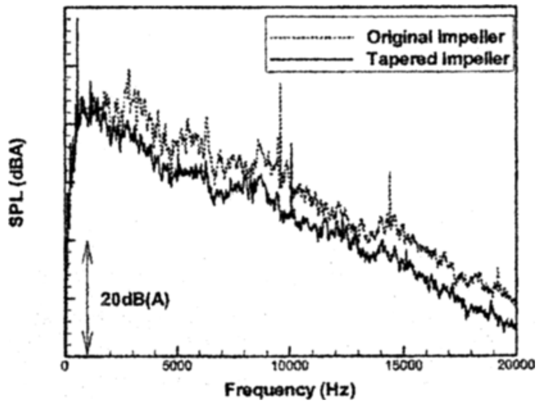


Fig. 16 Comparison of the measured SPL of the vacuum cleaner, which uses original impeller and tapered impeller

the effect of the tapered impeller. When the fan is installed in the vacuum cleaner, the operating condition is different from the free condition because of the system resistance of the vacuum cleaner. Operating point of the impeller is the 3rd point of the performance curve, which is shown in Fig. 14. In order to compare the acoustic behavior of two different fans, the specific noise level of the fan was carefully examined. The specific noise level is defined as Eq. (2)

$$K_f = SPL - 10\log Q - 20\log P_s \qquad (2)$$

Here, SPL is measured noise level (dB(A)), Q means flow rate (m^3/sec), Ps means static pressure of the fan (cm Aq.) and K_f is specific noise level (dB(A)). In order to calculate the specific noise level in Eq. (2), the fan performance curve and the system resistance curve of the vacuum cleaner are necessary. The fan performance curve and the system resistance were obtained using the test box as shown in Fig. 3. The system resistance of the vacuum cleaner in this study satisfies the following formula:

$$\Delta P_s = K_s \times Q^2 = 590.5 Q^2 \qquad (3)$$

Here, ΔP_s means the pressure loss of the vacuum cleaner and Q means flow rate (m^3/sec). Because ΔP_s and Ps are the same, the flow rate could be obtained as shown Table 2. We calculated the operating point by using Eq. (3). and specific noise level was obtained using Eq. (2).

Table 2 Comparison of the specific noise level

	Rotating speed rpm	Fow rate m^3/min	Static pressure cmAq	Measured noise level dB(A)	Specific noise level dB(A)
Original fan	31973	1.61	153.5	70.1	42.1
Tapered Impeller	33573	1.49	130.7	66.6	40.3

We found out that the specific noise level of the tapered impeller is 1.7 dB(A) lower than that of the non-tapered one. Therefore, the tapered impeller generates a little more silent sound than the non-tapered impeller when it is implemented in the vacuum cleaner.

5. Conclusion

In order to reduce the noise level of the vacuum cleaner with the cyclone for dust collection, the aeroacoustic characteristics of a centrifugal fan and its noise reduction method were studied in this paper. We found out that the dominant aeroacoustic noise source of the fan is generated from the aerodynamic interaction between the impeller and diffuser. The resonance phenomenon of the impeller flow passage was observed at 30,000 rpm and this agreed with Sugimura and Watanabe's suggestion. In order to reduce the high noise level of tonal sound, unevenly pitched impeller and diffuser were applied. This changed only the sound quality of the centrifugal impeller while the overall noise level remained unchanged. We also designed a new impeller to reduce the high-level peak noise. The trailing edge of the new impeller design was tapered, which caused some phase shifts during interaction between the rotating impeller and the stationary diffuser blades, and therefore decreased fluctuating pressures. The static pressure efficiency of the tapered impeller was slightly lower than that of the non-tapered one, however the overall SPL was reduced by about 4 dB(A). The sound pressure levels at BPF and 2^{nd} BPF were reduced by about 6 dB(A) and 20 dB(A), respectively. The tapered impeller design showed better sound quality. The vacuum cleaner with the tapered impeller was a little more silent than that with the non-tapered one.

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