Study on High Specific Speed Airfoil Fans*

(1st Report, Effects of Tongue Clearance & Mouth Ring Clearance)

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The performance and noise level of the high specific speed airfoil fan are affected by the structural factors of its casing. Of such factors the following two are thought to produce particularly pronounced effects: (1) clearance and shape of the tongue; and (2) clearance and lap of the suction cone and mouth ring.

The work presented here is a study to determine the effects of such structural factors. For experimental purposes two types of tongues were developed: a skew type slanting in the direction parallel to the vane outlet width; and a modified type having a clearance changed in the same direction as above. A series of experiments was performed in single suction type airfoil fans in which these parameters were incorporated. As a result, their effects on the performance, noise level and noise spectrum of the fan were clarified, thus enabling us to obtain useful information for fan design optimization.

1. Introduction

One of the authors made a prototype of airfoil fan when fans of this type were not commercially manufactured in Japan. This fan was found to give better performance and to produce less noise than the conventional turbo fans with plate vanes. As a search of the literature both at home and abroad failed to reveal any report of a study on airfoil fan, one of the authors, Oshida et al. started a systematic study⁽¹⁾⁻⁽⁵⁾ on the airfoil fan in order to obtain technical information for design optimization. The study was undertaken so as to elucidate the influence of the structural factors of mainly the impeller and casing on the performance and noise of the airfoil fan.

The airfoil fan can have a greater impeller outlet width than the turbo fan, because the vane, being wing-shaped in cross section, has greater modulus of section and bending strength than the plate vane. This means that the specific speed of the airfoil fan can be increased accordingly. As a result, the flow rate can be increased, or the size of the fan can be reduced by use of a higher rotating speed. Inspired by these findings obtained in the previous study, the authors investigated the following two of the structural factors of the airfoil fan casing were investigated in order to improve the performance and noise characteristics of

the high specific speed airfoil fan.

The two structural factors were (a) casing tongue clearance and shape and (b) casing suction cone and impeller mouth ring clearance and lap.

Except for the report of B. Eck⁽⁶⁾ concerning the relationship between tongue clearance and noise, little or no work has been done in this specific field. Though this rare study is valuable, the investigator failed to make mention of his experimental conditions and the performance of the fan used. In this study two variations of the tongue were developed for experimental purposes: skew type slanting in the direction parallel to the vane outlet width, as shown in Fig. 3, and *Hengeki* type or clearance changed type shown in Fig. 4.

In this study experiments were carried out on each of these structural factors of the airfoil fan casing, and their influence on the performance and noise characteristics of the high specific speed airfoil fan was clarified.

2. Nomenclature

Q	: flow rate, m ³ /min
Q*	: maximum efficiency capacity, m ³ /min
Q/Q*	: flow rate ratio
P _T	: total pressure, kgf/m ² (mmAq)
P _{Tmax}	: maximum total pressure, kgf/m ² (mmAq)
P _{TO}	: shut-off total pressure, kgf/m ² (mmAq)
L	: shaft power, kW
Lo	: shut-off shaft power, kW
η	: total pressure efficiency, %
η _{max}	: maximum total pressure efficiency, %
SPL	: average of sound pressure levels at points A, B and C, dB

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- $\overline{SPL}_A$: average of noise levels at points A, B and C, dB
 K_s : specific noise level = $\overline{SPL}_A - 10 \log QP^2$
 f_r : blade passing frequency, Hz
 f'_r : center frequency of third octave band including f_r , Hz
 t : clearance, mm
 t/D_2 : tongue clearance ratio (where, D_2 is outside diameter of impeller in mm)
 r : tongue radius, mm
 e : tongue protrusion or difference in height between tongue and impeller circumference, mm
 ϕ : angle formed by the straight line connecting the ends of adjacent blades to the axis of rotation, deg. (Fig. 3)
 θ : angle formed by the straight line axis of rotation to the ends of tongues corresponding to the impeller back plate and side plate, deg. (Fig. 3)
 θ/ϕ : rate of skew, %
 t_1 : *Hengeki* tongue clearance on the plane perpendicular to the axis on the side of the side plate, mm
 t_2 : *Hengeki* tongue clearance on the plane perpendicular to the axis on the side of the impeller back plate, mm
 T : radial clearance between suction cone and impeller mouth ring, mm
 S : axial lap of the suction cone and impeller mouth ring, mm

Table 1. Profile of impeller vanes

Type	A	B	C
Outlet angle β_2	42°	38°	34°
Inlet angle β_1	39°	34°	30°
Number of vanes Z	17	16	15
Airfoil profile	Howell C4

Fig.1 Cross section of the fan

Fig.2 Test casing (I)

3. Experimental apparatus and procedure

3.1 Study of the tongue clearance and shape

The test fans were No.4 single suction type airfoil fans having outside and inside impeller diameters of 604 and 440mm and inside and outside impeller widths of 150 and 200mm, as shown in Fig. 1. Three types of impellers, A, B and C, were used. The profile of impeller vanes is shown in Table 1. The casing was 420mm wide, and the dimensions of the spiral were as shown in Fig. 2.

The casing had a wooden tongue and was so designed that only the tongue could be replaced. Tongue clearance t was varied from 6 through 10, 15, 23, 40 to 60mm, tongue radius r from 4 through 12, 20, 26, 32 to 48mm, and tongue protrusion e from 0 through 35 to 70mm. In the study to examine the effect of the tongue shape on fan performance the values of t and r were kept at 23 and 12mm, respectively. Skew ratio θ/ϕ was varied from 0 through 25, 50, 75, 100, 125 to 150% by changing the value of θ in Fig. 3. In the *Hengeki* tongue the value of r was kept at 12mm, while the tongue clearance was changed in the direction parallel to the vane outlet width, as shown in Fig. 4.

Experiments were carried out, maintaining t_2 or *Hengeki* tongue clearance on the side of the impeller back plate at a fixed value of 60mm, while changing t_1 or *Hengeki* tongue clearance on the side of

Fig.3 Test casing (II) Fig.4 Test casing (III)

the side plate from 10 through 26 and 40 to 60mm.

3.2 Measuring methods for noise and performance

Performance test was performed in accordance with JIS-B8330 (1962) "Testing Methods for Fans," and the flow rate and total pressure were determined by the pitot tube method. Measurements of dynamic pressure and total pressure were made with an inclined manometer of Füess, and the shaft power measured with a DC dynamometer. The rotating speed was kept at a fixed value of 1,500 rpm in almost all experiments. The output of a magnetic pickup mounted on one end of the shaft of a dynamometer was fed to a digital tachometer to determine the rotating speed. The flow rate was varied through adjustment of the damper. A disk was used as a damper instead of a conical damper because the latter whistles at a low rate of air flow. This necessitated the use of a duct which was longer than specified by JIS. As the purpose of this study consisted in obtaining technical data for fan design optimization, the noise level was measured at three points, A, B and C, specified by JIS. A condenser microphone was set at point A_0 in front of the suction port, and the frequency of noise was analyzed by a frequency analyzer. Considerable space around the fan was secured so that the sound field might provide a nearly semi-free field, and as much reflection as possible was eliminated. As a result, the attenuation due to the distance from the suction port was above 5 dB in terms of doubling distance. The following is a partial list of instruments used in noise measurement.

Precision Sound Level Meter, #2203, Brüel-Kjaer
Audio Frequency Spectrometer, #2112, Brüel-Kjaer
Frequency Analyzer, #2107, Brüel-Kjaer
Level Recorder, #2305, Brüel-Kjaer
Tape recorder, Nagra III B, Kudelski

Fig.5 Arrangement of test apparatus

Fig.6 Cross-section view of the suction cone and the impeller mouth ring

3.3 Study of the suction cone and impeller mouth ring clearance

The test fan was a No.3 single suction airfoil fan. The outside and inside diameters of its impeller were 455 and 333 mm, and its outlet and inlet widths 110 and 147mm. The number of vanes was 16. The specific speed of the fan was approximately 90 (m^3/s , m, rpm). The suction cone consisted of a body (1) and a tip (2), as shown in Fig. 6, and was designed to permit the replacement of tip (2). The impeller mouth ring and tip (2) of the suction cone were machined to an accurate finish, as it was impossible to obtain as accurate an impeller mouth ring clearance and an axial lap as required by the ordinary fan manufacturing method. Large fans of No.4 class were used in experiment (a), whereas fans of No.3 class were made for the purpose of this experiment because of the restrictions imposed on machining.

When studying the mouth ring clearance in this experiment, axial lap S of the suction cone and impeller was maintained at a fixed value of 4mm, while T was given six different values in the range of 1 to 10mm. When studying the axial lap, T was maintained at a fixed value of 2mm, while S was given five different values within the range of 8 to -16mm.

The explanation of the measuring methods for noise and performance employed in this experiment will be omitted here, for they did not differ from the aforementioned ones except that the flow rate was determined from the differential pressure of orifice, varying it by changing the orifice opening ratio (refer to Fig.7).

4. Experimental results and discussion

4.1 Influence of the tongue clearance

The performance curve of an airfoil fan which was plotted taking t as a parameter with $e = 70mm$ and $r = 12mm$ is shown in Fig. 8. The influence of the tongue clearance on total pressure is conspicuous, when Q is in the range of 0 to $110 m^3/min$, and the total pressure increases with an increasing tongue clearance. With $Q > 110 m^3/min$, however, the tongue clearance gives little effect on total pressure. The maximum efficiency value becomes largest when the tongue clearance is 40mm, but becomes smallest when the tongue clearance is 10mm. The difference is about 5%. There is such a relationship

Fig.7 Arrangement of test apparatus

between noise and flow rate that noise increases to a peak at the shut-off flow rate but gradually decreases with an increasing flow rate and then again begins to increase in a slow curve. A close study of the noise curve reveals that the flow rate producing the minimum noise is not necessarily the maximum efficiency flow rate and that the noise ($\overline{SPL}$) is lowest at a flow rate higher than Q^* when the tongue clearance is small and at a flow rate lower than Q^* when the tongue clearance is large. The flow rate which produces the minimum noise ($\overline{SPL}_{min}$) has an exponential relationship to the tongue clearance. When t equals 20mm, the flow rate is in agreement with Q^* .

Fig. 9 shows the relationship between $\overline{SPL}$ and tongue clearance at η_{max} in which outlet angle β_2 is used as a parameter. Noise sharply decreases with an increasing tongue clearance, but no more decreases when the tongue clearance is increased to 40mm ($t/D_2 = 6.6\%$) or more. This tendency does not change regardless of outlet angle β_2 , but less noise is produced at a small outlet angle. With $t = 60\text{mm}$ ($t/D_2 = 9.9\%$), there is a difference of about 4 dB between $\beta_2 = 42^\circ$ and $\beta_2 = 34^\circ$. Fig. 10 shows a noise spectrum. A marked influence of the tongue clearance is reflected in this noise spectrum. When $t = 6\text{mm}$, the sound pressure level is high in the whole frequency range, reaching a peak at 400Hz of f_r equal to the blade passing frequency

Fig.8 Performance and noise curves

Fig.9 Sound pressure level for tongue clearances

and its higher harmonics. When t increases to 10mm, the sound pressure level at a frequency lower than f_r is little different from that reached when t is 23mm. The sound pressure level of rotation noise equal to f_r and its higher harmonics is extremely high when $t = 6\text{mm}$, and the noise and its higher harmonics predominate. With an increasing tongue clearance, however, the sound pressure level of f_r and its higher harmonics decreases until it shows no peak at $t \geq 40\text{mm}$, becoming flat in all the frequency range. The sound pressure level of f_r and its higher harmonics predominates for a low value of tongue clearance because of a significant interference by the wake from the vane outlet. The rotation noise due to such interference may be reduced by increasing the tongue clearance, as the wake at the tongue with a large clearance is attenuated.

4.2 Influence of the tongue radius

Fig. 11 shows an example of the performance of an airfoil fan in which the tongue radius is varied, while tongue clearance t and tongue protrusion e are fixed. Tongue radius r produces a marked effect when Q is in the range of 0 to $100\text{m}^3/\text{min}$. The shut-off total pressure increases with an increasing value of r . As a necessary corollary, almost all performance curves fall when the value of r is large, for instance 32 or 48mm ($r/D_2 = 5.3, 8.0\%$) as in the case cited. In the

Fig.10 Noise spectrum for tongue clearances

Fig.11 Performance and noise curves (Impeller B)

neighborhood of Q^* , however, the shut-off total pressure is nearly constant regardless of the value of r , and the value of $\eta_{\max}$ also reflects little influence of r . As for noise, on the other hand, the flow rate which produces the minimum noise ($\overline{SPL}$) does not depend on r but nearly equals Q^* . The peak of the $\overline{SPL}$ curve in which the total pressure forms its rising portion disappears, when the value of r increases. A reasonable explanation of this phenomenon may be that, as the total pressure curve falls and becomes stabilized with increasing value of r , an unsteady phenomenon of the impeller at a low flow rate like a rotating stall disappears so that low-frequency discrete noise due to the rotating stall is no more produced. It may therefore be inferred from this explanation that the tongue radius also contributes a great deal to the stabilization of the total pressure curve at the low flow rate. The $\overline{SPL}$ curve has no peak in the neighborhood of $Q = 50 \text{ m}^3/\text{min}$, because the low-frequency components of noise are cut as a result of correction of the hearing sense. $\overline{SPL}$ and $\overline{SPL}_A$ of Q^* are hardly affected by the value of r .

Fig. 9 shows the noise levels for two values of r , 12 and 32mm, with $\beta_2 = 38^\circ$. At $t > 20\text{mm}$, the noise level does not differ from one value of r to the other, whereas at $t < 20\text{mm}$ the noise level is so much influenced by the value of r that there is a difference of about 5 dB at $t = 10\text{mm}$. The difference seems to be ascribable to the change which occurs in the condition of interference at the tongue when t is small and r is large. The performance curve concerning tongue protrusion e is omitted here. The noise level does not differ at Q^* , no matter which value is assigned to e , 0, 35 or 70mm, and $\overline{SPL}$ at the flow rate corresponding to the rising portion of the total pressure curve is lower by about 3 dB with $e = 70\text{mm}$, than with the noise level determined for other

Fig.12 Performance and noise curves (Impeller B)

values of e . When revolutions were varied between 600 and 1,875 rpm, the performance and noise were in agreement with the similarity law and sound similarity law, respectively.

4.3 Influence of the skew type tongue

Skew angle θ of the skew type tongue can be easily understood, when its ratio to angle ϕ corresponding to one pitch of vanes is considered. The skew ratio was defined as θ/ϕ in this study.

As has already been mentioned, experiments were conducted, using seven values of θ/ϕ in the range of 0 to 150%. The performance curve ($\beta_2 = 38^\circ$) for impeller B is shown in Fig. 12 (The performance shown in Fig. 12 was determined at the fixed rotating speed of 1,500 rpm).

The total pressure was influenced by the skew ratio in both the low and high ranges of flow rate, but the influence was not significant. The tongue of zero skew ratio was a conventional one of a two dimensional shape. The efficiency of the skew type tongue was highest when θ/ϕ was 150% and was 3% higher, than with the efficiency determined with $\theta/\phi = 100\%$. Its efficiency was 1% higher than that of the conventional tongue.

The noise curves showed much the same tendency for all values of skew ratio, but the noise level differed from one value of skew ratio to another. $\overline{SPL}$ and $\overline{SPL}_A$ showed a difference of 3 dB at the point of $\eta_{\max}$, and the noise level was high with $\theta/\phi = 0$, lowest with $\theta/\phi = 100\%$ and second lowest with $\theta/\phi = 150\%$. The noise level also differed by 2 to 3 dB from one value of skew ratio to another at flow rates other than $\eta_{\max}$. Fig. 13 shows the noise spectrum at $\eta_{\max}$ in which the skew ratio

Fig.13 Noise spectrum for rate of skew

Fig.14 Sound pressure level for rate of skew

serves as a parameter. When attention is focused on f_r' (400 Hz) of the rotation noise, it can be known from Fig. 13 that SPL predominates with $\theta/\phi = 0$ or 50% but does not do so with $\theta/\phi \geq 100\%$. Fig. 14 shows the relationship between rotation noise and skew ratio. As is clear from this figure, SPL decreases with increasing skew ratio, and the noise level differs by as much as 10 dB from zero skew ratio to the value of 150%. The tongue of zero skew ratio, being two dimensional in shape, runs parallel to the impeller outlet so that the mutual interference bears similarity to line contact. When the tongue is skewed, the mutual interference becomes similar to point contact. In the case of the skew tongue, therefore, it may not appear that the effect differs from the value of 50% to the value of 150%. As a matter of fact, however, the tongue is not limitless in size and interference similar to point contact occurs in a certain zone. Interference bears greater similarity to point contact when the skew ratio is 150%. This may explain why noise reduction was more pronounced when the skew ratio was increased from 50% to 100%.

As shown in Fig. 13, SPL is lowest in the frequency range higher than f_r' when θ/ϕ is 100%. It is lower by 7 dB at 4,000 Hz, than with SPL with $\theta/\phi = 0$, but is higher by 3 dB when θ/ϕ is 50% or 150%. Noise due to flow turbulence is considered to be dominant at high frequencies, and increasing noise is thought to result from a uniform flow due to mismatching with the flow at the impeller outlet depending on the skew ratio. These assumptions will explain the effect of the skew ratio on the noise level mentioned above. The flow rate ratio (Q/Q^*) was changed in the skew tongue, but the noise spectrum showed no remarkable change but a similar tendency to the conventional tongue.

4.4 Influence of the Hengeki type tongue

The Hengeki type tongue is a tongue

Fig.15 Performance and noise curves (Impeller B)

having a clearance changed in the direction parallel to the vane outlet width. This modification was made for the purpose of increasing the clearance of the impeller back plate having a high speed of flow from the vane and decreasing the clearance of the impeller side plate having a low speed of flow from the vane.

In the experiments t_2 was maintained at the constant value of 60mm, while t_1 was varied. Fig. 15 shows the performance and noise curves for impeller B. The influence of the Hengeki type tongue on fan performance and noise became appreciable in the low flow rate range. The value of P_{Tmin} decreased with a decreasing t_1 , but efficiency decreased when too small a value was selected for t_1 . At the point of η_{max} , there was little influence on total pressure. Fan efficiency was about 2% higher when t_1 was 26 or 40mm than when t_1 was 60mm (that is, the tongue was parallel to the tip of the vane). Fig. 16 shows the relationship between noise and t_1 . Noise was lowest when $t_1 = 26$ mm, and noise became constant when t_1 was greater than a certain value > 26 mm. Fig. 17

Fig.16 Sound pressure level for the Hengeki tongue clearances

Fig.17 Noise spectrum for the Hengeki tongue clearances

Fig.18 Rotation noise for the Hengeki tongue clearances

shows the noise spectrum in which t_1 is taken as a parameter. As is clear from this spectrum, t_1 produces a remarkable effect on noise. When $t_1 = 26\text{mm}$, SPL is lowest in all the frequency range and rises as t_1 is changed to 40 to 60 to 10mm in order of mention. When the skew type tongue was tested, the skew ratio produced no effect in the low frequency range, whereas the effect of the tongue clearance was significant in the low frequency range as well in the *Hengeki* type tongue. At 4,000 Hz, for instance, noise was lower by 6 dB when t_1 was 26mm than when t_1 was 60mm (that is, the tongue was parallel to the tip of the vane). On the other hand, noise was higher by 7 dB when t_1 was 10mm than when t_1 was 60mm. Fig. 18 shows the relationship between rotation noise ($f_r' = 400\text{ Hz}$) and t_1 . Noise is lowest when $t_1 = 26\text{mm}$. The *Hengeki* type tongue was found to have something to do with the pattern of air flow at the vane outlet, as anticipated. It was inferred from the results of this study that, when the shape of *Hengeki* type tongue matched the air flow at the vane outlet, it helped decrease flow turbulence, thus helping reduce noise in the wide frequency range.

4.5 Influence of the clearance between suction cone and impeller mouth ring

It is empirically known that the performance of centrifugal fans is not a little influenced by T or the radial clearance between suction cone and impeller mouth ring. However, a search of the literature has failed to reveal any systematic study of these factors. Fig. 19 shows the performance and noise characteristics of the centrifugal fan which are determined when the lap is kept at a fixed value, taking T as a parameter. The total pressure is significantly influenced by T at all flow rates. With $Q = 73\text{m}^3/\text{min}$, the total pressure decreases by 19%, when T is increased from 1mm to 3mm, and the total

Fig.19 Performance and noise curves

pressure decreases by as much as 44%, when T is increased from 3mm to 10mm. It is also evident that the efficiency and noise of the fan are also influenced by T . Fig. 20 shows the relationship between the flow rate at $\eta_{\max}$, total pressure and efficiency on one hand and T on the other. Q^* sharply decreases with increasing T , differing by as much as 31.5% from the value of 1mm to the value of 10mm for T . The total pressure decreases when T is in the range of 1 to 5mm, but shows little change when $T > 5\text{mm}$. The efficiency of the fan is also so much affected by T that there is a difference of 10% in fan efficiency between T values of 1 and 10mm. Assuming that Q is a fixed value of $73\text{m}^3/\text{min}$, the fan efficiency differs by 21% between T values of 1 and 10mm. Noise, on the other hand, is influenced by T at all flow rates, as shown in Fig. 19. $\overline{\text{SPL}}_A$ and K_s increase with increasing T . When $Q = 73\text{m}^3/\text{min}$, $\overline{\text{SPL}}_A$ differs by about 3dB between T values of 1 and 10mm, and K_s by 9 dB. At $\eta_{\max}$, however, $\overline{\text{SPL}}_A$ is

Fig.20 Performance for impeller mouth ring clearances

Fig.21 Noise for impeller mouth ring clearances

Fig.22 Noise spectrum for impeller mouth ring clearances

constant regardless of T , as shown in Fig. 21, whereas K_s increases in proportion to T . K_s differs by 4.5 dB between T values of 1 and 10mm. Fig. 22 shows the noise spectrum at $\eta_{\max}$ in which T is taken as a parameter. The noise spectrum shows much the same tendency in all the frequency range regardless of T , and the difference in sound pressure level between center frequencies of the third octave bands is within ± 3 dB. The noise spectrum shows that there is predominant rotation noise at blade passing frequency $f_r' = 400$ Hz.

Fig.23 Performance and noise curves

Fig.24 Performance for mouth ring lap

Fig.25 Noise for mouth ring lap

4.6 Influence of the lap of the suction cone and impeller mouth ring

Fig. 23 shows the performance and noise curves which were plotted, taking axial lap S of the suction cone and impeller mouth ring as a parameter. In the case cited, S was changed in the range of 8 to -16mm. Both pressure and efficiency show little change, when $S = 0, 4$ or 8mm. The pressure and efficiency are thought to show little response to the change in S , because the tip of the suction cone which is located at the parallel portion of the mouth ring makes the condition of mouth ring clearance much the same for all values of S . When $S = -9$ or -16mm, both total pressure and efficiency are decreased. With $Q = 66\text{m}^3/\text{min}$, the total pressure decreases by 23% and the efficiency by 8% when S is changed from 0 to -16mm. The shut-off total pressure, however, is hardly influenced by the value of S . Fig. 24 shows the relationship between the flow rate at $\eta_{\max}$, total pressure and efficiency on one hand and lap S on the other. The flow rate, total pressure and efficiency decrease with an increasing $-S$. Fig. 25 shows the relationship between noise at $\eta_{\max}$, and lap. $\overline{\text{SPL}}_A$ is constant regardless of S , and K_s shows no remarkable change but decreases when S is in the neighborhood of 0.

5. Conclusions

The influence of the clearance and shape of the casing tongue and the clearance and lap of the suction cone and mouth ring on the performance and noise of high specific speed airfoil fans was experimentally clarified, and the following conclusions were drawn from the results of the study.

- (1) When tongue clearance t was in the range of 6 to 60mm ($t/D_2 \approx 1 \sim 10\%$), noise decreased with an increasing t , but the efficiency and total pressure of the fan showed little change.
- (2) The tongue clearance had a significant bearing on the reduction of the noise of blade passing frequency and its higher harmonic rotation noise.
- (3) The conclusions (1) and (2) above remained the same regardless of the values of impeller outlet angle.
- (4) When tongue radius r was in the range of 4 to 48mm ($t/D_2 \approx 0.7 \sim 8\%$), a larger radius produced a slightly favorable effect on noise and efficiency, but the total pressure showed little or no change.
- (5) When tongue radius r was large, the shut-off pressure rose, and the pressure curve showed a similar shape to the falling curve. In other words, a large tongue radius was conducive to the stabilization of pressure in the low flow rate range.
- (6) Tongue protrusion e had little bearing on the performance and noise, so long as it was in the range of 0 to 70mm ($e/D_2 \approx 0 \sim 11.6\%$).

- (7) When the skew ratio of the skew tongue was in the range of 0 to 150%, a large skew ratio produced a more pronounced inhibitory effect on noise, especially rotation noise, and the performance was not adversely affected.
- (8) The *Hengeki* tongue proved to be effective in all the frequency range of noise. The noise spectrum revealed that the *Hengeki* tongue was effective in all the frequency range. The inference was made that the *Hengeki* tongue of a suitable shape ($t_1 = 26\text{mm}$ and $t_2 = 60\text{mm}$ in this study) well matched the flow at the impeller outlet, helping reduce the flow turbulence and thus reducing noise of a wide frequency range.
- (9) The *Hengeki* tongue was more effective than the skew type.
- (10) The clearance between suction cone and impeller mouth ring had a significant bearing on the performance and noise. The smaller the clearance was, the better effect it produced on the performance and noise.
- (11) As suggested by the conclusion (10), it is recommended that special care be taken in manufacture of fans with an accurate clearance, as low-pressure fans are in general practice manufactured in can-making plants.
- (12) The lap of suction cone and impeller mouth ring also had a significant influence on the performance and noise. It appeared that both performance and

noise showed the best response when S was 4mm ($S/D_2 \approx 0.7\%$).

- (13) The noise spectrum showed a similar tendency, no matter how T and S were changed, and the sound pressure level of third octave bands was within ± 3 dB.

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