

Characteristics of Aerodynamic and Noise for Tubular Centrifugal Fan (1 st report: Effects of the Areal ratio of Inlet to Outlet of Impeller and the Geometry of Casing)

by

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An experimental investigation of a centrifugal fan was conducted with attention to the effects of the geometry of the casing on the aerodynamic and fan noise characteristics. A comparison of the fan noise and the aerodynamic characteristics of the tubular casing with those of scroll casing was made. As a result it was concluded that the aerodynamic characteristics of the fan with scroll casing were superior to those with tubular casing. The measured values of overall sound pressure level for both fans roughly same. When the ratio of inlet area to the outlet area of the impeller becomes nearly unity, the fan characteristics improve. For centrifugal fan with tubular casing, the rotating noise is generated by the interaction of impeller blades of the number with stator vanes and distorted inflow.

1. Introduction

Recently, the fan inserted an impeller into a tubular casing instead of a scroll casing is on the market in Japan but few fans of this type are used because of its low efficiency. This fan is called as a tubular centrifugal fan or an axial centrifugal fan in Japan. In Europe and America, it is called as a straight line flow fan. As this fan doesn't have a tongue, the interaction noise between the impeller and the tongue does not cause. Therefore it is expected that the noise of this fan is lower than that of the centrifugal fan.

By this time, a part of author has researched the aerodynamic and noise characteristics of the axial flow fan^{(1), (2)}, the mixed flow fan^{(3), (4)} and the centrifugal fan^{(5), (6)} from both points of view of experiment and theory. And we have discussed a special way to improve fan. The efficiency of tubular centrifugal fan is lower than that of the others, because in the former, pressure loss causes the stream line to bend and the flow impacts on the duct wall. The measurement examples of flow condition and noise characteristics for the tubular centrifugal fan are very few now.

From the standpoint of such a background, in this research, we examined by relating flow condition around impeller, with the effect that areal ratio (ratio of area of inlet to that of outlet of impeller) and the geometry of the casing on aerodynamic and noise characteristics by using three different impellers. We compared characteristics of the tubular centrifugal fan with that of the mixed flow fan on the market that has about the same value of the fan flow rate at maximum efficiency as the former one.

2. List of Main Symbols

A_r : Areal ratio of inlet to outlet of impeller

B : Number of blades

C : Chord length in mm or m

D_0 : Diameter of mouthpiece in mm or m

D_1 : Inner diameter of impeller in mm or m

D_2 : Outer diameter of impeller in mm or m

f : Frequency in Hz

g : Acceleration of gravity in m/s^2

$K_s(A)$: Specific noise level with A weighting characteristic in dB

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$K_s(L)$: Specific noise level with L weighting characteristic in dB

U_t : Circumferential speed at blade tip in m/s

V_a : Axial velocity in m/s

V_r : Radial velocity in m/s

W : Relative velocity in m/s

X : Spanwise distance in mm or m

Y_1 : Span length at inlet of blade in mm or m

Y_2 : Span length at outlet of blade in mm or m

β_2 : Relative flow angle in degree

δ : Deviation angle in degree

η : Combined efficiency of motor and fan

λ : Input power coefficient to electric motor

ν : Hub-tip ratio

ξ : Stagger angle in degree

ρ : Air density in kg/m^3

ϕ : Flow coefficient

ψ : Pressure coefficient

3. Experimental Apparatus and Procedure

Figure 1 shows the schematic diagram of the experimental apparatus. The total length of the duct is about 11 m. An inlet nozzle is installed at the inlet. A conical damper is provided to adjust the flow rate at the exit. A test fan is connected via a diffuser with convergent angle 6° , i.e., a connecting duct to a circular duct of 624 mm I.D., which is equipped with a honeycomb and an orifice flow meter designed in accordance with Japanese Industrial Standards. The tubular duct used in this experiment is shown in figure 2. A driving motor is fixed on the tubular casing. An output of the motor is transmitted to a principal axis through a V belt, which is fixed in a center of duct of 460 mm I.D., which is supported by twelve circular-arch stator vanes. The air inflows from inlet nozzle and it is accelerated to the axial direction by the impeller. After the air flow issued from the outlet of impeller, it is impacted on the duct wall and turned to 90° and flown into the stator vane.

The flow condition at the outlet of blade was measured by using five-pole pitot tube. A circumferential measuring positions are four points of 90° interval in the radius of 247.5 mm and the spanwise measuring points are fourteen points of 10 mm interval. The axial distance between rear shroud of impeller and the leading edge of stator vane is 154 mm.

In this investigation, the comparison of characteristic of

the centrifugal fan with the scroll casing and tubular casing is done. Figure 3 shows the scroll casing which is consisted of parallel wall and side wall, which is extended with a spiral extending angle of 8° . A gap defined by a space between the outer edge of impeller and a tongue of scroll, is 44.4 mm.

The impellers used in this experiment are shown in figure 4(a)~(c). The outer diameter of the impeller is the same as all impellers. (a) is the Clark Y-blade impeller with 10 blades and of 475 mm in diameter, which is made as a main plate(rear shroud) and a side plate(front shroud) become perpendicular to the rotational axis and the ratio of area of inlet to outlet of impeller is 0.29; it is referred to this fan as a No.1 impeller in the following. (b) is a flat plate-blade impeller (No.2 impeller) with 12 blades which is made of sheet metal of 2 mm thickness and whose main plate is perpendicular to the rotational axis but whose side plate has an angle of inclination of 60° to the rotational axis, and whose areal ratio is 0.47. (c) is referred to as No.3 impeller which is the same as the No.2 impeller in number of blades except that the areal ratio is 0.41 and the side plate has an angle of inclination of 75° to the rotational axis. These impellers are also listed in Table 1.

All the impellers in Table 1 are driven by a 2.2 kW 4-pole induction motor at about 1800 rpm. The rotational speed at blade tip is about 43.7 m/s.

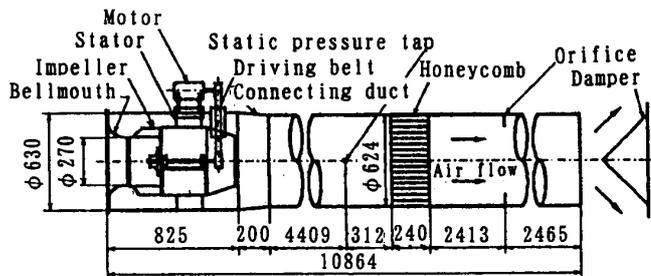


Fig. 1 Schematic diagram of experimental apparatus.

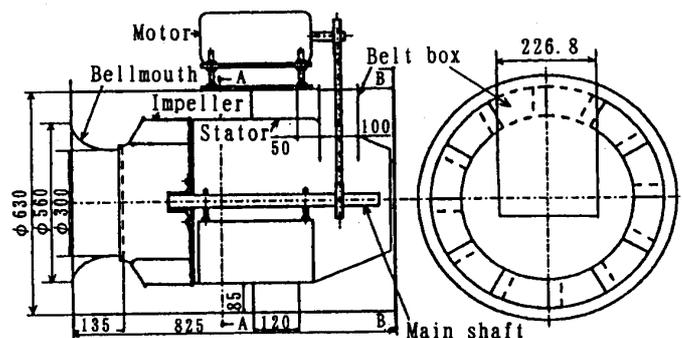


Fig. 2 Tubular casing

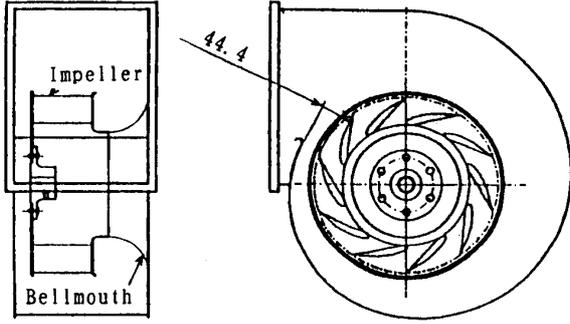
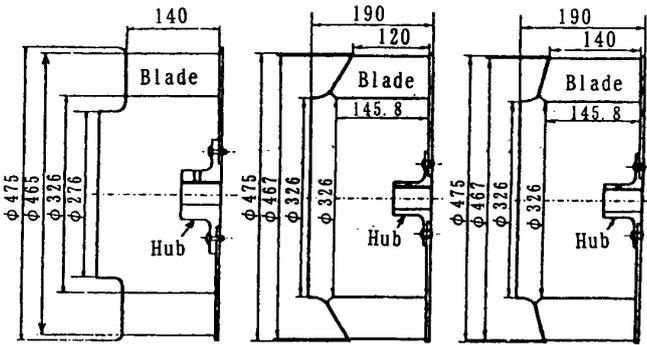


Fig. 3 Volute casing.



(a) No. 1 Impeller (b) No. 2 Impeller (c) No. 3 Impeller

Fig. 4 Schematic diagram of impeller.

4. Experimental Results and Discussion

4.1 Aerodynamic characteristics

Figure 5 shows a characteristic curves of centrifugal fan with scroll casing. In the figure, ψ , ϕ , λ and η are the pressure coefficient, the flow coefficient, the input power coefficient to the electric motor and the combined efficiency of motor and fan, respectively. They are expressed as follows.

$$\begin{aligned} \psi &= 2P_T / (\rho U_i^2), \phi = 4Q / (\pi D_2^2 U_i) \\ \lambda &= 8L / (\pi \rho D_2^2 U_i^3), \eta = \psi \phi / \lambda \end{aligned} \quad (1)$$

Where P_T is the total pressure in Pa, ρ is the air density in kg/m³, U_i is the tip speed of impeller in m/s, Q is the flow rate in m³/s, D_2 is the diameter of impeller in m, L is the input power to the electric motor in W.

In the figure 5, the broken lines, the dotted lines, the solid lines indicate the data corresponding respectively to the No.1 impeller, the No.2 impeller and the No.3 impeller. It can be seen in this figure that the maximum flow rate, Q_{max} of No.1 impeller is lower than that of others and the fan efficiency of the No.1 impeller is lower than that of other impellers by 10 %. This decreases in Q_{max} and the fan

efficiency is partly due to the fact that the No.1 impeller is the smallest among fans in the areal ratio. Therefore, in the No.1 impeller, the flow separates at the side plate in the inlet at the impeller and the back flow region enlarges. Due to this reason, in the flow rate region except below $\phi = 0.06$, the pressure coefficient is the lowest among fans.

In the other hand, the areal ratio of No.2 impeller is the largest among three fans. The maximum fan efficiency $\eta_{max} = 68.7\%$ of No.2 impeller is the highest among three fans. This increases in Q_{max} and η_{max} is due to the fact that the areal ratio can be enlarged and inclined to the side plate largely.

An inlet area of No.3 impeller is the same as No.2 impeller and an outlet area of that is the same as No.1 impeller. Therefore the areal ratio is 0.406. The Q_{max} of No.3 impeller is lower than that of No.2 impeller by 9 % but higher than that of No.1 fan by 70 %. The η_{max} of No.3

Table. 1 Main dimensions of the impellers.

Impeller	No.1	No.2	No.3
Blade section	Airfoil	Flat plate	Flat plate
B	10	12	12
D ₁ mm	326	326	326
D ₂ mm	465	467	467
D ₀ mm	276	326	326
Y ₁ mm	140	145.8	145.8
Y ₂ mm	140	120	140
C	138	140.8	140.8
F	43	44.6	44.6
Ar	0.293	0.474	0.406

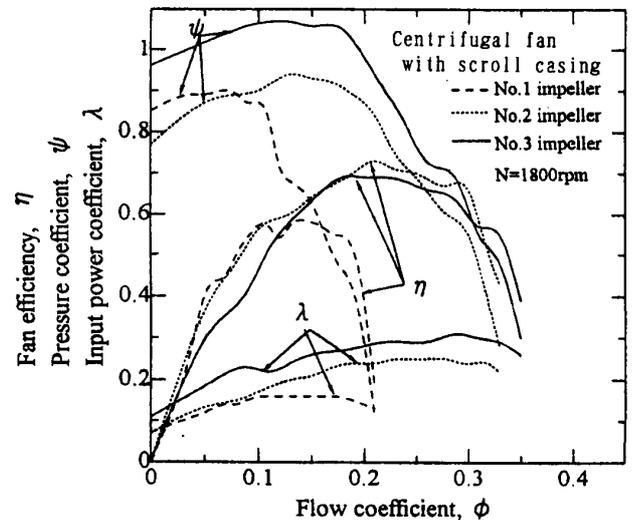


Fig. 5 Characteristic curves. (Centrifugal fan)

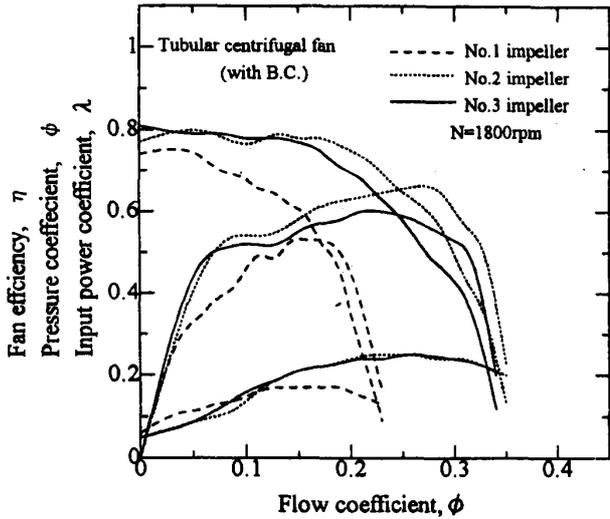


Fig. 6 Characteristic curves.
(Tubular centrifugal fan)

fan is lower than that No.2 fan by 1% but higher than that of No.1 fan by 15%. These data mean that the areal ratio of the impeller must be close to unity.

The aerodynamic characteristics of a centrifugal fan with tubular casing (tubular centrifugal fan) is shown in figure 6. The maximum flow rate increases in order of No.1, No.3, No.2 impeller but these values are lower than that of centrifugal fan with scroll casing shown in figure 6. Moreover the maximum efficiency of the tubular fan is lower than that of centrifugal fan with scroll casing except No.1 impeller by about 10%. It is also true of the pressure coefficient. This is because, in the No.1 impeller, the major part of the pressure loss is generated due to the back flow and flow separation at the inlet of impeller. In the No.2 and No.3 impeller, the pressure loss is caused by the air flow impact on duct wall and the stream lines 90° turn. From the results which we compared the tubular centrifugal fan with the mixed flow fan which has the same maximum flow rate as the tubular centrifugal fan, the latter is superior to the former by 15 % in efficiency and is inferior to the former by 30 % in the total pressure.

4.2 Flow Condition

Figure 7 shows a difference of a mean total pressure caused by three fans at the radius longer than the outlet of impeller by 10 mm. In the mean value of total pressure over the span, No.1 fan is the lowest among three fans. The primary cause is that the areal ratio is so small that the separated flow and the back flow cause at the front shroud (See to figure 8). There is no difference between No.2 and

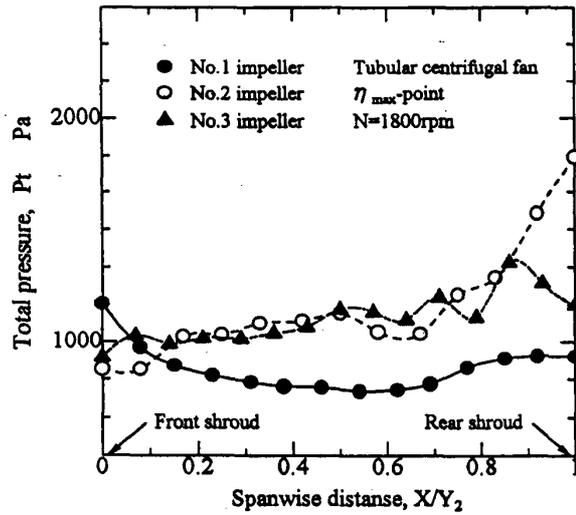


Fig. 7 Spanwise distributions of total pressure.
(Tubular centrifugal fan)

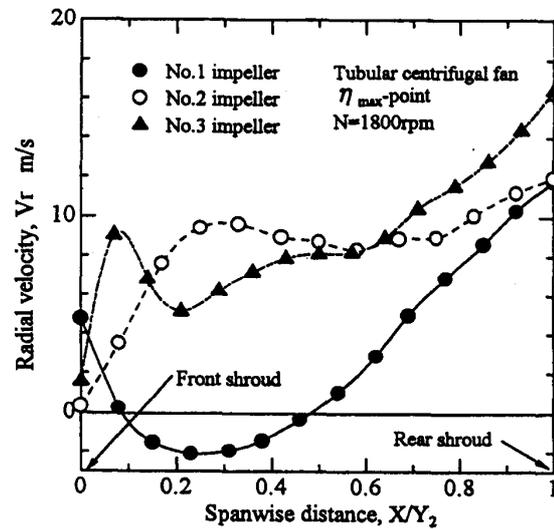


Fig. 8 Spanwise distributions of radial velocity.
(Tubular centrifugal fan)

No.3 fan. The total pressure of these two fans is higher than that of No.1 fan over the span.

Figure 8 shows the effects of impellers on the spanwise distribution of a radial velocity. In the No.1 fan, the air flows downward in the spanwise distance $X/Y_2=0.1\sim 0.4$, but upward in that distance longer than 0.5. On the other hand, in the No.2 and No.3 fan whose areal ratios are larger than that of No.1 fan, the downward flow does not cause. The areal ratio, A , has great effects on the pressure loss. As mentioned above, the A , would be enlarged more.

Figure 9 shows the spanwise distribution of axial velocity at η_{max} -point. From this figure, it is seen that the back flow is generated over 60 % of the span in No.1 fan with the small inlet area. In No.2 and No.3 fan, whose inlet areas of the impellers are increased and whose outlet areas of

impellers are decreased, it is seen that the back flow is not generated. It is due to the fact in the fan efficiency and pressure coefficient, No.1 fan is the lowest among three fans with scroll casing. In the comparison between No.2 and No.3 fan, No.2 fan is higher than No.3 fan in the mean value of the axial velocities over the span and flatter in the distribution of axial velocities.

Figure 10 shows the spanwise distribution of a relative velocity for three tubular centrifugal fans. Take into consideration that the noise generated by the fan is in proportion to six power of the relative velocity, As mentioned above, we can say that the relative velocity is a very important factor. No.1 fan is the lowest among fans in the region of $X/Y_2 = 0 \sim 0.2$ but in the other regions, there are no differences among three fans.

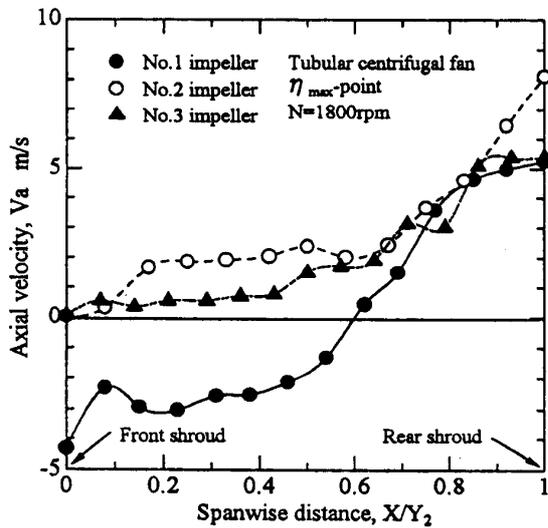


Fig. 9 Spanwise distributions of axial velocity. (Tubular centrifugal fan)

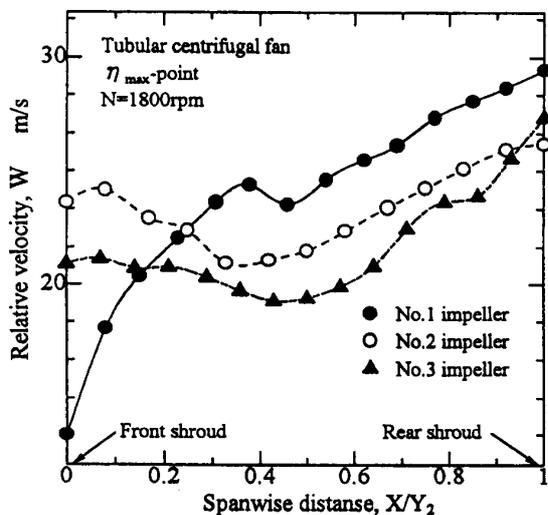


Fig. 10 Spanwise distributions of relative velocity. (Tubular centrifugal fan)

Figure 11 shows a deviation angle, δ used as a factor that shows the goodness of the stream. The smaller δ is, the smoother the stream flows along the blade. From this figure, the δ becomes smaller in order of No.1, No.2 and No.3 fan in the spanwise distance, $X/Y_2 = 0.4 \sim 1.0$. The δ of the No.1 fan is the largest among three fans over most regions. The fan efficiency of No.1 fan is the lowest among three fans with scroll and tubular casing. This is the causes that the air does not flow along blade and the back flow causes at inlet of the impeller.

It is guessed that the air flowed out from outlet of the impeller impacts on the duct wall and turns out at an angle of 90° and the pressure loss is generated by the impact and a bend of stream line. Figure 12 shows the change in total pressure averaged at the measuring section in axial

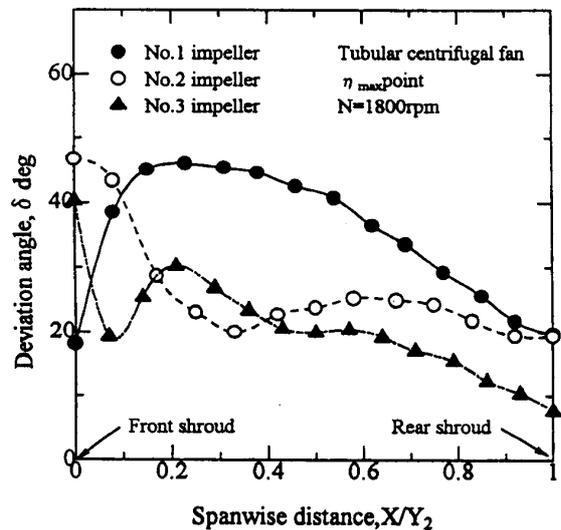


Fig. 11 Spanwise distributions of deviation angle. (Tubular centrifugal fan)

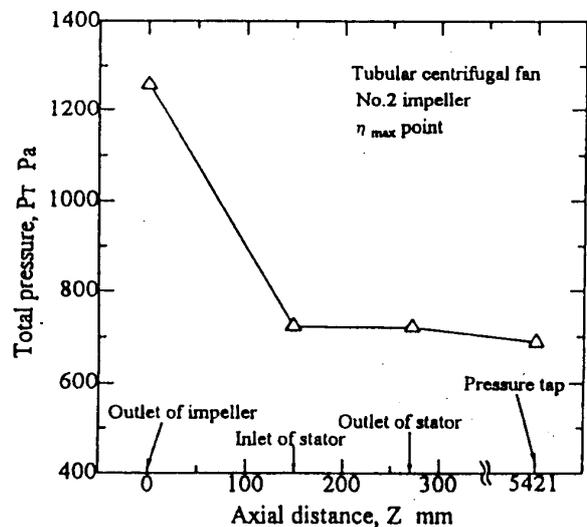


Fig. 12 Total pressure variation after the rotor. (Tubular centrifugal fan)

direction. From this figure, it is found that the total pressure decreases by 530 Pa at the inlet of leading edge of stator, this value corresponds to 40 % of total pressure at outlet of the impeller. From the experimental results, we can say that the pressure loss is very small in the stator vane.

4.3 Noise Characteristics

Figure 13 shows the spectral distribution of fan noise at η_{\max} -point for three centrifugal fan with scroll casing. It is shown in this figure that a rotating noise caused at fundamental frequency ($n=1$) and over tone ($n=2$). We shall examine this in the following. Tyler and Sofrin induced next equation with regard to a interaction noise between impeller and stator⁽⁸⁾

$$m = nB + kV \quad (2)$$

$$\Delta dB/\Delta X = 17.38m[(M_m^*)^2 - M_m^2]^{1/2}/D_2 \quad (3)$$

Where m is the number of lobes, n is the harmonic index, B is the number of blades of impeller, k is the integer, V is the number of vanes of stator or obstacles, ΔdB is the axial decay of sound pressure level, ΔX is the axial distance,

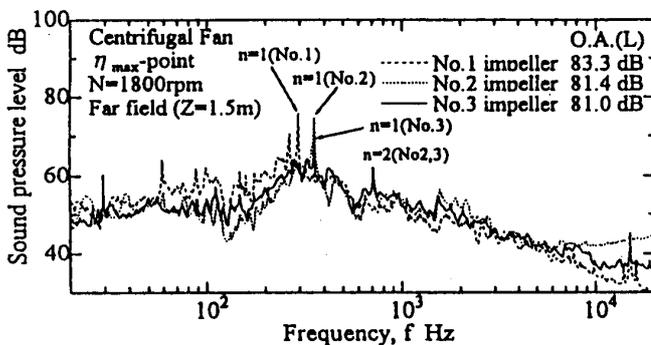


Fig. 13 Spectral distribution of fan noise.
(Centrifugal fan)

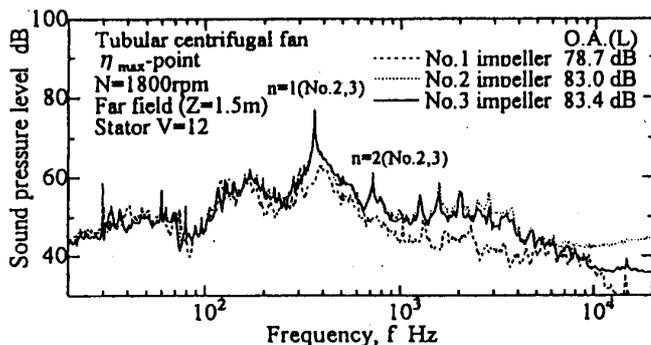


Fig. 14 Spectral distribution of fan noise.
(Tubular centrifugal fan)

M_m^* is the tip critical Mach number, M_m is the Mach number at blade tip, D_2 is the outer diameter of impeller.

The centrifugal fan has a tongue in the scroll casing. The airflow flowed out from the impeller impacts on the tongue and an interaction noise is generated. In this case, the tongue can be regarded as $V=1$. Insert $V=1$ into equation (2), m becomes zero, ΔdB becomes zero. Therefore, these tones do not decay and radiate into the atmosphere from the inlet edge of bellmouth. As mentioned above, in the centrifugal fan with scroll casing, the rotating noise (interaction noise) can be generated and usually it does not decay in duct. This is the cause that the overall noise of the centrifugal fan is high.

In comparison with the turbulent noise among three fans, the sound pressure level of No.1 fan is the highest in the frequency less than 250 Hz. This is the cause of the turbulent flow generated by the back flow at outlet of impeller and separated flow generated at inlet of impeller.

Figure 14 shows the spectral distribution of tubular centrifugal fan noise. The peak of sound pressure level at 360 Hz ($n=1$) is seen for No.2 and No.3 fan but is not seen for No.1 fan. In the former, insert $n=1$, $B=10$, $k=-1$ and $V=12$ into equation (2), the value of m become zero, so that the fundamental tone ($n=1$) does not decay and propagates in duct. On the other hand, insert $n=1$, $B=10$, $k=-1$ and $V=12$ into equation (2), the value of m becomes 2 for No.1 fan, so that the fundamental tone ($n=1$) decays. From a calculation, tone decays to the turbulent noise level at inlet edge of bellmouth. In comparison to overall noise between tubular centrifugal fan and centrifugal fan with scroll casing, the former is higher than the latter by 1-2 dB. Especially, the sound pressure level in 60-120 Hz frequency regions is high. In our guess, the sound is generated, due to the turbulent generated by the impact of flow on the wall or, due to the belt case, which effects the flow along the blade and thickness of the boundary layer on the vane.

Figure 15 shows overall noise, $SPL(L)$ and specific noise level, $K_s(L)$ based on a L characteristics of sound level meter changed with the flow rate for the centrifugal fan with scroll casing. The specific noise level is expressed as a rule by adding sound to the flow rate and the total pressure as shown next equation. It is preferable that this level is lower.

$$K_s = SPL - 10\log_{10}(QP_T^2) + 20 \quad (4)$$

Where SPL is the sound pressure level in dB, Q is the flow rate in m^3/min , P_T is the total pressure in Pa. From the SPL in figure 15, the minimum value of SPL of the No.2 and the No.3 fan is lower than that of the No.1 fan by about 3 dB. On the other hand, in the maximum value and the minimum value of the K_s , No.1 fan is the highest. And No.1 fan has the narrowest in the low specific noise level region which has in the rough the same level as the minimum value. The difference between the No.2 and the No.3 fan is small in both the A and L characteristics.

Figure 16 shows the change in the K_s and the SPL with

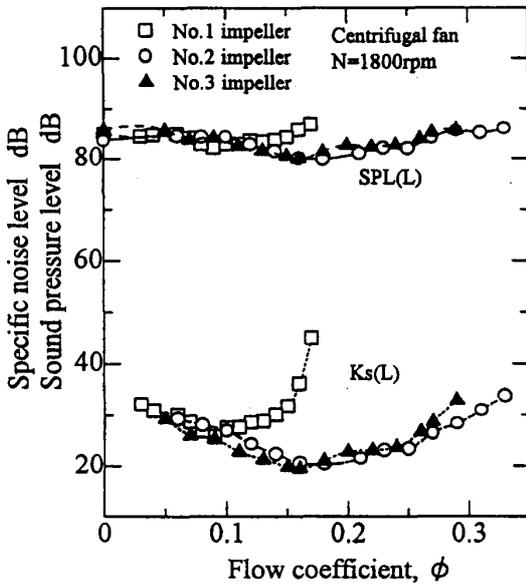


Fig. 15 Change in SPL and K_s with flow rate. (Centrifugal fan)

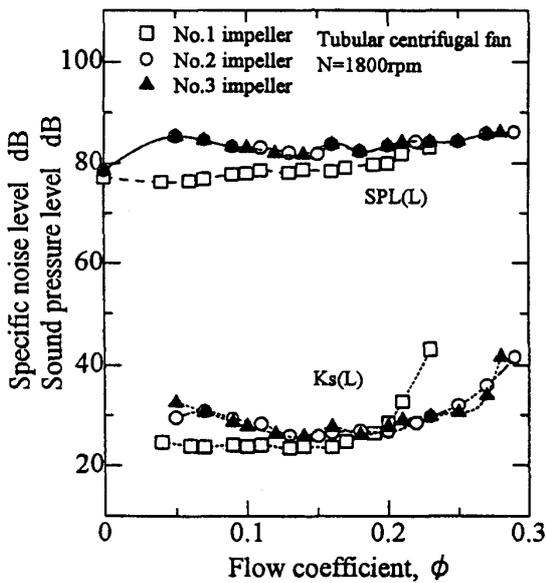


Fig. 16 Change in SPL and K_s with flow rate. (Tubular centrifugal fan)

flow rate for the tubular centrifugal fan. The No.1 fan is the lowest among fans and there is no difference between the No.2 and the No.3 fan for overall noise. Also, the K_s of No.1 fan is the lowest among fans in the flow region where the flow coefficient is less than 0.15.

From the results mentioned above, there is no difference due to the geometry of casing in the No.2 and the No.3 fan but there is difference in the No.1 fan. This is because that in the No.2 and the No.3 fan, the interaction noise causes due to the interaction between impeller and stator vane or the tongue and tone does not decay but in the No.1 fan with tubular casing, noise decays axially.

On the other hand, the K_s of the fan with scroll casing is lower than that of fan with tubular in the No.2 and the No.3 fan because, in the total pressure the former is higher than the latter. As far as examining this experiment, the fan with scroll casing is superior to one with tubular casing in the K_s .

In order to improve the characteristics of tubular centrifugal fan, we have two methods, to reduce the pressure loss due to impact by enlarging duct diameter or to reduce the impact loss and the bent loss by reducing the velocity by enlarging the distance in which air flow impacts on the wall by sloping the blade to the shaft. In the former method, impact loss and bent loss are expected to decrease, but the casing will be larger.

Figure 17 shows the comparison of specific noise level among the mixed flow fan, the No.2 tubular centrifugal fan and the No.2 centrifugal fan. In the comparison at η_{max} -point ($Q = 85m^3/min$), the centrifugal fan is lower than the mixed flow fan and the tubular centrifugal fan by 8 dB in both A and L characteristics. Therefore centrifugal fan is the best among these fans. In comparison of the K_s (L) between the tubular centrifugal fan and the mixed flow fan at the neighborhood η_{max} -point, the former is lower than the latter by 1~2 dB. But in the other regions except these regions, the former is lower than the latter. On the other hand, in the K_s (A), the former is lower than the latter over all flow regions. From the results as mentioned above, by devising method to decrease the pressure loss, it can be expected that the tubular centrifugal fan can play a good roll of the centrifugal fan with scroll casing.

5. conclusions

In this investigation, the effects of tubular casing and scroll casing on the characteristics of aerodynamic and

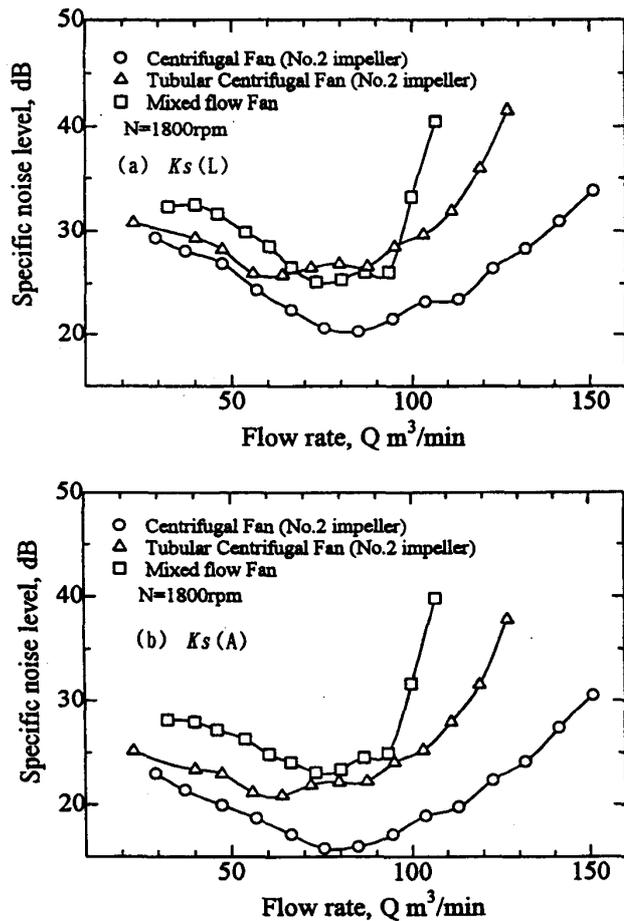


Fig. 17 Comparison of K_s with fans

noise are examined by using three impellers. As a result, the following conclusions are obtained.

- (1) In the case of the fan with scroll casing, the areal ratio of inlet to outlet have an important effect on the characteristics of the aerodynamic. When the ratio is small as No.1 fan, the pressure loss increases due to separating and the back flow at the neighborhood of the front shroud so that the total pressure and the fan efficiency decrease. Maximum flow rate decrease.
- (2) The fall of the efficiency and pressure of the tubular centrifugal fan is caused mainly by the pressure loss due to the impact. The air flowed from impeller impacts on the casing wall and it turns with an angle of 90° and it flows down and the impact explained above causes.
- (3) In the case of the centrifugal fan with scroll casing, the $m=0$ mode discrete frequency noise causes by the interaction between impeller and tongue.
- (4) In the case of the centrifugal fan with tubular casing, when the number of blades and stator vane is the same, the $m=0$ mode discrete frequency noise which never

decreases to the direction of rotating axis of impeller is generated by the interaction between impeller and stator or flow distortion into impeller. These tones do not decay in the duct and radiates into the atmosphere from the inlet nozzle. Therefore, you must pay much attention to the combination of the number of impeller and stator vane and manufacture of duct.

- (5) In comparison of overall noise between the No.1 centrifugal fan with scroll casing and the No.1 tubular centrifugal fan, the former is higher than the latter, because the discrete frequency noise due to interaction between the impeller and the tongue causes in the former.
- (6) As far as this investigation is concerned, the specific noise level of the centrifugal fan with scroll casing is lower than that of the tubular centrifugal fan. On the other hand, the level of the tubular centrifugal fan is a little lower than that of the mixed flow fan.

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