Characteristics of Aerodynamic and Noise for Tubular Centrifugal Fan

(2 nd report: Effects of Belt Case, Inclination of Blade, Size of Casing and Preventive Plate against Reverse Flow)

by

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The effects of the belt case, the inclination of blade, the size of casing and the preventive plate of a tubular centrifugal fan on both fan noise and the aerodynamic characteristics are experimentally investigated by using three impellers. The results are summarized as follows: A considerable amount of a rise of pressure and fan efficiency can be expected by using the inclined blade and taking off belt case. Therefore, the specific noise level of the tubular centrifugal fan decrease considerably. In the ratio of casing diameter to impeller diameter is 1.32, the more the distance between the preventive plate against reverse flow and impeller becomes short, the more the fan efficiency and total pressure become high.

1. Introduction

It is called as a tubular centrifugal fan which uses a centrifugal impeller in a tubular casing in spite of a scroll casing. In the first report, we studied experimentally the effects of scroll casing and tubular casing, and an areal ratio of inlet to outlet of impeller on the aerodynamic and noise characteristics. In the result, we reavealed two points. The first, usually, a centrifugal fan with scroll casing is superior to a fan with tubular casing in the aerodynamic and the noise characteristics. The second, closer to unity, Ar that is the inlet/outlet area ratio of impeller is, higher the maximum efficiency is and lower the specific noise level is.

As a rule, the tubular centrifugal fan adopt special way to transmit the power from electric motor. The way uses a pulley and V belt because it can change a revolution of impeller easily. There are some reasons why the tubular centrifugal fan is lower than usual centrifugal fan with scroll casing in the efficiency and the total pressure. One is that the pressure loss causes when the air flow impacts on the duct wall and other is the belt case for driving belt. It can't be ignored that the belt case of the tubular fan which is on the market take 30% in sectional area of the duct. It is considered that a way to reduce the pressure loss by impact of the air flow and a bend of the stream line is to make incline blade to axial direction. By this way, we can increase the distance that flow reaches duct wall and we can reduce the pressure loss by the impact and by the bend of stream line becase an angle of bent become small. Similarly, by increasing the diameter of tubular casing, we can reduce the pressure loss.

From a point mentioned above, we investigate experimentally that the effects of belt case, angle of the inclination of blade, the dimensions of casing and the preventive plate against reverse flow on the characteristics of aerodynamic and noise.

2. Main Symbols

A_r : Inlet/outlet area ratio of impeller

B : Number of blades

Received on 14. 4. 1999

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C : Chord length m, mm
D_c : Diameter of casing m, mm
D_o : Diameter of mouthpiece m, mm
D_1 : Inner diameter of impeller m, mm
D_2 : Outer diameter of impeller m, mm
f : Frequency Hz
g : Acceration of gravity m/s^2
$K_s(A)$: Specific noise level based on A characteristic of sound level meter dB
$K_s(L)$: Specific noise level based on L characteristic of sound level meter dB
L : Input power of electric motor W
n : harmonic index
N : Rotational frequency rpm, rps
P_T : Total pressure Pa
Q : Flow rate m^3/min, m^3/s
R : Radial distance m, mm
R_C : radius of casing m, mm
$SPL(A)$: Sound pressure level based on A characteristic of sound level meter dB
$SPL(L)$: Sound pressure level based on L characteristic of sound level meter dB
U_t : Circumferential velocity at blade tip m/s
V_a : Axial velocity m/s
V_{r2}: Radial velocity at outlet of impeller m/s
W_2: Relative velocity at outlet of impeller m/s
Y_1 : Span length at inlet of impeller m, mm
Y_2 : Span length at outlet of impeller m, mm
β_2 : Relative flow angle at outlet of impeller deg.
δ : Deviation angle deg.
η : Fan efficiency
λ : Input power coefficient
ν : Hub-tip ratio
ξ : Stagger angle deg.
ρ : Air density kg/m^3
ϕ : Flow coefficient
ψ : Pressure coefficient

3. Experimental apparatus and procedure

The schematic diagram of experimental appatraus is shown in figure 1. The total length of experimental apparatus is about 11 m and a test fan is mounted near the suction end of the duct, where a bellmouth is installed. Downstream of the fan there is a long straight duct, on which an orifice flow meter and a honeycomb which satisfy the JIS standard are mounted, and at the exit of the duct a conical damper is provided to adjust the flow rate. We adjust flow rate by closing and openning the damper. The test fan and discharge duct are connected by a cylindrical diffuser with converging angle of 6°.

Figure 2 shows the schematic diagram of fan. The electric motor for driving impeller is fixed on the tubular casing. The power from motor is transmitted to main shaft of the fan through V belt. The belt case that protects V belt is 267mm in wide and 48mm in thick. The main shaft is located in the center of cyrindrical duct by twelve stator vane. The diameter of this duct is 460mm. To research the influence of belt case, we made a fan that the impeller is mounted motor shaft directly and compared the efficiency of two fans. The flow condition at outlet of impeller is measured in four sections at intervals of 90° four section in the radius larger than impeller radius by 10mm. The spanweise measuring point is fourteen points at the intervals of 10mm.

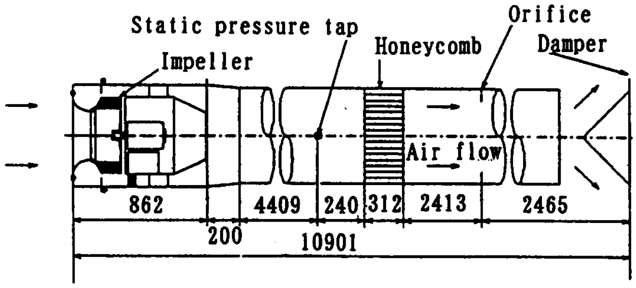
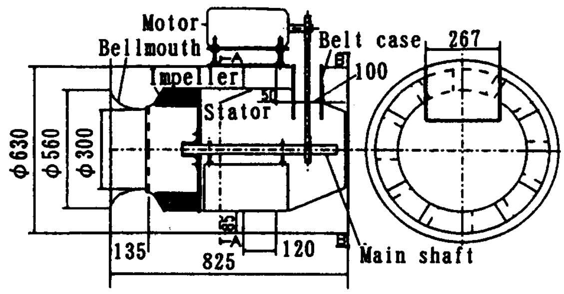
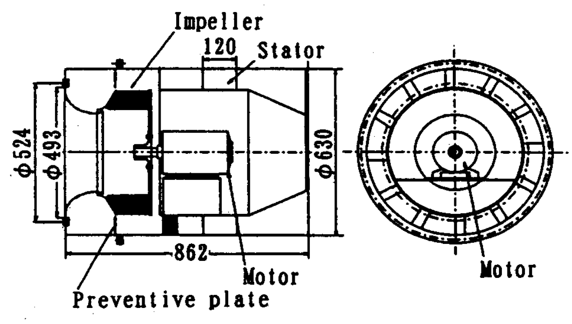


Fig. 1 Schematic diagram of experimental apparatus.



(a) With belt case



(b) Without belt case

Fig. 2 Schematic diagram of test fans.

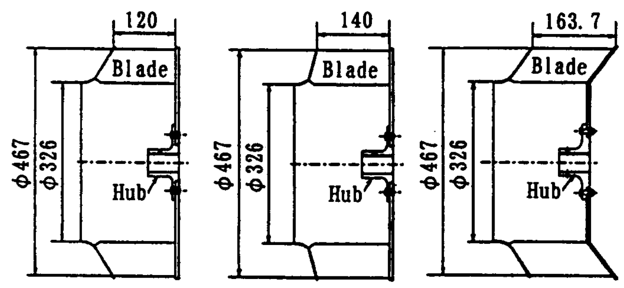
The distance between rear shroud of impeller and leading edge of stator vane is 154mm. We installed the preventive plate in order to avoid back flow at outlet of impeller and made experiment to research that effects of the preventive plate on the characteristics of aerodynamic and noise by changing relative location between the preventive plate and the impeller. The preventive plate is doughnut shape and its inner radius is 10mm larger than outer radius of impeller. It is 4mm thick and made of iron.

Figure 3 shows the schematic diagram of test impellers. The impeller is shown in figure 3 (a), which rear shroud holds vertical to rotating shaft and front shroud holds the angle of 62° to rotating shaft. The impeller is made up of 12 flat plate blade and the area ratio, A_r is 0.47 (called as No. 2 impeller). The impeller is shown in figure 3 (b), which rear shroud holds vertical to rotating shaft and front shroud holds the angle of 75° to rotating shaft. The area ratio, A_r of this impeller is 0.41 (referred No. 3 impeller). The No. 4 impeller is shown in figure 3 (c), the angle of inclination of the front shroud and rear shroud is the same as that of front shroud of the No. 2 impeller. The sectional area at outlet of impeller vertical to stream line is almost same as the No. 3 impeller.

These impellers are driven by 2.2 kW 4 pole induction motor at about 1800rpm. The tip speed of impeller is about 43.7m/s in the experiments. From now on, we call fan with No. 2, No. 3, No. 4 impeller as the No. 2, the No. 3, the No. 4 fan. The main dimensions of impellers are listed in Table 1.

Table 1 Main dimensions of the impellers.

Impeller		No. 2	No. 3	No. 4
B		12	12	12
D_1	mm	326	326	326
D_2	mm	467	467	467
D_0	mm	326	326	326
Y_1	mm	145.8	145.8	145.8
Y_2	mm	120	141	163.7
C	mm	140.8	140.8	140.8
ξ	deg.	44.6	44.6	44.6
A_r		0.474	0.406	0.335



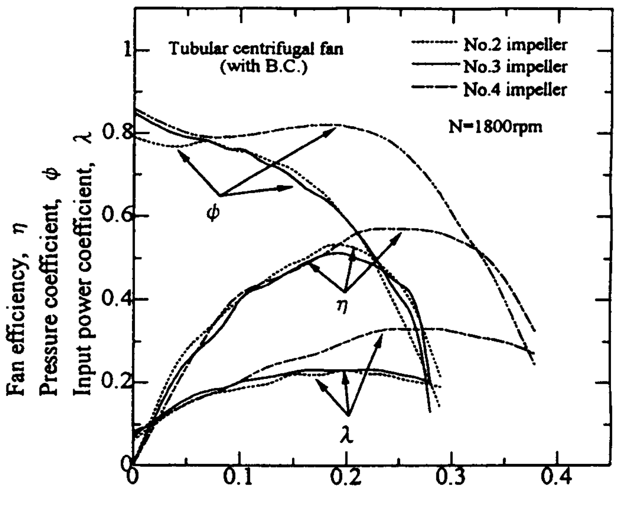
(a) No. 2 Impeller (b) No. 3 Impeller (c) No. 4 Impeller

Fig. 3 Schematic diagram of impeller.

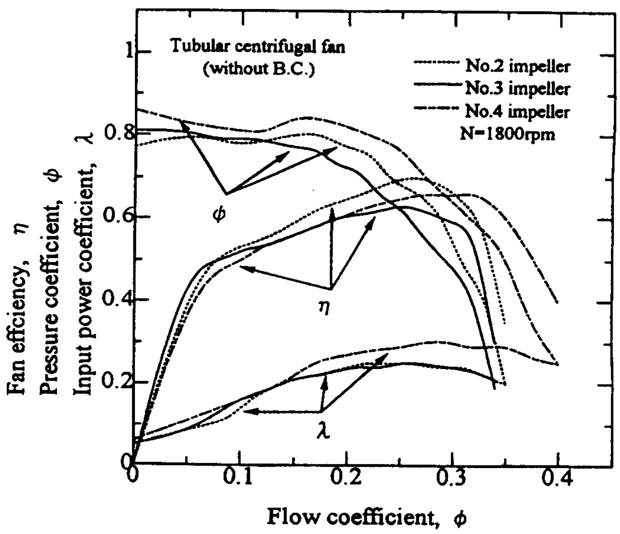
4. Experimental Results and Discussion

4.1 Aerodynamic Characteristics

The effects of existence of the belt case on characteristic curves for the tubular centrifugal fan are shown in figure 4. In figure 4 (a), we show the case with belt case and in figure 4 (b), the case without belt case. In the figure, ϕ, ψ, λ and η are the pressure coefficient, the flow coefficient, the input power coefficient to the electric motor and the combined efficiency of the motor and the fan, and they are expressed as follows.



(a) With belt case



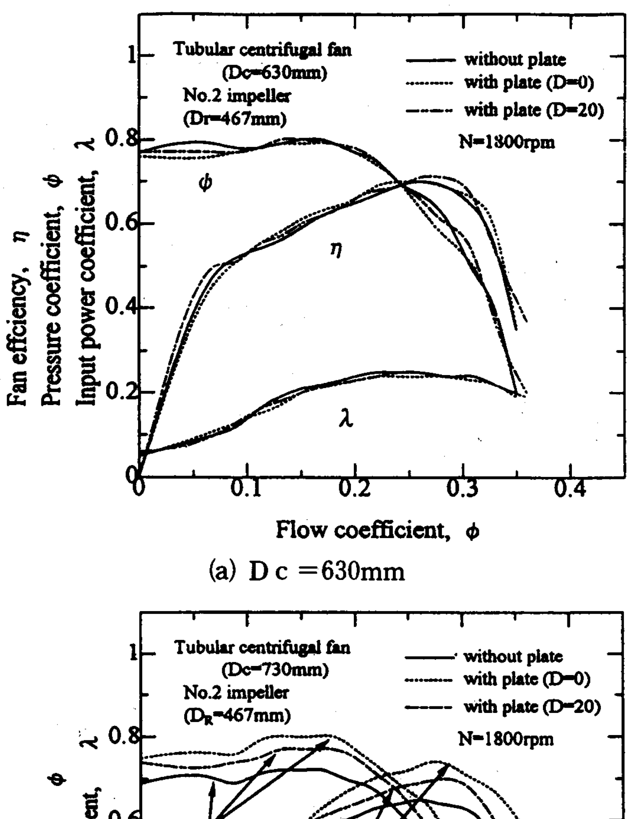
(b) Without belt case

Fig. 4 Effects of belt case on characteristic curves.

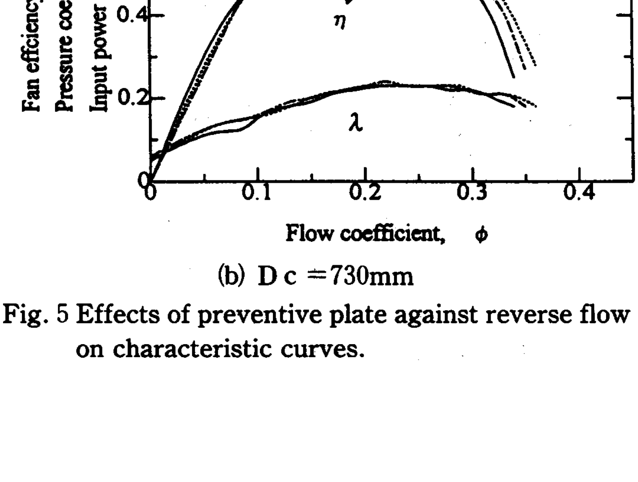
$$\psi = 2P_T/(\rho U_t^2),\ \phi = 4Q/(\pi D_2^2\ U_t)$$
$$\lambda = 8L/(\rho\pi D_2^2\ U_t^3),\ \eta = \phi\psi/\lambda \qquad (1)$$

Where P_T is the total pressure in Pa, ρ is the air density in kg/m^3, U_t is the tip speed in m/s, Q is the flow rate in m^3/s, D_2 is the outer diameter of impeller in m, L is the input power to the electric motor in W.

The dotted line in figure 4 (a) is the experimental result of No. 2 fan, the solid line is that of No. 3 fan, the two dots chain line is that of No. 4 fan. In comparison with the maximum flow rate of No. 2 and No. 3 fan, that of No. 4 fan is the most and the efficiency of No. 4 fan is the highest. There are some reasons for that No. 4 impeller blade has an angle of 60° to rotating axis but others have that of 90° so the distance



(a) D c =630mm



(b) D c =730mm

Fig. 5 Effects of preventive plate against reverse flow on characteristic curves.

that the air flowed out the impeller impacts on the duct wall is longer and the pressure loss due to the impact and the reduction in bend of the stream line is reduced.

The reason why efficiency is low generally may be the pressure loss due to the belt case. We show the characteristics of tubular centrifugal fan without belt case in figure 4 (b). In this figure, it is seen that the maximum efficiency of No. 4 and No. 2 fan is almost same and the rise of efficiency and pressure for the No. 2 and the No. 3 fan is remarkable as compared with the No. 4 fan. This is because the velocity at neighborhood of the duct wall for No. 2 fan is higher than that of No. 4 fan in the radial distribution of velocity, so the pressure loss due to the belt case become greater (refer to figure 6). As compared with figure 4 (a) and 4 (b) the fan efficiency, the maximum flow rate and the pressure for the fan without belt case is higher than that of the fan with belt case in all fans. As mentioned above, it is found that the belt case causes large resistance.

Figure 5 shows the effects of three locations of the preventive plate against back flow on characteristic curves for the diameter of casing, D_c=730mm. In this figure, as compare with the fan which mounted the preventive plate over the leading edge of blade (D=0) and is not installed that, the former is 10%higher in the efficiency, 0.07~0.1 in the pressure coefficient and 0.02 in the maximum flow rate than the latter. Moreover, the pressure coefficient, the maximum flow rate and the maximum efficiency increase according as the preventive plate closes leading edge of blade. This is cause the attachment of preventive plate prevents the back flow at outlet of impeller and the pressure loss reduces.

Next, we discuss the effects of diameter of casing on the fan efficiency and the pressure coefficient by used No. 2 fan. In the fan without preventive plate, the fan with casing of 630mm diameter [dotted line in figure 4 (b)] is somewhat higher than that with casing of 730mm diameter in the maximum flow rate, the maximum efficiency and the pressure. This is main cause the back flow region of the latter is wider than that of the former. The pressure loss is generated by next three sources. The first is due to back flow. The second is due to the bend of stream line. The third is due to impact on duct wall. In the total pressure loss of the second and third source, the latter (D_c=730 mm) is smaller than the former (D_c=630mm) because the velocity at neighborhood of the duct wall is smaller the latter than the former. In the total pressure loss of the three sources, the latter is larger than the the former. From these facts, the pressure loss due to the back flow in the latter is larger than that in the former. Therefore, it is need to attach the preventive plate in the casing of large size. In the casing of 630mm size, when the preventive plate locates over the leading edge of impeller, the total pressure reduces a little due to skin friction of plate. When the preventive plate is separated more than 20mm from the leading edge of impeller, the total pressure, the fan efficiency and the maximum flow rate does not change in spite of the existence of preventive plate.

The velocity was measured in the section at 165mm upstream to examine the effects of belt case on the aerodynamic characteristics. Figure 6 shows the radial distribution of axial velocity at 50mm ahead of the belt case. In this figure, ● mark is the results in No. 2 fan and ○ mark is that in No. 4 fan. From this figure, the velocity at neighborhood of the duct wall is higher the No. 2 fan than the No. 4 fan. Taking into consideration that the pressure loss is proportional to the second power of velocity and as compared with figure 4 (b) and figure 4 (a) in the pressure and the fan efficiency, it is found that the quantity of increase of the No. 2 fan higher than that of the No. 4 fan shall be cause the velocity difference in both fans.

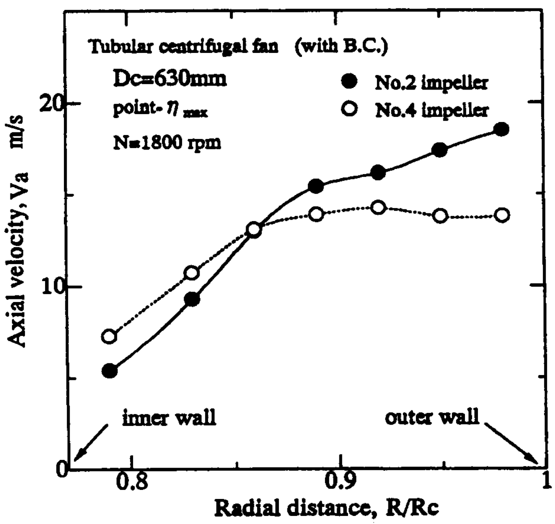


Fig. 6 Spanwise distributions of axial velocity. (Effects of inclination of the blade)

4.2 Spectral distribution of fan noise

Figure 7 shows the spectral distribution of fan noise at η_{max}-point for fans with belt case. The sound pressure level rising in 200 Hz frequency region is seen in the No. 2 and the No. 3 fan but is not seen in the fan without belt case (See figure 8). Therefore, this rising shall be generated by the belt case. The peaks of sound pressure level at $n=1$ and $n=2$ are shown in this figure. These are the interaction noise generated by interaction between the impeller and the stator. The number of lobes of circumferential sound pressure variation, induced by Tyler and Sofrin is shown in next equation[1].

$$m=nB+kV \tag{2}$$

Where m is the number of lobes, n is the harmonic number(n= 1 for fundamental tone), B is the number of blades, k is the index (k=$\cdots -1, 0, 1 \cdots$), V is the number of vanes. When m= 0 in equation(2), interaction noise does not decay in duct and propagate[2][3]. In the No, 2 ~No. 4 fan, the interaction noise

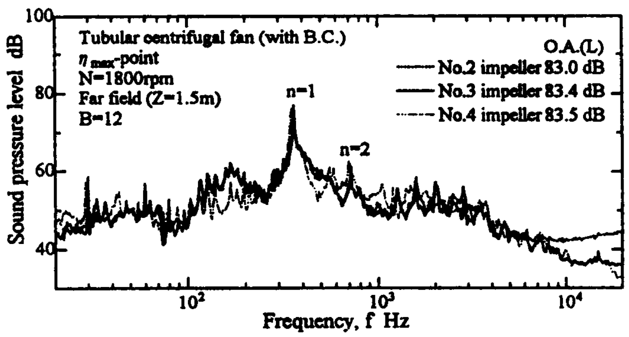


Fig. 7 Spectral distribution of fan noise. (With belt case)

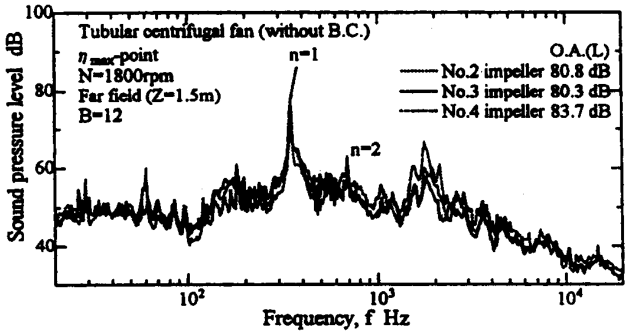


Fig. 8 Spectral distribution of fan noise. (Without belt case)

due to the impeller with 12 blade and stator with 12 vane at the fundamental frequency and the overtone frequency does not decay because m is zero at $n=1$, $k=-1$ for the fundamental frequency and $n=2$ $k=-2$ for the overtone frequency. In the over all noise for three fans, the difference between No. 2 fan and No. 3 fan is a little but the noise of No. 4 fan is lower by 3 dB than that of No. 2 fan and No. 3 fan.

As shown in figure 8, the interaction noise generated by impeller-stator interaction also occur in fans without belt case. The difference between fans with belt case and without belt case is a little. In the fan without belt case, rising of the sound pressure level at 2000 Hz frequency region. This noise results from the vibration of duct wall, it is causes that the intensity of duct wall became weaker due to take off the belt case. The effects of location of the preventive plate on the noise generated at η_{max}−point is shown in figure 9. From overall noise in figure (L characteristic), the noise of fan without the preventive plate is 1.3 dB lower than that of fan with that. This result is due to the difference based on interaction noise. In the turbulent noise, the effects of preventive plate is not seen.

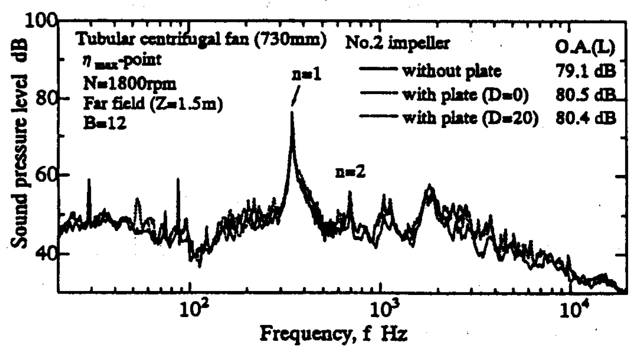


Fig. 9 Spectral distribution of fan noise. (Effects of preventive plate against reverse flow).

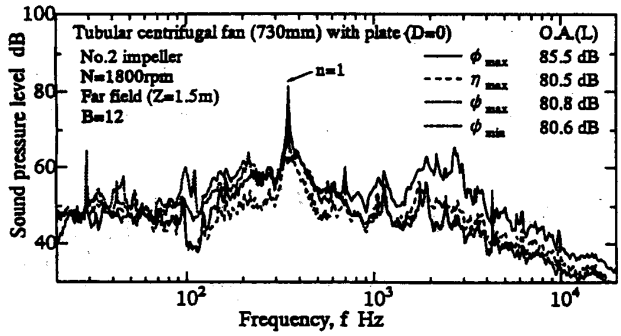
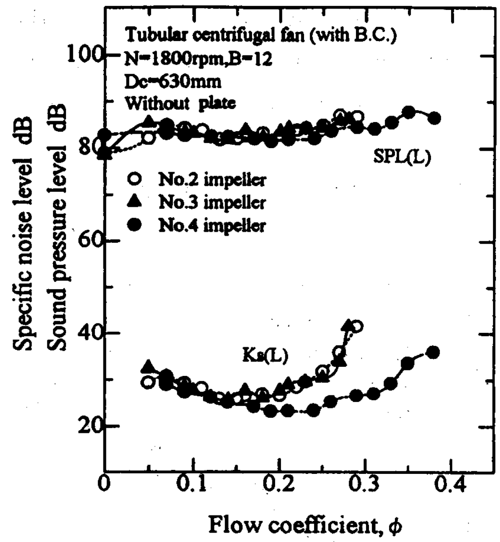
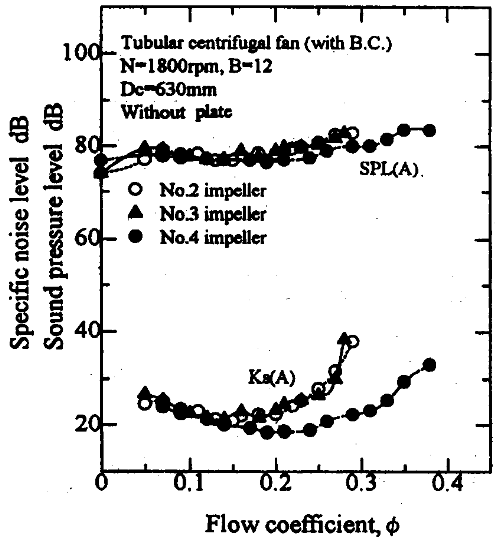


Fig.10 Spectral distribution of fan noise for each flow (Without belt case).

The effects of flow rate on the distribution of sound pressure level for the fan with casing of 730mm size is shown in figure 10. It is found from this figure that (1) the turbulent noise over broad band frequency is the highest at maximum flow rate point (ϕ_{max}) and the noise become low according to reduce of flow rate. (2) The interaction noise contribute overall noise considerably. (3) The interaction noise reduce somewhat at the maximum point (ψ_{max}) and the minimumu point (ψ_{min}) of pressure coefficient because the wake of blade diffuses. But in comparison of overall noise at these flow rate, that level is almost same except the maxmum flow rate point.



(a) SPL(L) and K_s(L)



(b) SPL(A) and K_s(A)

Fig.11 Dependence of noise and specific noise. level on flow rate (With belt case).

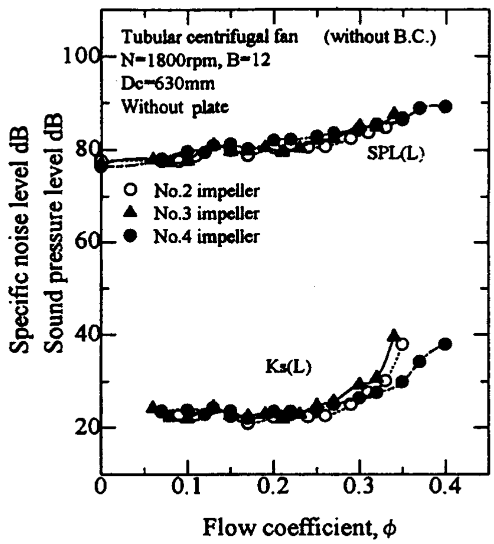
4.3 Overall noise and specific noise level

To judge the quality of a fan from the view-point noise reduction, the specific noise level, K_s, has been used. It is difined by

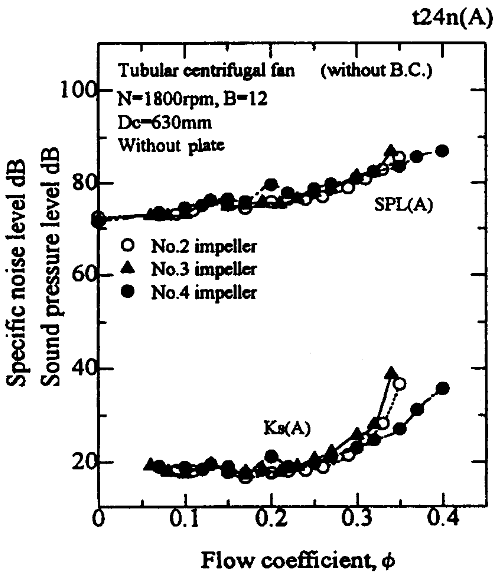
$$K_s = SPL - 10\log10(QP_T^2) + 20$$

where SPL is the sound pressure level in dB, Q is the fan flow rate in m^3/min, and P_T is the total pressure rise in Pa. Figure11 shows the change in the overall noise and the specific noise level with flow rate for tubular centrifugal fan with belt case. Figure11 (a)is the results based on the overall noise, SPL(L) measured with use of L characteristics of sound level meter and the specific noise level, K_s(L) based on SPL(L) and figure 11 (b)is SPL(A) measured with use of A characteristics and K_s(A). From these results, both SPL(L) and SPL(A) become the maximum at maximum flow coefficient point (ϕ_{max}) and begin to decrease with reducing the flow coefficient and become flat at the flow rate region which is less than the flow coefficient of 0.2.

On the other hand, both K_s(L), K_s(A) become maximum at the maximum flow coefficient like the overall noise and begin to decrease adruptly with reducing the flow coefficient to $\phi=0.2$ and increase

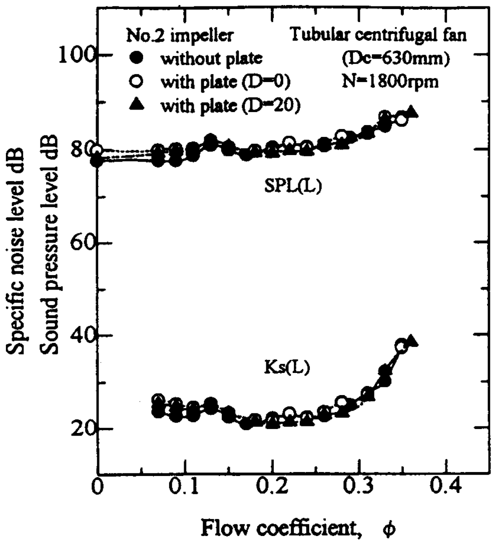


(a) SPL(L) and K_s(L)

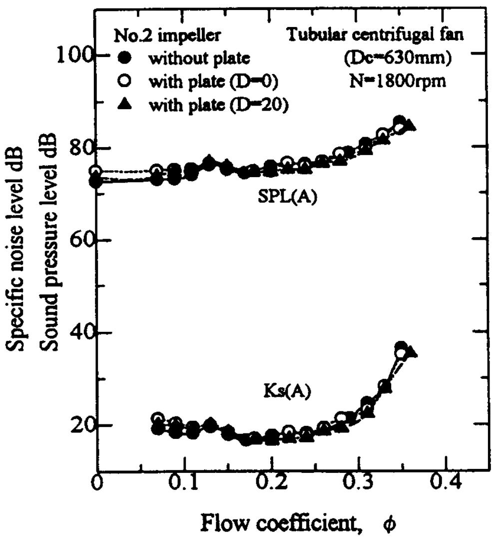


(b) SPL(A) and K_s(A)

Fig.12 Dependence of noise and specific noise. level on flow rate (Without belt case).



(a) SPL(L) and K_s(L)

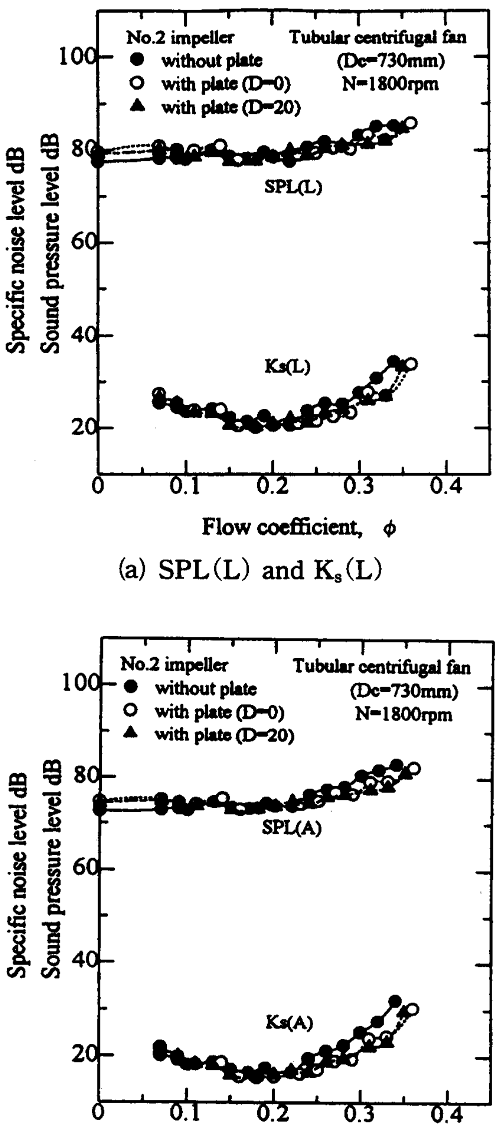


(b) SPL(A) and K_s(A)

Fig.13 Dependence of noise and specific noise. level on flow rate(Effects of preventive plate, D_c= 630mm).

toward the closing point again. In *SPL* and K_s, the No. 4 fan (● mark) had impeller with inclined blade is the lowest among fans and is the most among fans in the maximumu flow. This is cause the pressure loss based on the impact and the bend of stream line reduces due to incline blade.

The results of *SPL* and K_s for fan without belt case are shown in figure 12. It is found that the overall noise decreases with reducing the flow rate. The specific noise level decreases adruptly from maximum flow coefficient point to $\phi=0.2$ and become nearly constant from this point ($\phi=0.2$) to closing point ($\phi=0.2$).



(a) SPL(L) and K_s(L)

(b) SPL(A) and K_s(A)

Fig.14 Dependence of noise and specific noise. level on flow rate (Effects of preventive plate, D_c= 730mm).

As compared with the fan with belt case shown in figure 11 and without belt case in figure 12, it is found that by taking off the belt case, an reduction in overall noise by about 2 dB and the specific noise level by 5 ~10 dB can be realized. This effect is remakable in the No. 2 and the No. 3 fan with a little inclined angle blade. This fact suggest that the pressure drop due to the belt case larger than that due to the impact and the bend of stream line.

The effects of preventive plate on overall noise, *SPL* and specific noise level, K_s for fan with casing of 630mm size and 730mm size are shown in figure 13 and 14 respectively. In fugure 13, both *SPL*(L) and *SPL*(A) of fan attached the preventive plate located over the leading edge of impeller is higher than that of another location by about 1 dB. Similarly, by reason of this fact, the specific noise level is higher than that of another location by about 1 dB.

The *SPL* and K_s of the fan with casing of 730mm size are shown in figure 14, when the preventive plate is installed over the leading edge of impeller ($D=0$, ● mark), K_s and *SPL* for the fan with the preventive plate are become lower than that of fan without preventive plate by 1 ~ 2 dB. This level is by no means inferior to the normal centrifugal fan with scroll casing.

5. Conclusions

The effects of belt case, inclined angle of blade, the casing of size and preventive plate on the aerodynamic and fan noise characteristics were examined by using three kinds of fans. The results are summarized as follows.

(1) The existence of belt case influence the aerodynamic characteristics of fan. The total pressure of the fan with belt case is lower than that of the fan without belt case by 10~20% because of the pressure loss due to the belt case. Therefore, the fan efficiency decreases by the same degree. Moreover, the maximum flow rate of fan with belt case reduces by about 20% and the flow rate region having low specific noise level the same as η_{max} -point decreases by about 25% as compared fan without belt case.

(2) The existence of belt case influence the characteristics of overall noise. That of fan fan with belt

case reducese as reducing the flow rate from the maximum flow rate point to $\phi=0.2$ point and become almost constant between $\phi=0.2$ and $\phi=0$. On the other hand, the overall noise for the fan without belt case reducese linearly as reducing the flow rate from the maximum flow rate point to closing point. In the region of low specific noise level, the latter is wider than the former.

(3) As far as this experimental results, when the diameter of casing is 1.32 times as large as that of impeller, the fan efficiency and the pressure increase as the preventive plate approach the leading edge of impeller but if that is 1.16 times, when the preventive plate locates over the leading edge of impeller, the fan efficiency and total pressure somewhat reduce.

(4) As far as this experimental results, when the the diameter of casing is small, the aerodynamic characteristics are improved by using inclined blade.

Acknowledgment

The authors wish to express their gratitude to H. Nagamatsu, H. Iwashita, Y. Iwatsu, M. Kataoka for thier assistance in the experiments and S. Kondou and Y. Suzuki, Matsushita Seiko Co., Ltd for their support of this research project.

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