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Studies on the Noise of Axial Flow Fan*

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Among the noises of an axial-flow fan with stator and rotor, a discrete noise is most troublesome. This discrete noise is chiefly caused by the interaction between stator and rotor. In this paper we dealt with the axial spacing and the tilting angle in the spanwise direction between guide vane and rotor blades as parameters. Some experiments on the effects of those parameters on noise reduction were made, and at the same time measurements of the pressure fluctuation on the blade surface of the guide vane were carried out.

To abate the noise caused by the interference of blades with viscous wakes, we tried to make small the non-uniformity of the flow from the inlet cascade by inserting a flow-regulator between the stator and the rotor, and we found that this method is effective for the reduction of the noise.

Finally we clarified experimentally that the method of increasing the tilting angle is effective for noise reduction, if the axial spacing between the rotor and stator vanes is within a certain range.

1. Introduction

It is well known that the sound generated by an axial-flow fan consists of a discrete noise and a broad band noise. As the cause of generation of the discrete noise, we can consider several factors, but in an axial-flow fan with rotor and stator, it is mainly due to the lift fluctuation on blade caused by the potential and viscous interferences between the blade rows. In this paper we report the results of some measurements about the reduction of those interferences.

2. Experimental apparatus and experimental method

The axial-flow fan used in this experiment is, as shown in Fig. 1, a single stage axial-flow fan with rotor and stator. This fan may be available as the upstream guide vane type or downstream guide vane type by changing the arrangement of stator and rotor, and may be installed with 12-guide-vane ($N=12$) or 15-guide-vane ($N=15$) on the stator, whereas the number of rotor blades is constantly 12. We used a thin airfoil section or NACA 65-series airfoil section as a blade of experimental fan. The arrangement of cascades used in the experiment are shown in Fig. 2. This experimental fan may be varied in the spacing between the rotor and stator, and is driven at constant

speed of 2 000 rpm by belts.

In performing experiments the following attention was paid. The duct composed of two co-axial circular cylinders was used so that the cut-off might be considered and the acoustic impedance might be estimated⁽¹⁾. As the device of the flow control we avoided use of a usual damper after considering the impedance change, and adopted a wire-netting which showed a greater resistance to air flow but a smaller resistance to sound propagation. It was examined experimentally that the maximum sound-pressure drops due to wire-netting were 0.6 dB when six sheets of wire-netting of $36\# \times 60M$ were piled up one over another. The flow-control of the fan being done within six sheets of wire-netting above mentioned, sound-pressure drops can be neglected when the flow rate is small. The duct was wrapped with sound-arresting materials and the outlet of the duct was placed out of door through the wall of the laboratory. In this way the noise of the fan was measured separately at the suction and delivery sides of the fan.

On the occasion of noise-level measurements under several operating conditions (for instance, at several distances between the rotor and the stator, or under several combinations of cascades) it takes enormous

Fig. 1 Experimental fan

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time to obtain an acoustic output from the sound-pressure distribution. So after confirming that the similarity of the sound-pressure distribution holds, for instance, even if the axial spacing between the rotor and stator vanes was changed (Fig. 3), only the sound level at one point was measured as a representative level. A microphone was placed, not to be affected by the flow of the fan, in the direction of 20° from the center line either at the delivery side or at the suction side.

(a) Upstream guide vane type, whose guide-vane is NACA-65(13.1)13, $l_s=130.7\text{ mm}$ and $\beta_s=73.1^\circ$, and rotor blade is NACA-65(4.5)12, $l_r=136.1\text{ mm}$ and $\beta_r=29.0^\circ$. Axial mean flow velocity V under full discharge is 17.3 m/sec . (b) Downstream guide vane type, whose rotor blade is NACA-65(4.5)12, $l_r=136.1\text{ mm}$ and $\beta_r=35.5^\circ$, and guide vane is NACA-65(13.1)13, $l_s=130.7\text{ mm}$ and $\beta_s=78.6^\circ$, $V=18.5\text{ m/sec}$. (c) Upstream guide vane type, whose guide-vane and rotor blade are thin airfoil section, $l_s=142\text{ mm}$, $l_r=87\text{ mm}$, $\beta_s=60.4^\circ$, $\beta_r=25.0^\circ$ and $V=16.6\text{ m/sec}$. In this figure, following values are identical; $t_s=108.9\text{ mm}$ (in $N=12$), $t_s=87.1\text{ mm}$ (in $N=15$), $t_r=108.9\text{ mm}$ and $U=43.5\text{ m/sec}$.

Fig. 2 Cascades used in axial-flow fan (all the dimensions are the values in the arithmetic mean diameter $D=414\text{ mm}$)

○: Thin airfoil, $N=12$, $Z=4\text{ mm}$
●: Thin airfoil, $N=12$, $Z=59\text{ mm}$
×: NACA-65 series, $N=12$, $Z=12\text{ mm}$

Fig. 3 Distribution of over-all sound-pressure in delivery side

3. Methods of noise reduction

3.1 The method of increased axial spacing between rotor and stator vanes

The discrete noise generated by an axial-flow fan is considered chiefly to be due to the lift fluctuation on the stator and rotor blades caused by the potential and viscous interferences between the blade rows. It was already pointed out by I. J. Sharland⁽²⁾ that the magnitude of the interaction between the stator and rotor blade rows depended on their axial spacing. So we calculated the magnitude of the fluctuating lift induced by the potential flow, by applying complex velocity functions to the thin airfoil cascades which were the same as one used in the present experiment. The results of the computation are given in Fig. 4, showing that the fluctuating lift becomes suddenly smaller as the axial spacing between the rotor and stator vanes (in rotor blade chord) becomes greater. On the cascade blade situated downstream, the fluctuating lift induced by a viscous wake relates to the magnitude of maximum velocity-defect U_e and the breadth Y of the viscous wake. From the computational results⁽³⁾, it was found that the magnitude

Fig. 4 Fluctuating lifts on the rotor and stator blades, where axial-flow velocity is 16.6 m/sec

Fig. 5 Non-uniformity coefficient α

of fluctuating lift depended much on U_c rather than on Y .

Figure 5 shows the non-uniformity coefficient α of the blades used in the present experiment, where $\alpha = U_c/V_m$ and V_m is the main-flow velocity in the outside of a wake. It is clear from the figure that α becomes also smaller as the axial spacing between the stator and rotor becomes greater. From the above results it is easily guessed that a discrete noise made by an axial-flow fan may be reduced by increasing the axial spacing between the rotor and stator vanes.

3.1.1 Relation between sound levels and axial spacing Z

The sound levels and the characteristic curves

Fig. 6 Fan performance and noise levels of the cascades(a) shown in Fig. 2, where ψ means pressure coefficient and $N=12$

Fig. 7 Fan performance and noise levels of the cascades(b) shown in Fig. 2, where $N=12$

Fig. 8 Fan performance and noise levels of the cascades(c) shown in Fig. 2, where $N=12$

of the experimental fans are shown in Figs. 6~8. Using these fans, the relation between the sound level and the axial spacing Z between the rotor and stator vanes was investigated. Concerning the cascade (a) shown in Fig. 2, the sound level shown in Fig. 9 was obtained. The downward tendencies of the sound levels with an increase of Z/l_s are observed both in the suction and in the delivery side, irrespective of stator-vanes number and airfoil section. The experimental result shows that this method of sound level reduction by increasing Z is effective in $Z/l_s < 0.2$.

In Fig. 10 the frequency spectra of the noise are shown. From this figure we can see that increasing the axial spacing Z makes the level of discrete components much lower but, on the contrary, makes the low frequency components of a broad band noise much greater. The magnitude of low frequency discrete components being influenced by the number of stator vanes or the profile of an airfoil section, it may happen that the fundamental discrete component of 400 c/sec is not reduced by increasing Z , but the reduction of high-frequency discrete components is surely expected.

On the other hand, in the cascade (b) shown in Fig. 2, the noise level has a tendency to have a minimum value at $Z/l_s \approx 0.4$ as shown in Fig. 11. The reason is considered to be that a greater broad band

Fig. 9 Relation between noise levels and Z/l_s in the case of the cascades(a) shown in Fig. 2 under full discharge

Fig. 10 Frequency spectrum of the noise in delivery side in the case of the cascades(c) shown in Fig. 2, where $N=15$

noise appears in the cascades (b) than in the cascades (a) as shown in Fig. 12, and this broad band noise has a great influence on the over-all level in $Z/l_r > 0.4$.

3.1.2 Relation between fluctuating pressure on a blade and axial spacing Z

It is known that the potential and viscous interferences between the blade rows become smaller with an increase of axial spacing Z between the stator and rotor blades, and it was found from the experiment above mentioned that the noise level fell with an increase of Z within a certain range of Z . Then about the fluctuating pressure acting on a blade, it may be expected to obtain any relation of it with the spacing Z .

In order to measure the fluctuating pressure on a blade some pressure pick-ups of condenser type were specially made, which were $9\text{ mm}\phi$ in the outer diameter, $7\text{ mm}\phi$ in the inner diameter and 1 mm in the thickness and had flat frequency characteristics from 0 to 2 000 c/sec. These pick-ups were mounted on stator vanes as shown in Fig. 13. So as to be able to compare with the sound pressure wave, the microphone of a sound-pressure meter was installed at the entrance of the experimental fan, and each pressure wave on a dual beam synchroscope was simultaneously

photographed. All of these experiments were carried out under full discharge.

Some typical fluctuating pressure patterns are shown in Figs. 14 and 15. In these figures upper waves are pressure patterns from the pick-ups and lower ones are sound pressure patterns. However these pick-ups have various pressure sensitivities according to an inaccuracy of handiwork, while sweep-time of the synchroscope is 1 msec/div. in all these measurements. After reading the mean value of the amplitude (amplitude means here the sum of an upper and a lower amplitude), these values were converted to pressure magnitudes based on each static characteristics.

The change in the amplitudes of the fluctuating pressure with variation of the spacing Z is shown in Figs. 16~21. In the cascade (a) shown in Fig. 2 the fluctuating pressure on the stator vane becomes smaller rapidly with an increase of Z . On the other hand, if we see these figures as the distributions along the guide-vane chord, the nearer it comes to the trailing edge, the greater the fluctuating pressure becomes. This indicates that the nearer it comes to

Fig. 11 Relation between noise levels and Z/l_r in the case of the cascades(b) shown in Fig. 2 under full discharge, where $N=12$

Fig. 12 Frequency spectrum of the suction-side-noise of the cascades(b) shown in Fig. 2, where $N=12$

(a) NACA-65 series airfoil section

(b) Thin airfoil section

Fig. 13 Setting position of pick-up

The pressure wave pattern of, (i-a) the No. 4 pick-up installed on the concave surface of the guide-vane in the cascades(a) shown in Fig. 2, where $Z=4$ mm, (i-b) the No. 4 pick-up installed on the convex surface, (ii-a) the No. 4 pick-up installed on the pressure surface of the guide-vane in the cascades(c), where $Z=5$ mm, (ii-b) the No. 4 pick-up installed on the suction surface. (iii-a) the No. 1 pick-up installed on the concave surface of the guide-vane in the cascades(b), where $Z=6.5$ mm, (iii-b) the No. 1 pick-up installed on the convex surface.

Fig. 14 Pressure wave patterns (upper) and sound-pressure wave patterns (lower) in the case of the 12-guide-vane assembly

The pressure wave pattern of, (i-a) the No. 1 pick-up installed on the concave surface of the guide-vane in the cascades(a) shown in Fig. 2, where $Z=4$ mm, (i-b) the No. 1 pick-up installed on the convex surface, (ii-a) the No. 4 pick-up installed on the pressure surface of the guide-vane in the cascades(c), where $Z=5$ mm, (ii-b) the No. 4 pick-up installed on the suction surface, (iii-a) the No. 1 pick-up installed on the concave surface of the guide-vane in the cascades(b), where $Z=6.5$ mm, (iii-b) the No. 1 pick-up installed on the convex surface.

Fig. 15 Pressure wave patterns (upper) and sound-pressure wave patterns (lower) in the case of the 15-guide-vane assembly

the trailing edge, the greater the interference of a potential flow becomes, and these experimental results agree with those of the potential theory. On the other hand, in the cascades (b) shown in Fig. 2 the fluctuating pressure distributions along the outlet-guide-vane chord do not have a clear tendency compared with ones along the inlet-guide-vane chord in the cascades (a) shown in Fig. 2, but it is understood from the above results that the fluctuating pressure becomes great rapidly at the vicinity of the leading edge when Z is small, and the variation of the fluctuating-pressure distribution along the guide-vane chord is almost small when Z becomes slightly great. As for such a tendency it is reasonable to interpret from Figs. 4 and 5: In the vicinity of the trailing edge there appear the influences of both interferences caused by a potential flow and a viscous wake of an inlet-cascade, but there remains only one caused by a viscous wake when the spacing Z is increased.

Next the case of $N=15$ will be considered. The fluctuating-pressure waves on both surfaces of a stator vane with $N=12$ have a phase-difference of π radian at the same chordwise positions. On the other hand, it was found that the fluctuating-pressure waves in $N=15$ have almost the same phase after consideration of the original phase-advance of $(2/5)\pi$ radian on a concave surface or pressure surface of a stator vane. The mean amplitude of the pressure fluctuation becomes

greater at $N=15$ than at $N=12$, but on the contrary the fluctuating-lift becomes greater at $N=12$ than at $N=15$ from the above phase relations. The corresponding noise level appears smaller at $N=15$ than at $N=12$ by the reduction of the lift fluctuation and the appearance of a duct cut-off.

3.2 Method of inserted flow-regulator

As investigated in the previous paragraph, the discrete noise level becomes lower with an increase of Z . In such a noise reduction method the interferences caused by both potential flow and viscous

Fig. 16 Mean fluctuating pressure on the guide-vane (in $N=12$) versus spacing Z in the cascades(a) shown in Fig. 2

Fig. 17 Mean fluctuating pressure on the guide-vane (in $N=15$) versus spacing Z in the cascades(a) shown in Fig. 2

Fig. 18 Mean fluctuating pressure on the guide-vane (in $N=12$) versus spacing Z in the cascades(c) shown in Fig. 2

Fig. 19 Mean fluctuating pressure on the guide-vane (in $N=15$) versus spacing Z in the cascades(c) shown in Fig. 2

Fig. 20 Mean fluctuating pressure on the guide-vane (in $N=12$) versus spacing Z in the cascades(b) shown in Fig. 2

wake become small. In this paragraph it is attempted to reduce the influence of a viscous wake on the fan noise.

3.2.1 Influence of a flow-regulator on the wake

As a flow-regulator which makes small the non-uniformity of air flow, a wire-netting or a wire-grate is available. But it is evident that a deterioration of fan performance is brought about by inserting such a flow-regulator inside the fan. Then it is important to find out a regulator that hardly gives a bad influence on the fan performance.

In the first place, using a cascade tunnel, we measured the velocity distribution of the wake behind a wire-netting of $24\# \times 10M$ or a wire-grate of 0.6 mm in the wire-diameter set at 12 mm behind the blade trailing edge. As shown in Fig. 22, the main flow outside the viscous wake of a blade is disturbed by viscous wake of the flow-regulator elements, but this disturbance rapidly decays with an increased distance from the blade trailing edge. Taking the mean flow velocity as the main velocity V_m , Fig. 23 was obtained as the relation between the non-uniformity coefficient and Z/l , where l is an airfoil chord-length. It is evident from this figure that the coefficient α becomes fairly small by inserting a flow-regulator. From the velocity distribution shown in Fig. 22, it is known that by inserting a flow-regulator between the rotor and the stator the high frequency components of a broad band noise may be increased but the low frequency components of a discrete noise may be decreased.

3.2.2 Effect of a flow-regulator on the fan noise

The effect of a flow-regulator on noise reduction was investigated using the cascades (a) shown in Fig. 2. As a flow-regulator, five kinds of wire-nettings

Fig. 21 Mean fluctuating pressure on the guide-vane (in $N=15$) versus spacing Z in the cascades (b) shown in Fig. 2

with various meshes were adopted, the flow-resistance of which was previously measured using a wind tunnel. Flow-regulators used in the present experiment are annular nettings mounted on frames. These flow-regulators were installed at the position of 3 mm behind the trailing edge of inlet-guide vanes. The fan with a flow-regulator was driven under full discharge.

After a series of performance tests the relation between the reduction of an over-all noise level and the

The velocity distribution in the case (a) without a flow-regulator, (b) with the wire-netting of $24\# \times 10M$, (c) with the wire-grate which was placed in the position of 12 mm behind trailing edges. The figure of the wire-grate is one looked from the behind of trailing edge.

Fig. 22 The velocity distribution of a viscous wake

Fig. 23 Non-uniformity coefficient α versus Z/l in the case with flow-regulators

pressure-loss coefficient ζ_k of a wire-netting was obtained as shown in Fig. 24, where $\zeta_k = \Delta p_s / \{ (1/2) \times \rho v^2 \}$ and Δp_s means a static pressure drop of the fan. From this experimental result it is found that a noise reduction may not be expected from a large mesh. Using two flow-regulators, the one being a wire-netting of $24\# \times 10M$ and the other a wire-grate which is shown in Fig. 26, the performance and the noise level of the fan with a 12-guide-vane assembly were measured and the results are shown in Figs. 25 and 26. From these figures it is found that in the case of the wire-netting the noise and the fan performance are both reduced, but in the case of the

Fig. 24 Pressure-loss coefficient ζ_k versus noise reduction (in $N=12$)

Fig. 25 Influences of flow-regulators on fan performance

Fig. 26 Flow coefficient versus noise levels in the case with various flow-regulators. The flow-regulator shown in the figure is the wire-grate used in the experiment

wire-grate the performance is hardly influenced and its noise level is reduced as much as in the wire-netting. Thus the conclusion is drawn that the use of a flow-regulator which shows a small resistance in air flow but a great effect of flow-regulation produces a good effect of noise reduction. On the other hand, according to the investigation about the relation between flow coefficient φ and noise level, the effect of flow-regulation is lost in the low-flow region, i.e. $\varphi < M$ as shown in Fig. 25, and the noise level approaches the level in the case with no flow-regulator. The above relation was the same in the suction side.

From the results of a frequency analysis shown in Fig. 27 it is found that the use of a wire-grate considerably reduces the fundamental discrete noise of 400 c/sec but increases somewhat the high frequency components above the third harmonics of 1600 c/sec. This result is understood from the velocity distribution behind the flow-regulator. In the case of $N=15$ as shown in Fig. 28 the over-all level is somewhat lower than the level of the one with no flow-regulator, but the low frequency discrete noise is largely reduced. Considering the easy decay and easy absorption of high frequency components, the above result at $N=15$ shows also the effectiveness of noise reduction.

Fig. 27 Frequency spectrum of the delivery-side noise caused by the cascades(a) shown in Fig. 2 under full discharge, where $Z/L_s=0.313$ and $N=12$

Fig. 28 Frequency spectrum of the delivery-side noise of the cascades(a) shown in Fig. 2 under full discharge, where $Z/L_s=0.313$ and $N=12$

Next it was investigated if the effectiveness of noise reduction by increasing the spacing Z between the rotor and the stator could be expected under the action of a flow-regulator. For this purpose the spacing Z was varied by installing a wire-netting of $36\# \times 60M$ immediately behind the trailing edges of a stator with a 12-guide vane assembly. As shown in Fig. 29, the effectiveness of noise reduction by increasing Z is not missed even under the insert of a flow-regulator.

3.3 Method of increasing the tilting angle between rotor and stator blades

As another parameter on the fan noise, the tilting angle in the spanwise direction between the stator and rotor blades is considered. Zero tilting angle means that a fluctuating lift caused by the interference between the guide vane and rotor blades has same phase in the spanwise direction. With a tilting angle, certain corresponding phase-difference in the spanwise direction arises and some reduction of a radiating sound power may be expected. Concerning this I.J. Sharland had already experimentally obtained

Fig. 29 Noise levels versus Z/l_s in the case with flow-regulators

Fig. 30 Variation of fan performance with tilting angle θ (in $N=15$)

the relation between the tilting angle and the sound level under the condition of a very short spacing of Z . In this paper the influence of tilting angle θ and the spacing Z on noise levels and frequency components was experimentally studied using cascades (c) shown in Fig. 2.

3.3.1 Effect of the tilting angle on the fan noise

Tilting angle θ was given by reclining stator vanes at 8° , 22° , and 33° in the counter direction of the revolution under same span length. The effect of the tilting angle on the fan performance may be negligible as shown in Fig. 30. Experiments were carried out in both cases of $N=12$ and $N=15$, but as we couldn't find any great differences in their results, only the results at $N=15$ are reported here.

The relations between the flow coefficient φ and the noise level are shown in Figs. 31 and 32. It is found from these figures that the noise levels have a minimum value at the point near the maximum flow coefficient in both the suction and the delivery side, and the noise level is decreasing with an increase of θ in the region of great values of φ , and the influence of θ on the noise level becomes smaller in the region of small values of φ . As is found from the comparison between Fig. 31 and Fig. 32, there is a great difference between suction and delivery side in the

Fig. 31 Delivery-side noise-levels versus flow coefficient φ , when θ is varied

Fig. 32 Suction-side noise-levels versus flow coefficient φ , when θ is varied

Fig. 33 Sound power levels versus tilting angle θ

decreasing rate of the noise level caused by an increase of θ . Then the sound power levels of both sides were measured under full discharge, and the results are shown in Fig. 33. From this figure it is found that the effects of the tilting angle θ on noise reduction appear, at least in this experiment, more remarkably in the delivery side than in the suction side, and noise reduction becomes greater with an increase of θ . The high frequency discrete components of noise are relatively weakened by increasing θ . The tendency of frequency components varying with a change in the flow quantity was almost the same irrespective of θ .

3.3.2 Effect of spacing Z in the case with a tilting angle

An experiment on the fan noise was performed by changing the spacing Z and the tilting angle θ under full discharge. The result is shown in Fig. 34 and it is found from the figure that the tendency of the noise level declining with an increase of Z/l_s is fundamentally almost same as that at $\theta=0^\circ$, but the influence of θ disappears in $Z/l_s > 0.15$. As one of the reasons it may be pointed out that the influence of θ on air flow inside the fan also disappears in $Z/l_s > 0.15$. From these facts it is concluded that the effects of θ on noise reduction can be expected for a certain values of Z/l_s , where noise reduction becomes greater with an increase of θ . When the spacing Z is increased keeping the tilting angle at a certain value,

Fig. 34 Noise levels versus spacing Z when θ is varied

high frequency discrete components decrease considerably just as in the case of $\theta=0^\circ$.

4. Conclusion

In order to reduce the fan noise, several methods may be considered. In this paper we noticed a discrete noise caused by interactions between the rotor and stator blades, and reported some experimental results of noise reduction. From this study it has been made clear that, when interference caused by a viscous wake is predominant, a flow-regulator, such as the wire-grate, is effective for noise reduction, and a further reduction may be expected from a more effective flow-regulator.

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