The Performance Characteristic of an Air-separator of an Axial Flow Fan*

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An air-separator as a new means for stabilizing the characteristics in the low flow rate region of an axial flow fan has been studied experimentally. The influence on the performance of this equipment of its parameters such as the structure of the flow passage and the dimensions has been examined as well as the influence of hub-tip ratio of the rotor and the aspect ratio of the rotor blades.

It is found that the equipment is useful even in such severe conditions that blade separators are useless

Key Words : Turbomachinery, Axial Flow Fan, Air-separator, Blade-separator, Performance Characteristics, Blower Instability

1. Introduction

The performance characteristics of an axial flow fan in partial flow rate region which may introduce unstable operation has been a serious problem to be solved. Among many means to stabilize the characteristics, a suction ring¹, a blade-separator^{1,2} and some kinds of casing treatments^{3,4,5} have been proved to be useful. In addition, a rotating inlet guide vane has been proposed⁶ to suppress the strong rotating flow in front of a rotor that is produced by the reverse flow from a rotor.

The mechanism of improvement of the low flow rate characteristics by a suction ring and a blade-separator is considered to be such that as soon as the rotor falls into stall regime, they make all blades, not some of them, stall and induce a centrifugal effect. However, both means are not effective enough for a heavily-loaded rotor because they fail to suppress completely the stall in partial flow rate region. Furthermore, the casing treatment which is one of the most promising means is complicated for manufacturing and expensive.

Compared with the above-mentioned means, an air-separator^{7,8} is expected to be more effective in view of its performance mechanism. Its principle is similar to that of a blade-separator or a suction ring, but the air-separator may possibly exhibit higher performance than the conventional means because it can accomplish complete separation of reverse flow from inlet flow and actively

discharge the stalled fluid from the tip region of rotor blades. Ivanov⁷ obtained a patent with respect to the fundamental structure of the equipment, but the details such as optimum dimension, constitution of its flow passage, etc. are not yet well established. Furthermore, satisfactory explanation of its operation mechanism is not yet presented.

In this work, the authors examined the influence of the dimensions of the constitutive elements of the equipment on its performance at first and studied the advantageous features of the equipment over a blade-separator.

In many devices of this sort, the deterioration of the rotor performance at the designing point is inevitable, but it may be possible with an air-separator to avoid this drawback because the reverse flow circuit can be closed when the rotor is in sound operation regime.

2. Experimental Set-up and Measurement

2.1 Rotor Hub-to-tip ratio B of the axial flow rotor used in the experiments is 0.5 and its outer diameter is 500 mm. The cascade of both the guide vane and the rotor are composed of blades of NACA 65 profile of constant chord length in the spanwise direction. The specific speed of the rotor n_s is 1770 [m³/min., m, rpm]. Flow pattern of the rotor is free-vortex type and the distributions of the setting angles β of the blade (inclination angle of the blade chord from the peripheral direction), solidity σ , diffusion ratio $DF=1-(W_2/W_1)+(\Delta W_u/2\sigma W_1)$ (W_1, W_2 : relative flow velocity at the inlet and exit of the rotor, respectively, ΔW_u : peripheral velocity deflection of the flow) and the attack angle α^* are specified in Fig.1.

Since the operation of an air-separator is considered to be related fundamentally with the smooth circulation of the reverse flow, the aspect ratio AR of blades of the rotor is expected to be one of the key factors determining its performance. Therefore, two rotors

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Fig.1 Designing parameters of the test rotors

Table 1 Major dimensions of the rotors

	IMPELLER I		IMPELLER II	
BLADE	ROTOR	STATOR	ROTOR	STATOR
CHORD LENGTH	100 mm	90 mm	40 mm	36 mm
BLADE NUMBER	12	17	30	37
ASPECT RATIO	1.2	1.4	3.1	3.4

Fig.3 Casings alternatively used in the tests

with blades of different aspect ratios, i.e., rotor I and II (Table 1) are used in the experiments. The blades are planted with pins on the boss drum so that their attack angle can be adjusted to arbitrary value. However, since the blades are castings and the manufacturing accuracy of the profile is not satisfactory, especially in rotor II, the performance characteristics of the two rotors are not identical, even at the designing point.

2.2 Flow passage of the test fan The overall schema of the test rig is shown in Fig.2. In order to avoid the inlet distortion of the flow, the driving motor is installed downstream of the rotor. At the part ③ of outer casing of the flow passage of Fig.2, either of the two casings I or II shown in Fig.3 can be alternatively installed. Casing I is a normal casing and has the same inner diameter as the rotor outer diameter. Together with cylindrical boss, casing I constitutes a long

Fig.2 Overall shema of the test rig

Fig.4 Enlarged boss

annular flow passage near the rotor and the stator. The blade-separator is installed just upstream of the rotor within the normal casing. The casing II has a part of 270 mm in length with enlarged diameter at the upstream of the rotor. Inner diameter of the enlarged part is by 100 mm larger than that of the normal casing. For an air-separator, casing II is used. The location of the rotor can be adjusted in axial direction within 100 mm. Although the design hub-tip ratio was 0.5, experiments were also conducted for $B=0.7$ and 0.85 . In those cases, the boss was enlarged in both upstream part and downstream part by 280~290 mm in length, as shown in Fig.4. The rotor boss diameter was at the same time enlarged. In Fig.4, numbers in brackets are for $B=0.85$ and those outside are for $B=0.70$.

2.3 Air-separator and blade-separator In the experiments, the performance of the blade separators was also tested in order to compare with that of air-separators. The air-separators were composed of casing II and a separator ring whose diameter was the same as that of the normal part of the casing. The

ring was installed within the enlarged part of the casing as shown in Fig.5(a). Both ends of the ring make clearances with the casing which constitute suction and exhaust ports of a reverse flow from the rotor. On the ring, generally fins are planted on the side of the flow passage between the ring and the casing. So that the separator rings look as shown in Fig.6 (a). The fins were not always bent, but some kinds of straight fins were also tested. The parameters determining the dimensions of an air-separator are l_b , the length of the part of the blade tip uncovered, l_a , l_d , the clearances of the air-separator passage at both ends of the ring, l_{as} , the fin length, h_a , height of fins, l_f , length of the part of fins overhung and θ , the inclination angle of fins. Table 2 shows the major dimensions assigned to each air-separator specified by the signs listed in the left column. The air-separators EI, EII, F, G, H, I, J have straight fins and others have bent ones, as shown in Fig.6(a). The fins used in EI~EIV

Table 2. Dimensions of major air-separators

	l_f mm	l_{as} mm	h_a mm	θ deg	z_f
E I	15	115	30	0	15
E II	10	235	45	0	15
E III	10	235	45	-15	15
E IV	10	235	45	30	15
E V	10	235	45	45	15
E VI	10	160	45	45	15
F	15	115	30	30	15
G	15	115	30	45	15
H	15	115	30	-15	15
I	15	115	30	0	32

Fig.5 Installation of equipments

are larger than ones in others as shown by broken lines in Fig.5(a). The fin length l_{as} is measured in axial direction. Blade-separator ring which is installed in the suction duct just upstream of the rotor is shown in Fig.6(b). The axial flow fan rotor equipped with the blade separator is shown in Fig.5(b). The dimensions of the blade separator were optimized referring to the experiments by Murata et al.² Three kinds of blade-separators A, B, C were tested in the experiments. The combinations of length l_{bs} and height h_b of the fins in each separator are A(100,10), B(100,5), C(200,20) (in mm) and in all of

Fig.6 Separator rings

Fig.7 Performance characteristics of the rotor I ($B=0.5$)

Fig.8 Performance characteristics of the rotor I ($B=0.7, 0.85$)

them, the distance between the leading edge of the rotor blade and that of separator fins l_a' is 5 mm.

2.4 Performance characteristics of test fan

2.4.1 Head coefficient and flow coefficient

Total head P_t [m], fan static pressure head P_s [m] and flow rate Q [m³/s] were measured at the section ⑦ of Fig.2. So P_t and P_s include losses due to friction of duct and bent effect. Head and flow rate are non-dimensionalized as follows.

$$\psi_t = \frac{P_t}{U_t^2/2g}, \quad \phi = \frac{Q}{\pi r_t^2 U_t} \quad (1)$$

where U_t, r_t are rotating speed of blade tip [m/s] and rotor outer radius [m], respectively.

2.4.2 Rotor I Fig.7 shows the performance characteristics of the rotor I for various blade setting angles. In the figure, $\Delta\beta$ is the amount of the inclination angle of the blade from the nominal one ($\beta_t=28.2^\circ$ at the blade tip which is by 5° larger than that shown in Fig.1) and is taken as positive

Fig.9 Performance characteristics of the rotor II

Fig.10 Effect of separator ring alone

when the stagger angle of the cascade decreases. The larger the value of $\Delta\beta$, the heavier the blade load is for the same flow rate. In Fig.7, the parts of the curve shown by broken lines are discontinuous corresponding to the inception and the suppression of a rotating stall. In these parts, generally a hysteresis appears. When $\Delta\beta=15^\circ$, since the rotor could not attain peak head even at maximum throttle opening, the curve of the larger flow rate were obtained by removing some parts of the delivery duct. So, the flow rate could not be changed continuously and a hysteresis of the curve could not be detected. The larger the value of $\Delta\beta$, the larger the flow rate of peak head is and in the flow rate region smaller than that of peak head, the rotor comes into stall regime, which is fundamentally the same for any value of $\Delta\beta$. The flow rate ϕ_m corresponding to the minimal head is independent of $\Delta\beta$ and the head coefficient ψ_m decreases as $\Delta\beta$ increases. Therefore, improvement of performance is more difficult for larger $\Delta\beta$. Fig.8 shows influence of hub-tip ratio B on the performance characteristics when $\Delta\beta=0^\circ$. Three curves can almost converge to a single curve, if the abscissa is taken as ϕ' defined by $\phi' = v_z'/U_t$ where v_z' is mean axial flow velocity in the annual rotor section. However, in the flow rate region larger than the peak head, the larger the hub-tip ratio, the lower the head is, and the hydraulic mean diameter of the flow passage becomes the smaller.

2.4.2 Rotor II Figure 9 shows the performance characteristics of the rotor II. It shows the cases when $\Delta\beta=0^\circ$ for both $B=0.5$ and 0.7 .

3. Improvement of Low Flow Rate Performance of an Axial Fan

3.1 Case of small $\Delta\beta$. In this section, the results of experiments using rotor I for the case of small blade setting angle ($\Delta\beta=0^\circ, 5^\circ$) will be presented.

3.1.1 Effect of separator ring alone The

Fig.11 Effect of hangover of separator Fins

performance of the rotor equipped with an air-separator having no fin is shown in Fig.10. The distance between the rotor and the stator is $d/c=1.5$ (d , the distance between the locations of $1/4$ chord length of rotor blade and stator blade). In Fig.10, l_b , the length not covered by the casing wall of the rotor blade is varied. As shown in the figure, the air-separator without fins is useful to a certain degree, if l_b is properly chosen. That is, if too large value of l_b is taken with the blade tip exposed ($l_b=45$ mm), the load in the tip region of blades decreases and the improvement effect drops. If the blade tip is thoroughly covered ($l_b=0$), the reverse flow into the air-separator flow passage is obstructed and the improvement effect drops. But the case $l_b=0$ is more favorable than the case $l_b=45$ mm. The case when l_b is about $1/3$ of chord length is the most favorable and the unstable characteristics can be almost eliminated in this case. All the tests in this

paper hereafter were carried out by taking l_b as $1/3$ of chord length.

3.1.2 Effect of fins A blade separator is more effective to eliminate unstable characteristics of a fan than a suction ring because of fins. Similarly, for an air-separator, fins planted on the separator ring are highly effective to improve its performance. Figure 11 shows the performance of a fan equipped with an air-separator having 15 straight axial fins and otherwise identical with the one used in Fig.10. Fins are desirably extended as close as possible to the blade leading edge ($l_f=15$ mm) and in this case the unstable performance of the rotor is completely eliminated. Furthermore, for a lightly loaded rotor as this case, the inclination angle of fins θ is to be chosen as zero, or in the axial direction. When θ is positive, the reverse flow streaming out from the air-separator passage to the suction duct rotates in the same direction as the rotor

(a) $\Delta\beta = 0^\circ$

(b) $\Delta\beta = 5^\circ$

Fig.12 Influence of inclination angle of fins

Fig.13 Influence of ring clearance

Fig.14. Influence of stator-rotor distance

and the effect of the air-separator drops, especially in very small flow rate region, though not very much in the moderate flow rate region when the reverse flow is not intensive. The curve with the mark $\blacktriangle$ in Fig.12 shows the rotor performance when equipped with the blade-separator A. In this case of $\Delta\beta=0^\circ$, the blade-separator is as effective as the air-separator. A blade-separator is sufficiently effective for the case where the blade load is so light that the stall inception of blades immediately induce an axi-symmetric stall and the rotor performance curve shows only a kink when normal casing is used. Light load of blades corresponding to small blade setting angle Fig.12(b), shows the rotor performance when $\Delta\beta=5^\circ$ and otherwise identical with the one in Fig.12(a). The curves are not completely smooth suggesting that the increase of blade load makes the elimination of the unstable characteristics

Fig.15 Influence of hub-tip ratio ($B = 0.7$)

Fig.16 Influence of hub-tip ratio ($B = 0.85$)

of a rotor more difficult.

3.1.3 Influence of ring clearance and the distance between rotor and a stator Figure 13 shows the effect of the ring clearance l_a , l_d on the response of rotor I in the case of $\Delta\beta=5^\circ$. The test results show little influence of these parameters. So in the later tests, $l_a=35$ mm and $l_d=20$ mm were exclusively used.

In Fig.14, the influence of stator on the effect of an air-separator is shown. Although for rotating stall phenomena, it is known that the distance between stator and rotor is fairly important, it affects little the effect of an air-separator supposingly because the air-separator controls only the reverse flow at the tip of blades.

3.1.4 Influence of hub-to-tip ratio Figure 15 and 16 show the influence of the hub-to-tip ratio on the performance of an air-separator when $\Delta\beta=0^\circ$ and the stator is removed. When $B=0.7$, the effect of the blade-separator is not largely different from when $B=0.5$, and the discontinuity of the characteristic curve of the rotor suggests that the discontinuity of the flow regime is also the same at these two hub-tip ratios. However, an air-separator becomes less effective for higher hub-tip ratio. An air-separator can satisfactorily improve the performance characteristics of the rotor even for $B=0.7$, although a slight deterioration of head at design point is inevitable. In the tests at larger hub-tip ratio, $B=0.85$, the height of blade separator passage h_b [fig.5(b)] was reduced and chosen as 4/15 and 2/15 of blade span (37.5 mm). As shown in Fig.16, blade-separators do not work effectively. It is because the rotating stall cell covers almost entire blade span and the centrifugal effect can not manifest itself. Nevertheless, an air-separator works satisfactorily in this case also. This equipment is thus expected to be usefully applied to compressor rotors of high hub-tip ratio.

The deterioration of head at high flow rate region is considered to be due to the

Fig.17 Effectiveness of an air-separator for heavily loaded rotors ($\Delta\beta=10^\circ$)

fact that the suction flow passage through the air-separator passage and the equivalent hydraulic mean diameter decreases. However, using some auxiliary devices, it may be possible to close the air-separator passage in the flow rate region larger than the design point. So the above-mentioned drawback is expected to be avoided.

3.2 Case of large $\Delta\beta$ In Fig.12(b), improvement effect of an air-separator when $\Delta\beta=5^\circ$ is presented. Comparing with the test results of the case of $\Delta\beta=0^\circ$, rise of head in low flow rate region in this case is smaller. Figure 17 shows the case of larger $\Delta\beta$, namely $\Delta\beta=10^\circ$, blade-separators keep their effect yet, but deterioration of their effects become more evident. Further increase in $\Delta\beta$ makes the deterioration more serious as shown in Fig.18 which is for $\Delta\beta=15^\circ$. In this case, the unstable characteristics of the rotor can not be eliminated suggesting that the ability of

blade-separator to improve the performance saturates at some level of blade load. The blade-separator C having larger fins than those of type A (both height and length of the fins are twice as large as those of A) was also proved to be useless. Therefore, it can be concluded that the effect of a blade-separator decreases as the blade load increases and beyond some limiting value of blade load, the blade-separator becomes useless.

In Fig.17, the effect of the air-separator EI which has been proved to be most effective for lightly loaded rotor ($\Delta\beta=0^\circ, 5^\circ$) is shown for $\Delta\beta=10^\circ$. It turns out that it does not work satisfactorily, though better than the most effective blade-separator. Then, the air-separator EII which had longer and higher straight fins was tested. It shows higher performance, especially in the small flow rate region. Furthermore, in the case of $\Delta\beta=15^\circ$, the large bent fins whose part of 100 mm in length near the rotor blades is bent to reduce the shock of a reverse flow entering the air-separator flow passage proved to be well effective as shown in Fig.18. The inclination angle θ of the bent part of the fins

Fig.18 Effectiveness of an air-separator for heavily loaded rotors ($\Delta\beta=15^\circ$)

Fig.20 Case of rotor of high hub-tip ratio having high aspect ratio blades

Fig.19 Case of rotors having high aspect ratio blades ($\Delta\beta=0^\circ$)

Fig.21 Case of rotors having high aspect ratio blades ($\Delta\beta=15^\circ$)

has little influence on the total performance of the rotor in the unstalled region, but it has its optimum value in the small flow rate region. That is, negative θ is not desirable and the optimum value is such that the reverse flow experiences least shock and at the same time the reverse flow streaming out from the air-separator passage has no rotation. In this test, $\theta=30^\circ$ was found optimum.

As mentioned above, an air-separator proved to be effective even if the blade load of the rotor was higher than the value for which blade-separators are no longer useless. 3.3 Case of rotor with high aspect ratio blades Figure 19 shows the test results about the rotor II for $\Delta\beta=0^\circ$. The fins of an air-separator are straight and within the test range, the inclination angle and the number of fins influence little the improvement effect. In the smallest flow rate region, unstable characteristics can not be thoroughly eliminated and the effect of an air-separator is less than for the rotor I. However, no effect of the inclination angle of fins suggests that the unstable characteristics in the very small flow rate region are not due to the flow regime of the blade tip, in this case. Later experiments show that in this flow rate region, a rotating stall took place at the blade root¹⁾. A blade-separator is as effective as an air-separator for this rotor and its performance also deteriorates in the very small flow rate region.

Figure 20 shows the test results about the rotor of $B=0.7$ for $\Delta\beta=0^\circ$. Both rotors are found satisfactorily effective.

Figure 21 is the test results for $\Delta\beta=15^\circ$ using an air-separator having small fins. Except in very small flow rate region, it shows better performance than in rotor I. A blade-separator in this case eliminates completely the rotating stall and the shoot of head at the stall point is smaller than in rotor I. Although it may not be general tendency that the stall is weaker for a rotor having blades of high aspect ratio, it may be concluded that the air-separator is equally effective for the rotors of these blades, if they are otherwise identical.

4. Conclusions

This work examined experimentally the influence of the parameters of an air-separator on the effectiveness to improve the characteristics of an axial flow rotor. The results are as follows:

- 1) For usual rotors of reasonable load with small aspect ratio blades, an air-separator is satisfactorily effective for the purpose now in consideration. A blade-separator is as effective as an air-separator.
- 2) An air-separator is well effective for highly loaded rotors, if auxiliary devices such as large fins are used. Meanwhile, a blade-separator is useless for extremely highly loaded rotors.
- 3) For the rotors of high hub-tip ratio generating a nearly full span stall in small flow rate region, an air-separator is as equally useful, while a blade-separator is quite useless.
- 4) For the rotors having blades of high aspect ratio, an air-separator is equally useful.

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