ACTIVE CONTROL OF VANELESS DIFFUSER ROTATING STALL

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SUMMARY

Experiments were carried out to study the feasibility of active stabilization of vaneless diffuser rotating stalls. Pressure fluctuation at the diffuser inlet was monitored and used to control the rotating stall. A.C. control flow, which is produced by a loud speaker, was introduced into the diffuser at the inlet. It is shown that the rotating stall can be suppressed when the phase of the control flow has certain relation with the phase of the rotating stall. By considering the energy flux due to the A.C. control flow, it is shown that the rotating stall is suppressed when the control flow has the phase such that the energy is subtracted out from the diffuser flow. Discussions are made on the relations between the energy flux and the amplitude of the pressure fluctuation due to the rotating stalls.

INTRODUCTION

Rotating stalls in vaneless diffusers of centrifugal compressors set the lower flow limit of stable operation. They cause not only pressure fluctuations but also shaft vibrations [ref.(1)]. Many experiments were carried out to predict the rotating stall onset flow rate [ref.(2),(3)], and to enlarge the flow range of stable operation [ref.(4)].

Compared with experimental works, theoretical studies are few. From a 2-D inviscid irrotational flow analysis, Jansen [ref.(5)] found that outward flow is stable and hence attributed the cause of the rotating stall to local inward flow in the boundary layer on the diffuser wall. Many previous experimental works, such as ref.(2) and (3), are based on this supposition. On the other hand, based on linearized Euler's equation, Abdelhamid [(ref.(6)] and,later, Moor[ref.(7)] have shown that 2-D flow instability can occur even with outward flow. It is shown by Tsujimoto et al.[ref.(8)], by a 2-D inviscid rotational flow analysis, those flow instabilities are related with the vorticity distribution in the diffuser. Although such theoretical flow models are not fully understood, partly due to the lack of experiments based on those theoretical models.

Recently, attempts are made to apply the active controls to suppress rotating stall and surge: Paduand[ref.(9)] used "wiggling" inlet guide vanes to control rotating stall in the axial compressor and Ffowcs Williams [ref.(10)] controlled surge in the centrifugal compressor.

The purpose of the present study is twofold: to confirm the feasibility of the active stabilization of vaneless diffuser rotating stalls, and to obtain the physical understanding of the rotating stall through studying the response of rotating stall due to external disturbance. The pressure fluctuations at the diffuser inlet are monitored and used to control the stall, by introducing external fluid periodically at the diffuser inlet.

NOMENCLATURE

A.R.: ratio of pressure amplitude [Δp_d(with control)/Δp_d (without control)]
b : impeller width
E : work of control flow [eq.(1)]
G : gain
L : theoretical work of impeller
n : number of rotating stall cells
p : static pressure
q : flow rate of control flow
r : radius
T : periodic time
t : time
u : impeller peripheral speed
Δv : velocity fluctuation
β : vane angle to circumference
θ : phase angle
ρ : density of fluid
ϕ : flow coefficient [$= Q/2\pi r_2 b_2 u_2$]
ψ : pressure coefficient [$p/\rho u_2^2$]
$\Delta\psi$: pressure fluctuation coefficient [$\Delta p/\rho u_2^2$]
ω : angular frequency of pressure fluctuation
Ω : angular velocity of impeller

subscript
1,2 : impeller inlet and outlet
3,4 : diffuser inlet and outlet
a : air-chamber
d : diffuser

APPARATUS

Fig.1 shows the experimental set up. The flow is introduced into the test impeller⑤ after passing through a flow measuring nozzle①, a booster fan②, a flow control valve③ and a flow straightener④. The straightener and the inlet pressure tap are located at 4.06d and 2.22d upstream of the impeller respectively, where d=40mm is the inner diameter of the inlet pipe. The flow from the diffuser is directly discharged into ambient space. The experimental vaneless diffuser⑥ is equipped with air-chambers⑦ outside the diffuser walls. The inlet portion of the diffuser walls is made with a flat plate with pores. The air-chambers are connected to a loud speaker⑨ (diameter is 300mm) through hardened vinyl tubes⑩. "A.C. control flow", which is produced by the loud speaker, is introduced into the diffuser through the air-chamber and the porous diffuser wall.

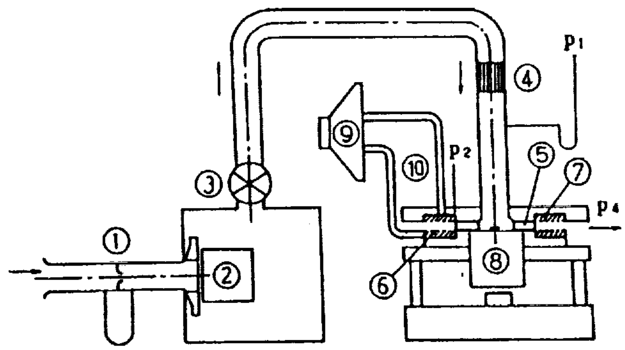
Major dimensions of test impeller are given in Table.1. The impeller is two dimensional and has logarithmic vanes, as shown in Fig.2. The rotational speed of the impeller is kept within 3,000±10rpm throughout the present study. Major dimensions of test diffuser are given in Table.2. Fig.3 shows the details around the vaneless diffuser. The air-chambers are composed of six fan-shaped rooms with the depth 5mm. The inlet portion of the diffuser wall(radius ratio $r/r_1=1.11\sim1.31$) are made of the flat plate with pores with diameter 0.55mm and pitch 1.1mm. The control flow is introduced into the diffuser through these pores.

For the control of three cells rotating stall, three of six air-chamber rooms are connected to the loud speaker, as shown in Fig.3(a), and the pores of unused air-chambers are covered with vinyl tapes. For the control of two cells rotating stall, two of the air-chamber rooms are used, as shown in Fig.3(b). As shown in Fig.3(c), twelve pressure taps with throat diameter 1mm, are evenly placed at radius ratio $r/r_1=1.04$ on front diffuser wall. In order to determine the number of rotating stall cells and cell propagation velocity ratio, two pressure transducers are inserted into two of the pressure taps.

CONTROL SYSTEM

The control system used is shown in Fig.4. The diffuser inlet pressure is monitored to detect the phase and magnitude of the rotating stall, which is fed back to the diffuser as a "A.C. control flow" after giving some time lag by the delay controller (Roland SDE-300A). The band pass filter is set to pass the frequency components $1\sim30$Hz, which includes the fundamental component of the diffuser rotating stall. "A.C. control flow" is produced by a loud speaker. The flow rate of the "A.C. control flow" cannot be measured directly under the operating condition. It is estimated from the air-chamber pressure p_a and the diffuser inlet pressure p_d using the flow coefficient determined quasi-steadily.

The "phase θ" of the air-chamber pressure p_a is defined relative to the diffuser inlet pressure p_d at the peripheral position of the center of the air-chamber. Actually, the diffuser inlet pressure p_d is monitored using a pressure tap apart from the air-chamber, in order to minimize the direct effect of the control flow. The "phase θ" of the above definition is determined by using the frequency and the propagation velocity of the rotating stall.



①. Flow measuring nozzle
②. Booster fan
③. Flow control valve
④. Straightener
⑤. Impeller
⑥. Vaneless diffuser
⑦. Air chamber
⑧. Motor
⑨. Loud speaker
⑩. Vinyl tube

Fig.1 Test apparatus

Table.1 Dimensions of impeller

Inlet radius	r_1 [mm]	25
Outlet radius	r_2 [mm]	50
Impeller width	b [mm]	10
Vane angle	β [°]	20
Number of vanes	Z	7

Table.2 Dimensions of vaneless diffuser

Inlet radius	r_3 [mm]	50.5
Outlet radius	r_4 [mm]	91
Diffuser width	b [mm]	12

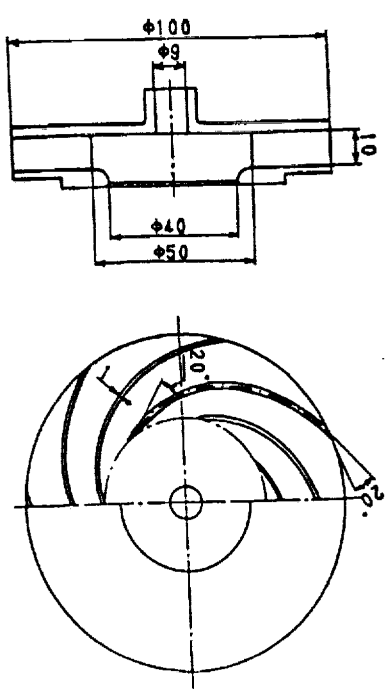
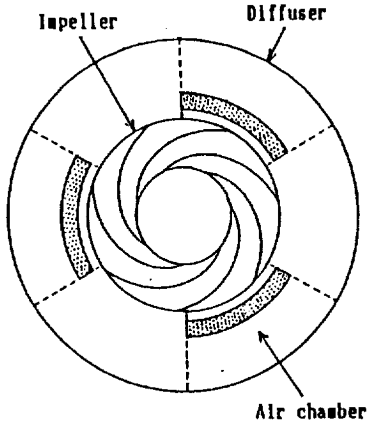
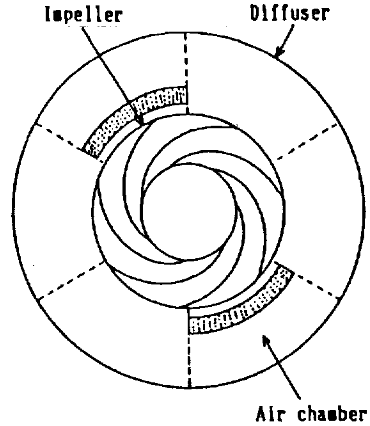


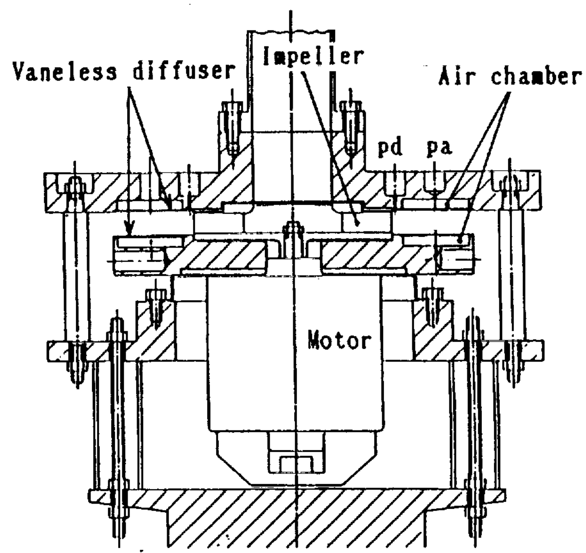
Fig.2 Impeller



(a) 3-Air chambers
(for the case of 3-cells)



(b) 2-Air chambers
(for the case of 2-cells)



(c) Sectional arrangement

Fig.3 Test apparatus (Detail around the vaneless diffuser)

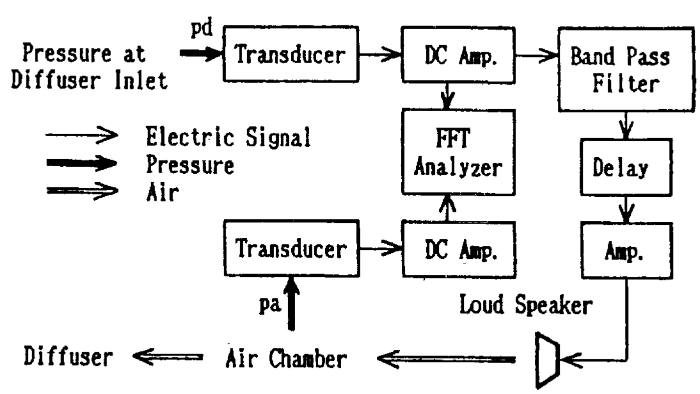


Fig.4 Control and measuring system

RESULTS WITHOUT CONTROL

Firstly, we examined the static pressure performance of the impeller and the characteristics of rotating stalls without the active control. Fig.5 shows the static pressure performance curves $p_2 - p_1$, and $p_4 - p_1$ for the cases with and without the air-chamber. Fig.6 shows the propagation velocity ratio, which are determined from the pressure fluctuations at the diffuser inlet [ref.(11)]. Rotating stalls occur at the flow rate less than $\phi = 0.11$. The number of rotating stall cells is two or three: three cells appear within the flow range of $\phi = 0.06 \sim 0.11$, and two cells in $\phi = 0.0 \sim 0.06$. Fig.7 shows the spectra of pressure fluctuations at the diffuser inlet, and velocity fluctuations at the diffuser outlet, which were measured by using the hot wire anemometer. In the present study, impeller rotating stalls were not observed. Fig.8 shows the velocity disturbance field due to the rotating stalls [same as ref.(12)], for the cases of two cells($\phi = 0.0$) and three cells ($\phi = 0.09$). Cell structure of the rotating stall can be observed clearly. Static pressure performance and the rotating stall propagation velocity ratio are shown in Figs.5,6 for the cases with and without the air-chamber. The differences are very small. The amplitudes of the pressure fluctuation are shown in Fig.9. The amplitude with the air-chamber is a little smaller than that without the air-chamber. We discuss this reason in the following section.

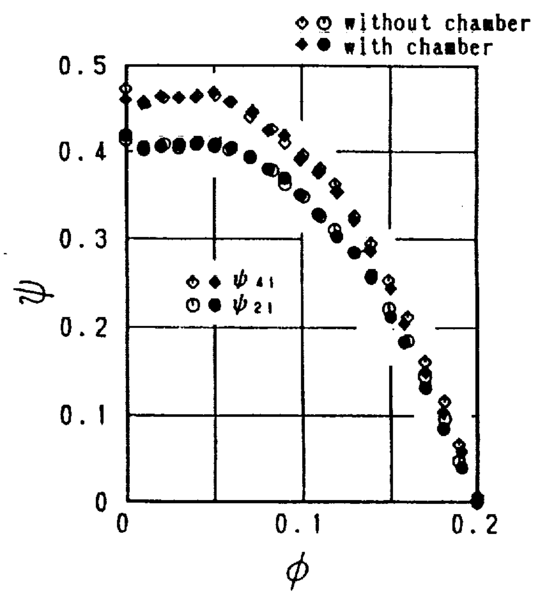


Fig.5 Static pressure performance curves (without control)

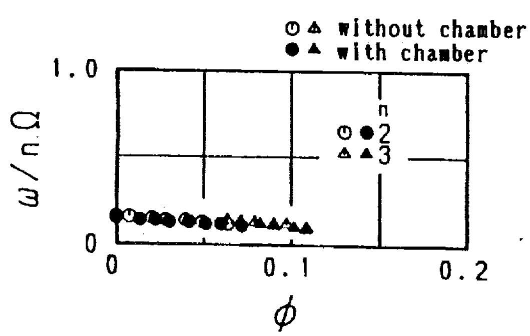
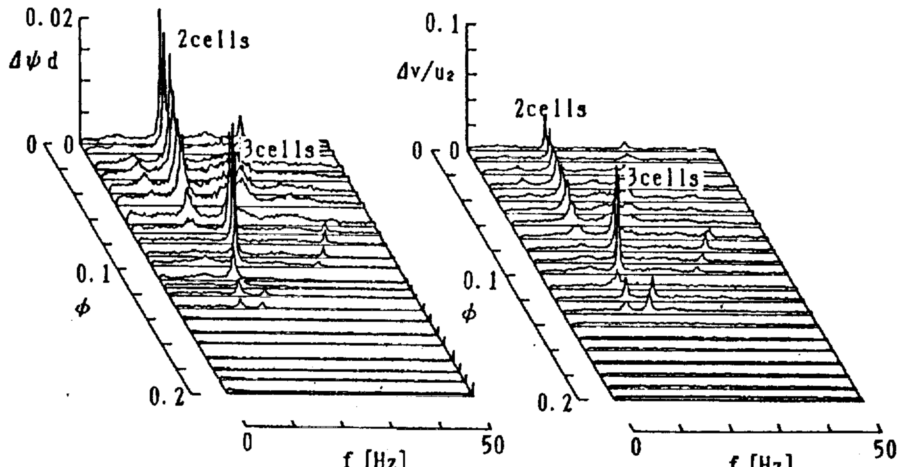


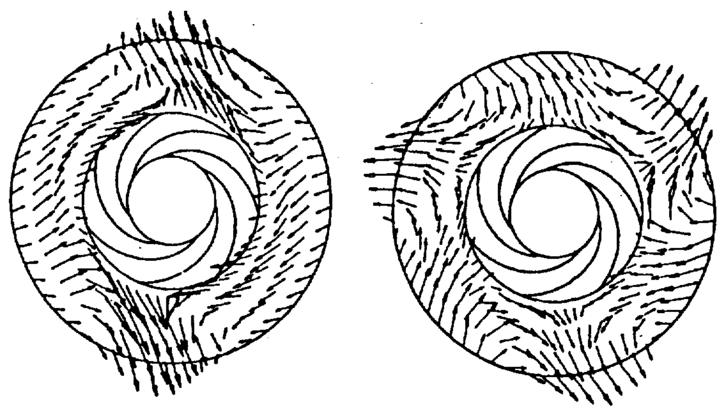
Fig.6 Rotating stall propagation velocity ratio



(a) Pressure fluctuation (at the diffuser inlet)

(b) Velocity fluctuation (at the diffuser outlet)

Fig.7 Pressure and velocity spectra(without control)



(a) ϕ=0.0

$\longrightarrow v/u_2=0.1$

(b) ϕ=0.09

$\longrightarrow v/u_2=0.1$

Fig.8 Velocity disturbance fields (without control)

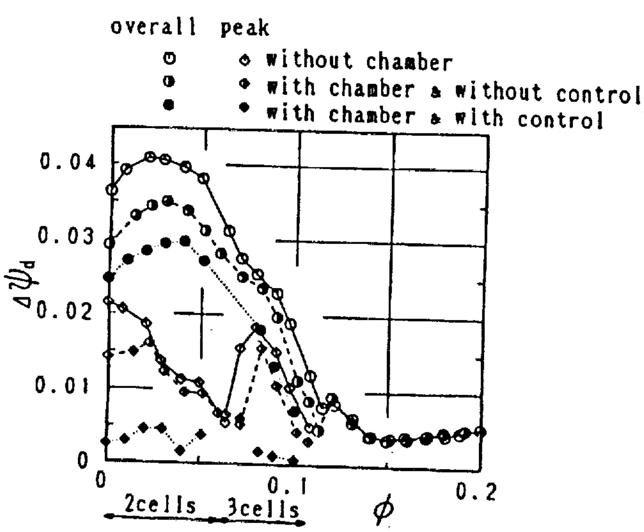


Fig.9 Comparison of the pressure fluctuation amplitudes

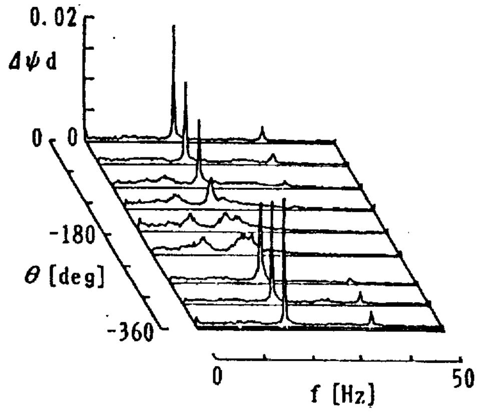
RESULTS WITH CONTROL

In this section, the effects of the active control are compared with the case with the air-chambers, but without the active control. In this study, it was found that the active control is effective for all flow region. As sample cases we describe the results at flow rate $\phi=0.09$ for the case of three rotating stall cells, and at $\phi=0.0$ for the case of two rotating stall cells.

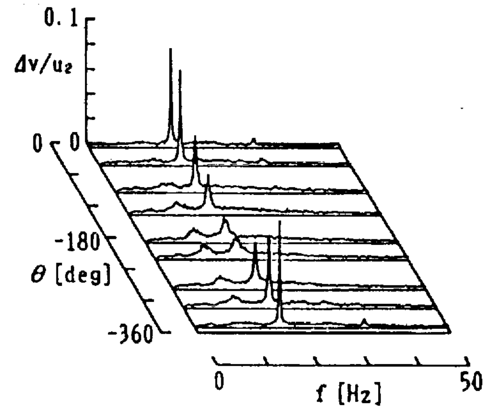
Fig.10 shows the change in the spectra of the pressure fluctuation at the diffuser inlet, and the velocity fluctuation at the diffuser outlet, when the "phase θ" is changed keeping the feed back gain constant. The "phase θ" is defined as the phase of the pressure in the air-chamber relative to the diffuser inlet pressure at the same circumferential location, as mentioned previously. When the phase is $\theta \fallingdotseq 0°$, the amplitude of the rotating stall frequency increases. However, when the phase is $\theta=-180° \sim -240°$, the rotating stall is suppressed with decreased peak value. Fig.11 shows the velocity disturbance field due to the rotating stall under the control. From the comparison with Fig.8, we find that the velocity field is controlled all over the diffuser. This means that the decrease in the inlet pressure fluctuation is not caused by the "cancelling", but caused by the control of the rotating stall itself. This is quite different from the noise control by "anti-sound". Figs.12(b),(c) show the spectra of the pressure and the velocity fluctuation at various flow rates with the control. The control is effective in all flow region. However the static pressure performance could not be improved by the control,as shown in Fig.12(a).

As mentioned above, the peak values at the rotating stall frequency could be suppressed excellently, but the over all value(1~50Hz) of the pressure fluctuation did not change remarkably, as shown in Fig.9. From the view point of over-all pressure fluctuation amplitude and of the static pressure recovery we cannot say that the diffuser rotating stall is completely controlled. This may be caused partially by the method of the control applied in the present study. That is, the delay controller gives the same absolute time delay for all the frequency components. Thus the delay time which gives the adequate phase is not adequate for other components.

Fig.13 shows the effects of the phase θ and the gain $G=20 \cdot \log_{10}(\Delta p_a/\Delta p_d)$ on the peak values of the spectrum of diffuser inlet pressure fluctuation. A.R. is the ratio of the peak value under control to that without control. A.R.is less than 1.0 for $\theta=-120° \sim -240°$. It is interesting that A.R. may become less than 1.0 even for G<0 dB, if the phase θ is appropriate.

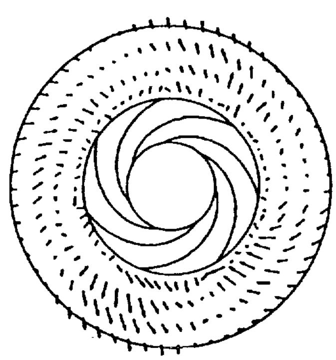


(a) Pressure fluctuation
(at the diffuser inlet)



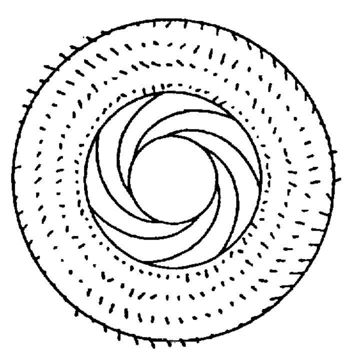
(b) Velocity fluctuation
(at the diffuser outlet)

Fig.10 Relation between the pressure, velocity spectra and the phase (ϕ=0.09)



(a) ϕ=0.0

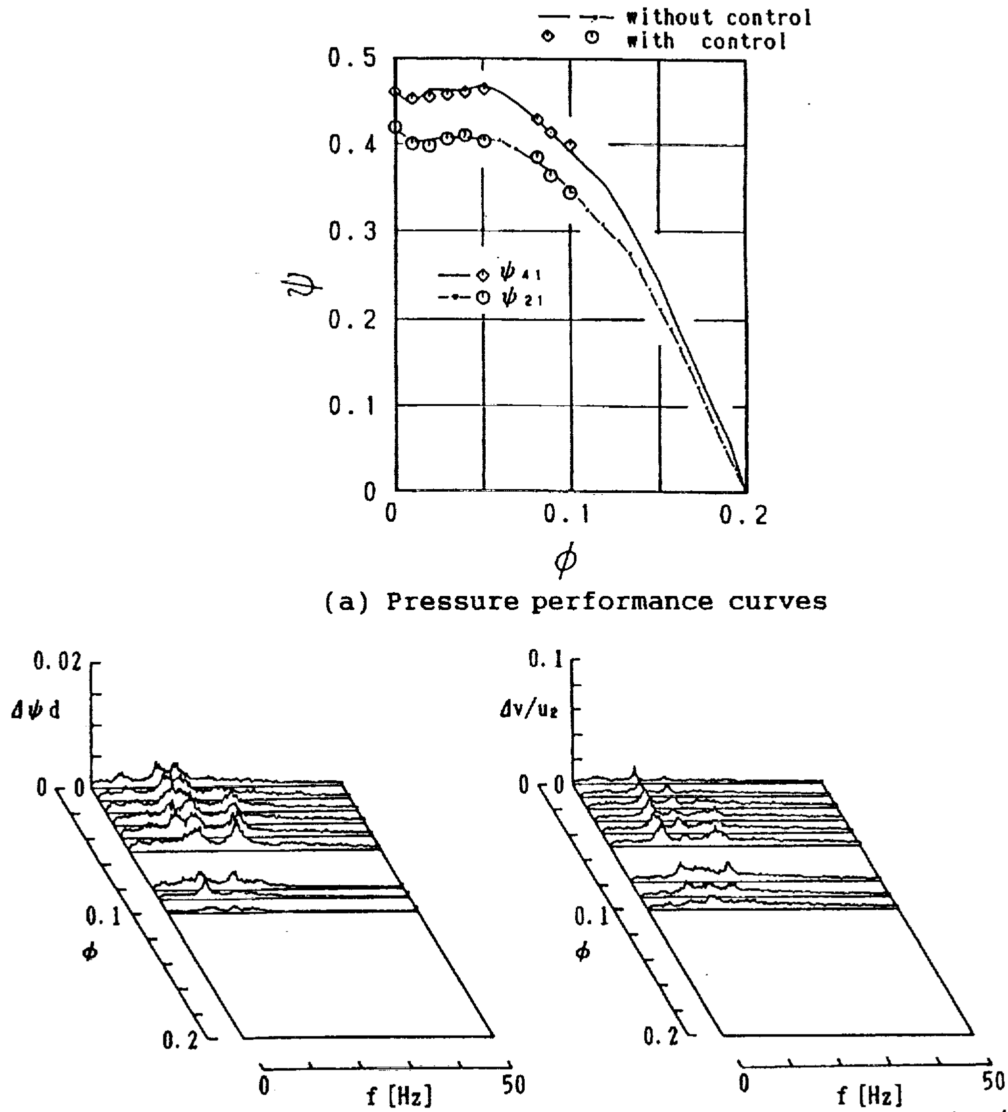
v/u_2=0.1



(b) ϕ=0.09

v/u_2=0.1

Fig.11 Velocity disturbance fields (with control)



(a) Pressure performance curves

(b) Pressure fluctuation (at the diffuser inlet)

(c) Velocity fluctuation (at the diffuser outlet)

Fig.12 Static pressure performance curves and pressure, velocity spectra(with control)

CONSIDERATION OF CONTROL ENERGY

In this section, we discuss about the mechanisms of the present control. The displacement work done by the control flow on the fluid in the diffuser is given by

$$E = \frac{1}{T}\int_0^T p_d(t) \times q(t)\, dt \qquad \text{------------- (1)}$$

where q(t) is the flow rate of the control flow, and is estimated from the diffuser(p_d) and the air-chamber(p_a) pressure. The phase between the pressure difference and the control flow rate were calibrated dynamically in advance.

In Fig.13, the region with E>0 in the gain-phase plane is shown by hatching. We find that, for the most cases, the peak values of the spectra are increased (A.R.>1) in the region with E>0. Fig.14 shows the quantitative relation between E/L and A.R., where L is the work done by the impeller. We find that the magnitude of the rotating stall can be diminished by extracting a small amount of energy from the diffuser flow, while it requires relatively large amount of energy addition to increase the magnitude. The amplitude of the rotating stall is determine so that the energy supplied to the rotating stall by a certain unknown mechanism is balanced by a energy dissipation. From this point of view, Fig.14 can be looked upon to show the amount of the energy imbalance as a function of the amplitude ratio. The fact that E<0 for A.R.<1 implies the possibility of the passive control of diffuser rotating stalls. In fact, the amplitude of the pressure fluctuation is diminished by the addition of air-chambers without the active control.[ref.(13)] It is worth trying to design passive control systems as well as sophisticating the present active system.

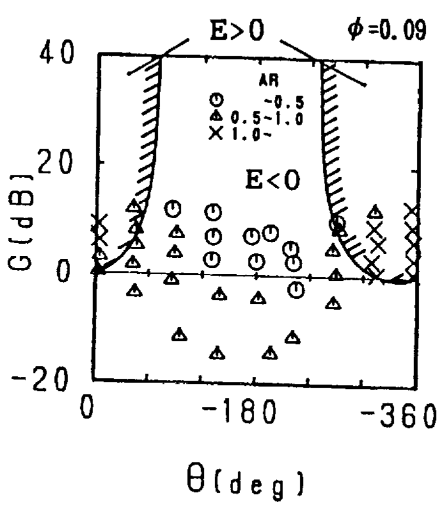


Fig.13 Effects of active control with Gain G and Phase θ

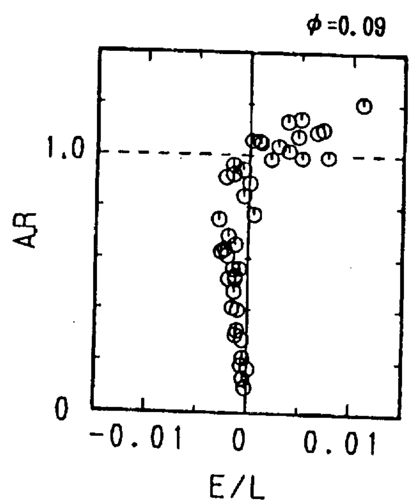


Fig.14 Relation between A.R. and E

(A.R.: ratio of the pressure amplitude
E : work of the control flow)

CONCLUSIONS

Major findings of present study are:

(1) Present active control can suppress the fundamental component of the diffuser rotating stall. The amplitude of the pressure fluctuation at the diffuser inlet can be suppressed about 1/10 of that without control. When the control is effective, the typical velocity disturbance of the rotating stall disappeared.
(2) The over all values of the pressure fluctuation could not be controlled by the present method. In this meaning, the "stall" itself could not be suppressed.
(3) It is made clear that the rotating stall is suppressed when the control flow has the phase such that the energy is subtracted out from the diffuser flow.
(4) From the relation between the energy flux of the control flow and the amplitude of the pressure fluctuation, an important information is obtained concerning the energy supply and dissipation in the vaneless diffuser rotating stall.
(5) The energy imbalance increases largely near the balancing point as the increase of the amplitude of the rotating stall.

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