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PRESSURE RECOVERY OF COLLECTORS WITH ANNULAR CURVED DIFFUSERS

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ABSTRACT

Pressure recovery is examined for axisymmetric annular curved diffusers with various values of inlet/exit area ratio. The conditions of inlet flow include three swirl angles and two values of boundary layer blockage. Then, one of the diffuser is connected to two types of collectors with different size. The discharge pressure is decreased by the collector. The cause is not deterioration of diffuser performance due to asymmetric pressure distribution but formation of a cork screw type vortex flow inside the collector. An attempt to retard the vortex motion was successful by inserting obstructions in the collector. If a large size collector is used together with proper obstructions inside the collector, pressure-rise is possible in the collector rather than pressure drop.

NOMENCLATURE

- AR : exit/inlet area ratio of diffuser
- Cp : pressure recovery coefficient of diffuser
- E1 : case of l=57mm
- E2 : case of l=157mm
- h : annular gap at inlet of diffuser
- l : length of straight annular duct at inlet of diffuser
- p : pressure
- θ : circumferential position at diffuser exit

Following letters indicate the values at the diffuser inlet

- B : boundary layer blockage $B=1-V/v(max)$
- q : dynamic pressure of V
- V : mean axial velocity
- α : ratio of real dynamic pressure and q
- β : flow angle

INTRODUCTION

At the exit of some turbomachines the annular passage is connected to an annular curved diffuser [1], which is sometimes surrounded by a collector and finally it is connected to a discharge duct. The flow out of a turbomachine has swirl and the swirl angle varies depending upon the operating condition. Although the flow in such system is quite complicated, it is important to recover the dynamic pressure of flow out of the turbomachine as much as possible regardless of the swirl angle.

At a gasturbine plant the collector and the discharge duct occupy the largest space and the gasturbine engine itself looks like a small pipe which connects the large intake duct and the large discharge duct. It is anticipated that a collector with a small cross-section is apt to distort the axisymmetric pressure distribution around the diffuser. If the highest pressure is limited by flow separation in the diffuser, the mean pressure rise across the diffuser becomes smaller as the circumferential distortion of pressure is larger. Therefore, special care has been taken on the mutual interference between the diffuser and the collector, and the collector has been made very large.

The preliminary experiment clarifys that non-axisymmetric pressure distribution does not reduce the performance of the diffuser and that the flow in the collector is by no means uniform across the cross-section and the effective area of the cross-section is not large. Therefore, in order to avoid pressure loss in the collector the cross-sectional area of the collector has to be made considerably larger than the area which is sufficient to pass uniform flow at a low velocity. If

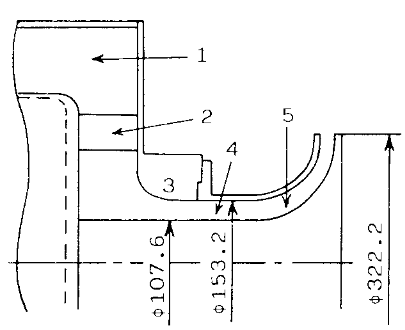


Fig.1 Experimental apparatus

the flow in the collector is made uniform, it may be possible to make a collector smaller without incurring excess pressure loss. For that purpose it is important to understand the flow behavior in the collector.

EXPERIMENTAL APPARATUS AND INSTRUMENTATION

The experimental apparatus without a collector is indicated in Fig.1. The flow rate of air is measured with a venturi flow meter and then the air is dumped into the plenum tank 1. At the exit of the plenum tank the air flows radially inward and then swirl is induced by guide vanes 2 before the axial component of velocity is increased at the annular nozzle 3. Between the annular nozzle and the curved annular diffuser 5 there is a straight annular duct 4 of variable length so that the boundary layer thickness is varied at the inlet of diffuser.

The curved annular diffuser was designed for exit/inlet area ratio of AR=2.1 for the flow without swirl[2]. The area ratio was changed in four steps by axially moving the hub body of the diffuser. The meridional geometry of these diffusers and the distribution of the cross-sectional area along the center-line of the passage m are presented in Fig.2. The hub ratio at the inlet section of the diffuser is 0.7 and the annular gap at the inlet is h=22.8mm. The length l of the straight annulus 4 is either 57mm or 157mm, and these cases are respectively expressed as E1 and E2.

After the test of the annular diffusers, the shroud diameter was reduced to 300mm and a collector S was attached around the diffuser as shown in Fig.3. The collector S was designed as small as possible hoping that it does not cause serious pressure loss. Since the diffuser system handles flow with various intensity of swirl, there is circumferential component of velocity at the diffuser exit, and the flow in the collector is not symmetric with respect to the center-line of the collector exit duct. In order to allow circumferential flow at any section of the collector, the cross-sectional area in the lower half of the collector was designed so that the mean velocity in the section was 0.272 time the diffuser inlet mean axial velocity V providing that 1/4 of the flow passes through the section. That is, the dynamic pressure is 0.074 time the dynamic pressure q of the mean inlet axial velocity V. In the upper half of the collector, the cross-sectional area of the collector is designed so that the mean velocity is constant providing that the discharge flow out of the diffuser is circumferentially uniform, and the collector wall is smoothly connected to the exit duct where the mean velocity is 0.31 V.

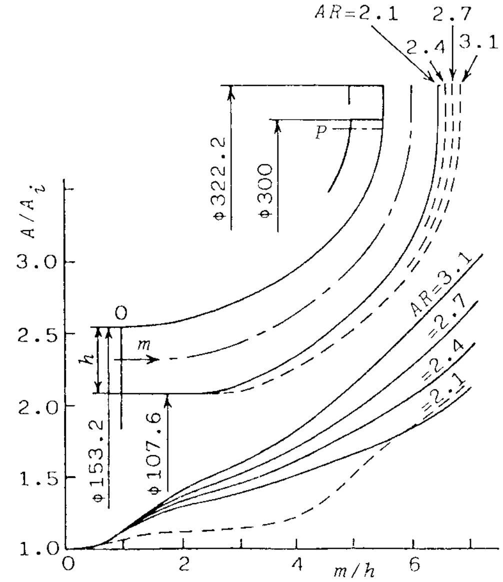
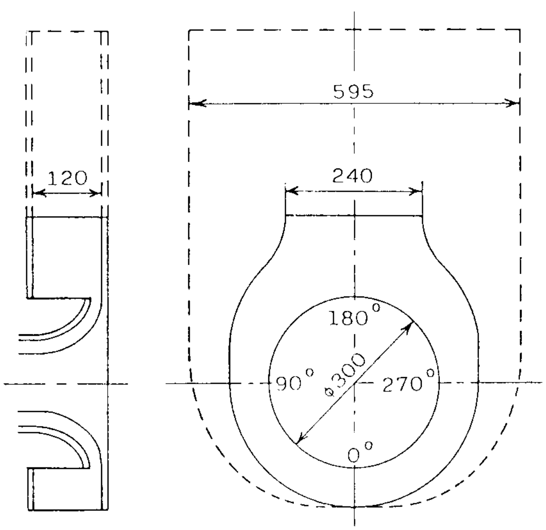


Fig.2 Meridional section of diffuser and distribution of cross-sectional area



Full line : S type collector

Dotted line : L type collector

Fig.3 Geometry of collectors

The second collector L shown as a dotted line in Fig.3 has a conventional geometry with a large cross-sectional area. The cross-sectional area at the bottom section is identical to that of collector S. The area at 45 deg from the bottom is 1.5 times the bottom area, the area at the horizontal section is twice the bottom area, and the cross-sectional area at the collector exit is 2.48 times the exit area of the smaller collector S.

The reference section of the diffuser inlet is 1.3 times h upstream from the diffuser inlet so that the pressure is uniform in the section in the case of no swirl, and four pressure taps are distributed circumferentially. The pressure recovery of the diffuser means the difference between the atmospheric pressure and the mean value of the four pressures in the reference section. When a collector was connected to the diffuser, the flow at the diffuser exit was not axisymmetric, and the circumferential distribution of pressure near the diffuser exit was measured at 24 pressure taps at the radial position P indicated in Fig.2.

EXPERIMENTAL RESULTS FOR CURVED DIFFUSERS

Flow at the Inlet Section

The intensity of swirl is expressed as the mean swirl angle $\bar{\beta}$ defined as follows:

$$\bar{\beta}=\tan^{-1}\{\int uvr^2dr/(\bar{r}\int v^2rdr)\} \qquad (1)$$

where v and u are respectively the axial and the circumferential component of velocity at r and $\bar{r}$ is the root mean square radius at the inlet section.

An example of velocity distribution at the inlet

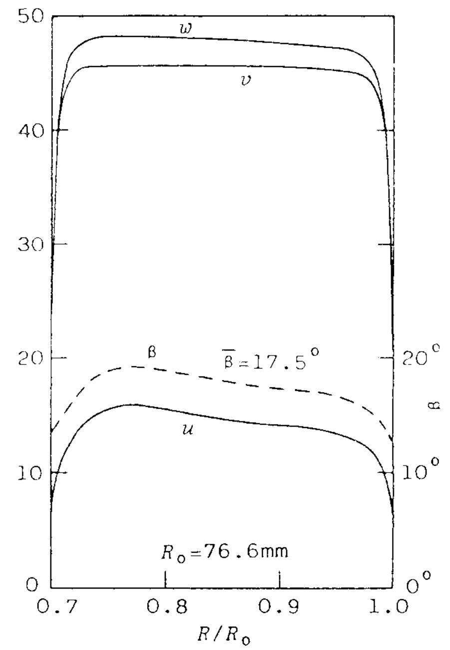


Fig.4 Inlet velocity distribution, E1 17 condition

section is presented in Fig.4 where the inlet straight duct is the short one E1. It is noticed that the mean swirl angle $\bar{\beta}$ almost agrees with the flow angle β at the radius $\bar{r}$. The distribution was hardly influenced by the area ratio of the downstream diffuser.

The boundary layer thickness at the inlet section is expressed by the blockage factor. In cases of zero swirl, the blockage factor B=1-V/v(max) was 0.022 for l=57mm and it was 0.040 for l=157mm. For a given setting angle of guide vanes, the mean swirl angle β at the reference section was decreased a little by increasing the length l due to the wall friction.

α is the ratio of the mass-averaged dynamic pressure at the inlet section to the dynamic pressure q of the mean axial velocity V . When the inlet duct was l=57mm, α was 1.02, 1.11 and 2.24 respectively for the swirl angle $\bar{\beta}$=0, 17.5 and 47.3 deg. These conditions are expressed as E1 0, E1 17 and E1 47. When the inlet duct was l=157mm, $\bar{\beta}$ was 0, 17.2 and 46.3 deg respectively for the three setting angles of guide vanes, and α was 1.02, 1.09 and 2.11 respectively. These conditions are expressed as E2 0, E2 17 and E2 46. All experiments were made at Reynolds number 2hV/ν=7.7 ~ 14x10 .

Diffuser Performance for Non-swirl Flow

The pressure recovery coefficient Cp is the ratio of the pressure rise in the diffuser to the mass averaged dynamic pressure at the inlet section. During experiment the flow rate was monitored all the time and the mass averaged dynamic pressure was calculated as the product of q and α assuming that α remained constant for a little change of the flow rate, and α was decided beforehand for all six combinations of the two inlet annular duct lengths and the three vane setting angles.

Variation of Cp with respect to the exit/inlet area ratio of diffuser is presented in Fig.5 for various conditions of inlet flow. Comparing curves E1 0 and E2 0 which respectively pass through open circles, it is noticed that the pressure recovery coefficient is very sensitive to the blockage factor at the inlet section.

This annular curved diffuser was designed for AR=2.1 for swirl free flow $\bar{\beta}$=0, and the length of the diffuser center line m is 7.0 times h. This combination of diffuser length and area ratio is close to the optimum one for straight wall annular diffusers [3], and the pressure recovery coefficient for the case of E1 0 is also close to that of the optimum straight wall annular diffuser. It can be said that the design of the present diffuser is successful, because it turns the flow from axial to radial direction without additional pressure loss.

In the present experiments, the pressure recovery coefficient was increased a little as the area ratio was increased up to 2.7, although in diffusers which have the area ratio larger than 2.4 tufts near the diffuser exit demonstrated that reverse flow occured occasionally and that the flow was unstable.

According to the literature [3] the pressure recovery coefficient of straight wall diffusers can be estimated by the following equation providing that the diffuser geometry satisfys the relationship expressed as Cp*-line, or the optimum diffuser length is secured for a given area ratio.

$$Cp = \frac{1}{E_i^2}\{1-(\frac{E_i}{E_e}\frac{1}{AR})^2\} \qquad (2)$$

where E=1-B. Subscript i and e represent the inlet and exit conditions, and E is related to the exit/inlet area ratio AR and E_i [3] for the case of straight wall diffusers.

In the present case, the blockage factor Bi due to the inlet boundary layer was evaluated from the flow rate and the pressure drop from the inlet plenum chamber to the inlet reference section, and it was found that Ei=0.978 for E1 0 and Ei=0.960 for E2 0, consequently it was estimated by Eq.(2) that Cp was 0.63 and 0.60 respectively. Comparing these values of Cp with the experimental data in Fig.5, it is noticed that in the present case the pressure recovery is deteriorated by the boundary layer blockage at the inlet section more severely than the prediction based on the straight wall diffusers. In the present case local pressure gradient along the diffuser wall is large due to the curvature, and the thick boundary layer and the large value of shape factor close to the value of separation may be the

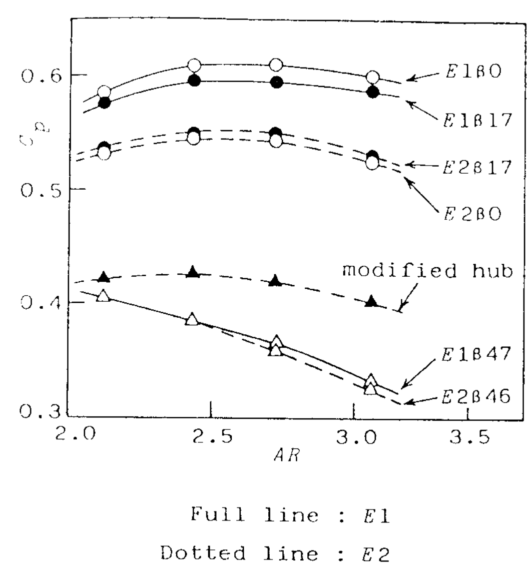


Fig.5 Pressure recovery coefficient influenced by exit/inlet area ratio and inlet conditions of flow

cause of the large reduction of pressure recovery for the case of thick inlet boundary layer.

Diffuser Performance for Swirl Flow

In Fig.5 the curves E1β17 and E2β17 which connect closed circles represent the relationship between the exit/inlet area ratio and the pressure recovery coefficient for the swirl flow β=17 deg. The relationship is not much different from that for the non-swirl flow. In the present case deterioration of Cp due to the longer inlet annular duct is not so large as that for the case of non-swirl flow. It should be noted that the increment of blockage factor B due to the longer inlet annular duct is less as the intensity of swirl is increased.

When the inlet swirl was strong, reverse flow was observed along the hub at the middle of the diffuser length, and the layer of reverse flow zone became thicker and longer and the pressure recovery coefficient was smaller as the exit/inlet area ratio was larger. The relationship is presented as open triangles in Fig.5. Modification of the hub geometry by filling up the reverse flow zone improved the pressure recovery considerably as shown by closed triangles which are also indicated as "modified hub" in Fig.5. The pressure recovery of the modified diffuser was poor for the non-swirl flow, because the cross-sectional area of the diffuser increased quickly near the exit as shown in Fig.2 by the dotted line and the steep pressure gradient separated flow from the shroud wall.

Effect of Diffuser Exit Diameter on Pressure Recovery

When the diffuser was tested with collectors, it was necessary to change the exit diameter of diffuser in three steps. The performance of these diffusers with different exit diameters was tested without the collector. The meridional geometry of these three diffusers A, B, and C is illustrated in Fig.6. The passage width at the diffuser exit is 19.6mm for all three cases, and the exit/inlet area ratio would be 2.1 if the diameter of the diffuser exit was 322.2mm as discussed in the last section.

The experiment was made with the long inlet annular duct E2, and the results are presented in Fig.7 as lines A, B and C, where the abscissa is the mean inlet flow angle while the ordinate is the apparent pressure recovery coefficient αCp. The pressure recovery is made dimensionless by comparing with the dynamic pressure q of the mean inlet axial velocity V. In the case of non-swirl flow, the increment of αCp due to the increase of exit radius is a little more than the change of the mean

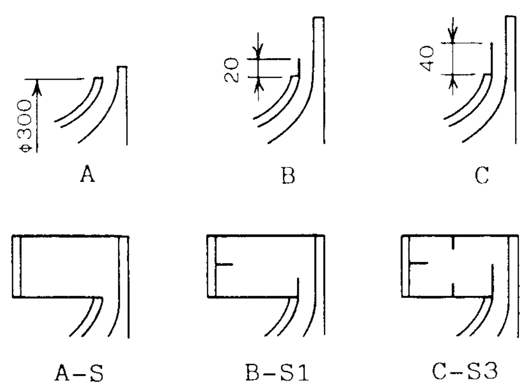


Fig.6 Meridional section of diffusers and the smaller collector

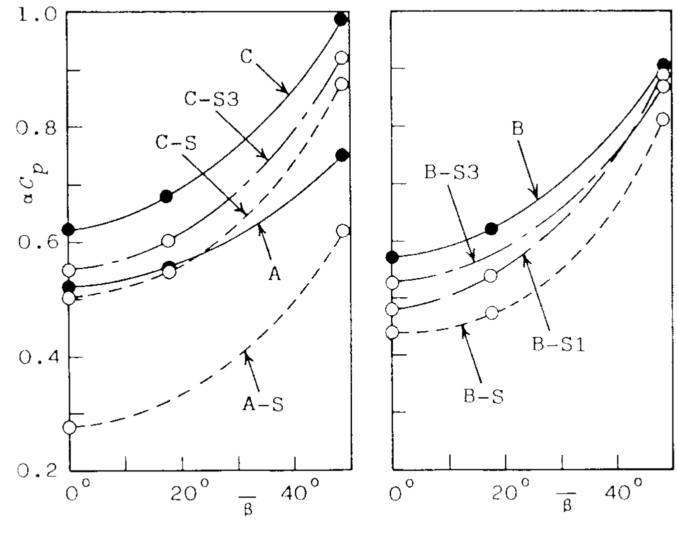


Fig.7 Virtual pressure recovery coefficient of diffusers with the smaller collector

dynamic pressure at the diffuser exit.

The increment of αCp due to the increase of the exit diameter becomes larger as β is larger. This is mostly due to the pressure recovery of the swirl velocity, and the increment of the net pressure recovery coefficient Cp is not much influenced by the intensity of swirl β.

DIFFUSERS WITH COLLECTORS

Performance of Diffusers with a Small Collector

Three diffusers A,B and C mentioned in the last section were surrounded by a small collector S shown in Fig. 3. These three systems are identified as A-S, B-S and C-S respectively. As a preliminary experiment, a short duct was connected downstream to the exit of collector and the pressure recovery of the diffuser system was compared with the original one. Since no change in the pressure recovery was recognized, in the following experiment there was no duct downstream the collector.

Comparing A-S line and A line in Fig.7 it is clear that the collector deteriorated pressure recovery considerably.

For the inlet condition of E2β0, the circumferential pressure distribution at the P radial position near the diffuser exit is presented in Fig.8 together with the collector exit pressure. The abscissa is the circumferential position, where 180 deg means the top of the collector which coincides with the center line of the exit duct. The mean value of the circumferentially

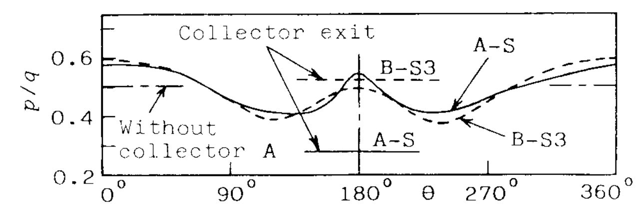


Fig.8 Circumferential pressure distribution at the radial position P, with the small collector, E2β0 condition

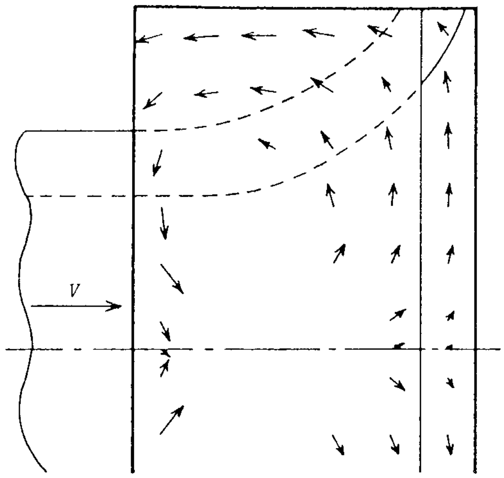


Fig.9 Secondary flow at the exit of the smaller collector A-S system, E2θ0 condition

distributed pressure is not much lower than the chain line which is the pressure distribution without the collector.

When there was no collector, the pressure rise from the P radial position to the exit of A diffuser was 0.017q. In the case of A-S system the collector exit pressure was very low as shown in Fig.8. That is, there was a large pressure drop of 0.27q from the top of diffuser to the exit of collector. Such a large pressure drop is inconceivable for simple ordinary flow. It is presumed that the flow was spiral in any section of the collector and as a result the pressure was low at the center of the cross-section, and the low pressure zone is connected to the collector exit.

The secondary flow at the collector exit section is demonstrated in Fig.9 where the openning of the diffuser exit appears at the right end. The arrows indicate the velocity vector of the secondary flow and the velocity vector at the diffuser inlet is also presented for comparison. The zone without arrows is the area where the flow was reversed and unsteady.

According to Fig.7, deterioration of αCp due to the collector S is 0.243 for diffuser A, and it is only 0.133 for diffuser B at =0 deg. It is presumed that the fence at the exit of diffuser not only increased the pressure recovery of the diffuser, but also it changed the spiral flow pattern in the collector considerably, i.e. the secondary flow in the collector was separated by the fence from the radial jet flow out of the diffuser, consequently the secondary flow was not so much energized by the jet flow as in the case of A-S system.

In order to reduce the secondary flow in the collector further, another fence of 20mm high was attached to the collector as shown in Fig.6 B-S1. Later, two more fences of 15mm high were added as shown in Fig.6 C-S3. The apparent pressure recovery coefficients αCp of these collector systems are presented in Fig.7. It is clear that these fences reduced the secondary flow in the collector, and the pressure drop in the collector was only 0.045 in the case of B-S3 system. These fences were effective to reduce the pressure loss in the collector regardless of the intensity of inlet swirl.

Comparing the circumferential pressure distributions at the P radial position for A-S and B-S3 systems in Fig.8, it is clear that the flow in the diffuser was hardly influenced by the fences in the collector, however the pressure drop in the collector was drastically reduced by the fences. Even in the case of B-S3 system the flow pattern at the exit of collector showed secondary flow similar to Fig.9, but there was no reverse flow zone. That is, secondary flow was weak at any cross-section of the collector and the pressure at the center of cross-section was not low.

The remarkable difference in αCp between B-S system and A-S system suggests significance of the fence portion of shroud which is sticking in the collector. In order to examine the effect of the fence further, the height of the fence was increased to 40mm and it is identified as C-S system.

In the case of non-swirl flow, the difference between αCp of C-S system and of B-S system was 0.066 which was a little larger than the difference between αCp of C system and of B system. That is, when the collector was attached to the diffuser, the flow pattern in the collector was improved by the high fence. The fence in the collector does not increase the overall size of the collector. Therefore, it is recommended that the exit diameter of diffuser should be made as large as possible within the collector unless the gap between the outer wall of the collector and the diffuser exit diameter is too small to freely pass the flow from the diffuser to the collector.

According to Fig.8, the circumferential variation of pressure is about 0.2q. Since the mean dynamic pressure at the diffuser exit was about 0.23q, it is clear that the major part of the flow was discharged from the upper half of the diffuser. That is, in the low pressure zone in Fig.8, from θ=60 deg to 300 deg, the cross-sectional area was not large enough as a good collector.

The circumferential distribution of pressure is presented in Fig.10 for three values of inlet swirl angle. The symmetric pressure distribution was distroyed by swirl, but the location of the minimum pressure did not move much and it was located at θ=100~130 deg, therefore the cross-sectional area of the collector should be made larger in this region. Since the circumferential variation of the cross-sectional area should be smooth, the entire collector should be made larger except the lowest part.

Performance of Diffusers With a Large Collector

In order to reduce the circumferential distortion of pressure distribution and to reduce the pressure drop in the collector, the collector was made larger as shown by the dotted line in Fig 2. The experiment was made with the long inlet annular duct E2 and the pressure recovery coefficient is presented in Fig 11 for various systems with the large collector L and for various values of the inlet swirl angle.

It is noticed that A-L line is not much lower than A line. That is, when the collector was large, the pressure drop in the collector was not large. In cases of B-L and C-L systems, there was pressure rise in the collector. That is, the height of fence at the exit of diffuser was quite important to reduce the secondary flow and the pressure drop in the collector. By adding three

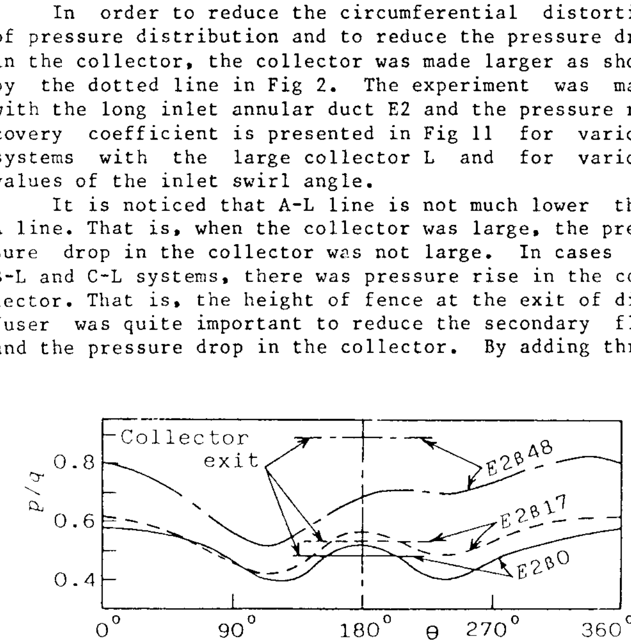


Fig.10 Circumferential pressure distribution at the radial position P, B-S1 system

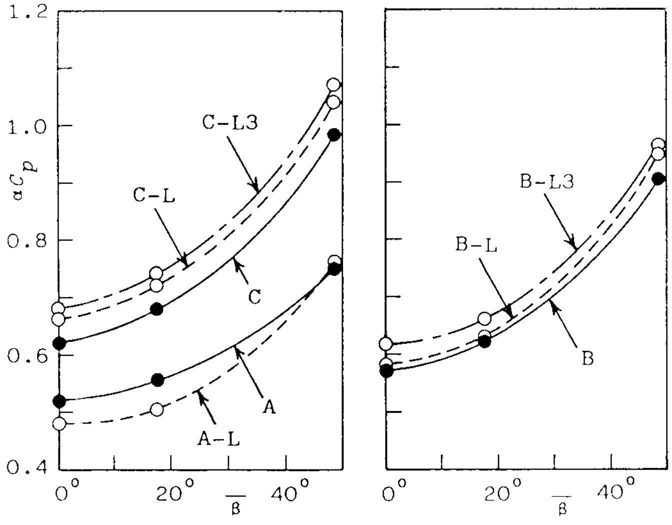


Fig.11 Virtual pressure recovery coefficient of diffusers with the larger collector

more fences in the collector, the pressure rise in the collector was increased further regardless of the inlet swirl angle. Therefore, the secondary flow in the collector was not negligibly small even in C-L system. The circumferential distribution of pressure at the P radial position near the exit of diffuser is presented in Fig 12 for diffusers with the large collector. The pressure at the exit of collector is indicated as the horizontal line at θ=180 deg. In the case of A-L system the pressure distribution was wavy and there was a pressure drop of 0.02q from the diffuser exit to the collector exit as shown by the full line. In the case of B-L3 system the pressure distribution was smooth and pressure was rising from the bottom of the collector to the exit of collector and there was a considerable pressure rise from the P position to the collector exit along the radial line of θ=180 deg. Such pressure distribution is reasonable for ordinary flow, that is, it is presumed that the secondary flow was weak in this case. It should be noted that the minimum pressure is observed at the bottom section and the flow rate in the lower half of the diffuser is larger than the flow rate in the upper half. Apparently the collector serves as a kind of diffuser. This distribution is contrary to that in the case with the small collector, where the maximum pressure was observed at the bottom of collector as shown in Fig 8.

The circumferential distributions of pressure for different inlet swirl angles are presented in Fig.13. Because of the large cross-sectional area of the collector, the circumferential variation of pressure is not large. The pressure-rise from the diffuser exit to the collector exit is larger as the inlet swirl angle is larger for a given value of q or flow rate, because the dynamic pressure in the diffuser and the collector is larger.

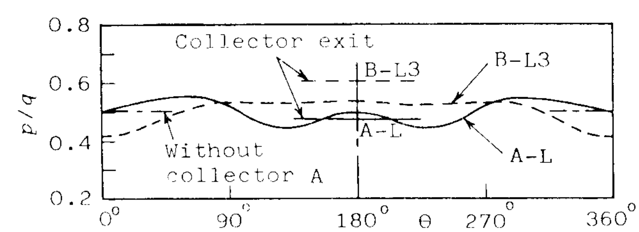


Fig.12 Circumferential pressure distribution at the radial position P, with the larger collector, E2β0 condition

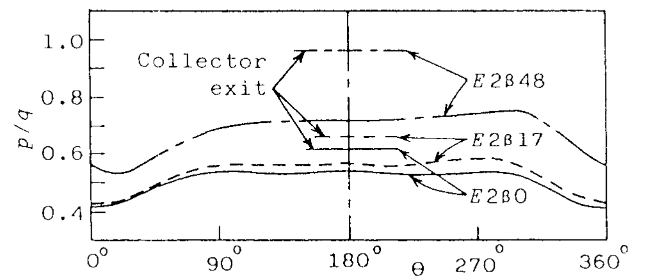


Fig.13 Circumferential pressure distribution at the radial position P, B-L3 system

CONCLUSIONS

(1) An annular curved diffuser can achieve a pressure recovery coefficient which is comparable to the optimum annular straight diffuser with the same exit/inlet area ratio by proper design of diffuser length and curvature.
(2) Deterioration of diffuser performance due to the inlet boundary layer thickness is larger for the curved diffuser than that for ordinary diffusers.
(3) The circumferential distribution of pressure is distorted by a collector around the diffuser, but the mean value is not much lower than the uniform value without a collector.
(4) The flow in the collector is spiral due to secondary flow which reduces the discharge pressure at the exit of collector. The effect is larger for the smaller collector.
(5) In order to reduce the pressure drop in the collector it is important that a certain length of shroud is sticking into the collector.
(6) A considerable pressure rise is possible in the collector even when the inlet flow has strong swirl, providing that the collector has proper geometry and dimensions.
(7) Pressure-rise in a collector is improved by reducing the secondary flow in the collector by adding a few fences circumferentially.

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