Effects of Swirling Inlet Flow on Pressure Recovery in Conical Diffusers

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An experimental investigation was conducted to determine the effect of swirling inlet flow on the performance and outlet flow profile of conical diffusers. Twenty four different diffusers were tested, with total divergence angles ranging from 4.0 to 31.2°, and with area ratios from 1.30 to 8.27. The effect of swirling inlet flow on diffuser performance was found to be a strong function of flow regime in the same diffuser with axial inlet flow. Swirling inlet flow did not affect performance of diffusers which were unseparated or only slightly separated with axial inlet flow. For diffusers which were moderately or badly separated for axial inlet flow, swirling inlet flow caused large performance increases based on total inlet kinetic energy. The results indicate that optimum diffuser performance for swirling inlet flow may be higher than that for axial inlet flow. However, the geometry of the optimum diffuser will differ considerably from that for axial inlet flow. New optimums are presented for the three swirl ratios investigated.

Nomenclature

A = area
AR = diffuser area ratio = A_2/A_1
C_{PR} = one-dimensional diffuser performance coefficient, Eq. (1)
C_{PR_i} = ideal diffuser performance coefficient, Eq. (2)
C_{PRS} = diffuser performance coefficient based on area average, Eq. (3)
L = axial length of diffuser
p = static pressure
r = radial coordinate in cylindrical coordinate system
R = local radius of conical diffuser
U = total velocity magnitude
V = axial velocity magnitude
z = axial distance from diffuser inlet
η_p = diffuser efficiency defined by Peters, Eq. (4)
θ = angular coordinate of cylindrical coordinate system
ϕ = diffuser half angle
$(\overline{})$ = area average

Subscripts

1 = diffuser inlet conditions
2 = diffuser exit conditions

I. Introduction

Diffuser performance is a strong function of many flow variables as well as geometric variables. In seeking an optimum diffuser for a given situation, the design engineer must be able to account for the effects of each variable. For uniform axial inlet flows, the effect of geometry on conical diffuser performance is fairly well established.[1,2] The effects of inlet boundary-layer thickness and of simple axial profile distortions have been established for plane-wall diffusers.[3] However, for conical and annular units, and for more complex inlet flow distortions, little design information exists.

An important practical case of inlet flow distortion is that of swirling inlet flow. Many times (e.g. in flow downstream from a pump or hydraulic turbine), some swirl will be present, and the designer must either try to remove it or live with its consequences, for few guidelines are available. The objective of this work was to help provide basic design information for diffusers with incompressible swirling inlet flow.

II. Evaluation of Diffuser Performance

The most common parameter used for rating diffusers is the performance, defined as the static pressure rise through the diffuser normalized on the inlet dynamic head. For one-dimensional flow the performance may be written

$$C_{PR} = (p_2 - p_1)/\tfrac{1}{2}\rho V_1^2 \qquad (1)$$

where subscripts 1 and 2 refer to diffuser inlet and outlet conditions, respectively. The one-dimensional relation is often used even for viscous flows, in cases where there is a thin wall boundary layer and a central, potential or uniform core flow.

For lossless flow, the Bernoulli equation may be used to determine the pressure rise in terms of the velocities at inlet and outlet. By applying continuity, the outlet velocity may be eliminated in favor of the area ratio. The result is the ideal performance

$$C_{PR_i} = 1 - 1/(AR)^2 \qquad (2)$$

Equation (1) often is applied to viscous flows with a potential core. The slight errors in average values which are due to the wall boundary layers are outweighed by the simplicity in applying the equation. However, if the velocity and/or pressure profiles are significantly nonuniform at a diffuser cross section (as is likely for swirling flow), Eq. (1) should not be applied. In the present work, diffuser performance is

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defined by the area average

$$C_{PRS} = \frac{1/A_2 \int p_2 dA_2 - 1/A_1 \int p_1 dA_1}{\frac{1}{2}\rho(1/A_1) \int U_1{}^2 dA_1} \qquad (3)$$

where U_1 is the total velocity at Sec. 1, i.e. the vector sum of the velocity components. Equation (3) is an obvious extension of Eq. (1). However, note that the denominator in Eq. (3) includes the kinetic energy in the swirling flow component as well as that in the axial component.

C_{PRS} is not the only performance parameter which can be defined for swirling inlet flow. Peters[4] was the first to study flow with swirl in conical diffusers. His results were presented in terms of an "efficiency" defined using massflow-weighted averages as

$$\eta_p = \frac{1/A_2 \int p_2 V_2 dA_2 - 1/A_1 \int p_1 V_1 dA_1}{\frac{1}{2}\rho[1/A_1 \int U_1{}^2 V_1 dA_1 - 1/A_2 \int U_2{}^2 V_2 dA_2]} \qquad (4)$$

Peters tested diffusers with total divergence angles between 5.2 and 180°, at an area ratio of 2.5. He reported an increase in η_p with increasing inlet swirl.

Peters' flow system consisted of a fixed swirl generator, then a variable length of pipe, followed by the conical diffuser. The maximum swirl was obtained with the shortest connecting pipe. Unfortunately the shortest pipe also produced the thinnest inlet boundary layer at the diffuser. Thus it is impossible to determine if the increase in η_p reported by Peters was due to increased swirl or to decreased inlet boundary-layer thickness.

More recently, Srinath[5] reported beneficial effects from swirl for annular diffusers with fully developed inlet flow. Thus the criticism of Peters' data cannot be levied against the work of Srinath. Moreover, the latter found that recovery was maximized for swirl angle (measured in a meridional plane) equal to the total divergence angle of the annular walls (both walls had equal divergence angles in this case). The swirl angle was essentially constant across the diffuser inlet.

The two studies just mentioned indicate that swirling flow at the inlet can change the recovery characteristics of a given diffuser. The data of Srinath suggest that the change may be for the better. Such effects further suggest that the performance map as a function of diffuser geometry may be affected significantly. To evaluate this change, systematic data must be obtained to permit the definition of optimum performance when swirl is present. Such data will also provide a useful design tool.

III. Test Facility and Procedure

The layout of the suction wind tunnel used for the experimental work is shown in Fig. 1. Air entered through the nozzle at the left. The next section was filled with 0.25-in. hexagonal-cross-section honeycomb with a 0.003 in. wall thickness. This section could be spun at various speeds to produce essentially solid-body swirl. After passing through the swirl generator, the flow went through another nozzle with an area contraction of 9 to 1, then through a 6-in. long straight section before entering the 2-in. inlet diameter diffuser.

The complication of data analysis for swirling flow is illustrated in the differences between Eqs. (1) and (3). For axial inlet flow, static pressure may be assumed constant across any diffuser cross section. With a thin inlet boundary layer, V_1 is nearly constant and all the terms in Eq. (1) can be evaluated from wall static pressure measurements. With swirling flow, neither pressure nor velocity is constant so that traverses must be made in order to evaluate area averages of velocity and pressure.

Two types of measurements were made during the tests, all of which were run at a Reynolds number (based on inlet properties) of approximately 1.5×10^5. For axial inlet flow, boundary-layer velocity profiles were measured at numerous axial stations. Complete details have been reported in Ref. 6, and a summary of the results has been submitted for publication.[7] Also measured were profiles of the axial and swirl components of velocity and static pressure at the diffuser inlet and outlet cross-sections.

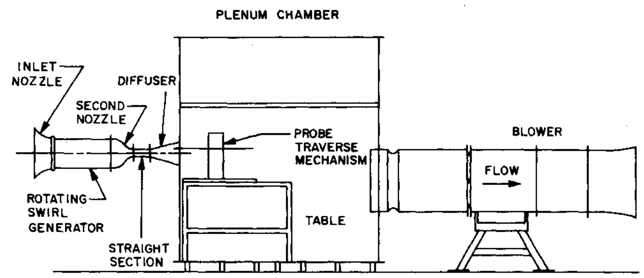

Fig. 1 Elevation view of diffuser test wind tunnel.

All measurements for swirling flow were made using a commercial 5-hole pressure probe,§ similar in principle to that described by Lee and Ash.[8] The probe was carefully calibrated before use over the range of velocities expected in operation, and for yaw angles as large as 45°. After calibration curves were obtained, a polynomial curve was fitted by computer for each flow quadrant. A program was written for reduction of data, based on the scheme suggested by Hale and Norrie.[9] The complete procedure is described in Ref. 6.

After calibration was completed, the probe was installed in the flow system. Alignment was verified aerodynamically by installing a special calibration jet in place of the second contraction nozzle. The estimated resolving ability for mean velocity direction is ±0.2°.

The probe axis was parallel to the axis of the diffuser. Traverses were made in both horizontal and vertical planes to check for three-dimensional effects. The flow was found to be symmetric within experimental error for all unstalled cases.

The rotating honeycomb swirl generator produced diffuser inlet velocity profiles very close to a solid body rotation superimposed on a uniform axial velocity. For $r/R < 0.8$, the tangential velocity profiles were essentially linear functions of the radius, as shown in Fig. 2. There were no significant deviations from a uniform axial velocity even for the highest swirl speed.

IV. Results

Diffuser performance coefficients were calculated from Eq. (3) by averaging the diametral traverses for each run. The re-

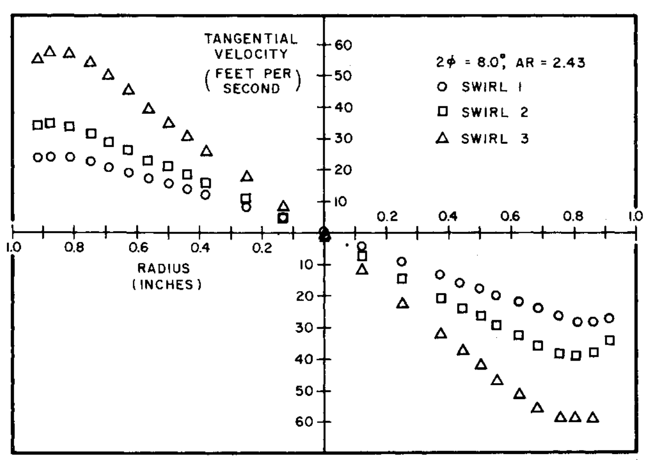

Fig. 2 Typical profiles of tangential velocity at diffuser inlet.

§ Model DC-125-36-F-32-CD, United Sensor and Control Corp., 89-91 Church St., East Hartford, Conn. 06108.

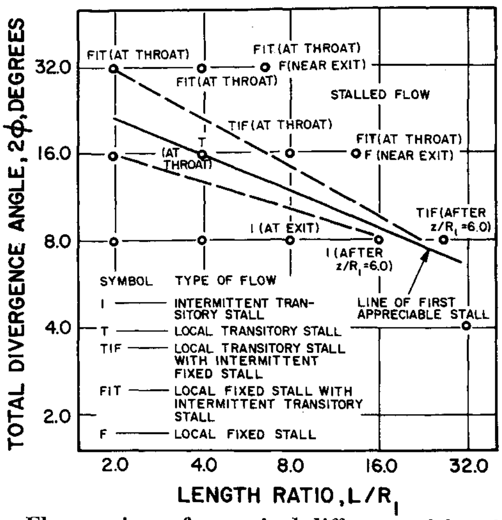

Fig. 3 Flow regimes for conical diffusers with axial inlet flow, as observed by McDonald and Fox.[1]

sults for swirl ratios 1, 2, and 3 are compared with data for axial inlet flow in Table 1.

Swirl did not affect the performance of some diffusers, but caused a rather large increase in performance for others. The increase in performance with swirl can be correlated with the flow regime in axial flow using the information contained in Fig. 3. Each diffuser showing an increase in performance in the present study was previously determined to exhibit some degree of separation with axial inlet flow. The most dramatic performance increases with swirl took place in diffusers which contained a considerable degree of separation (e.g. the higher area ratio 12 and 15.8° diffusers). The only exception to this generalization is in the 31.2° diffusers; in these diffusers swirl had no beneficial effect on performance except for the highest degree of swirl. In fact, there is a small loss of performance at lower swirl values. It appears that more swirl would be necessary to reduce or eliminate the very strong, stable separation present. However, higher degrees of swirl cause complications that will be considered later.

There is other evidence that reduction of separation due to swirl caused the performance to increase. The ideal and one-dimensional performance coefficients, Eqs. (1) and (2), are plotted in Fig. 4 for no swirl. The one-dimensional values fall well below the C_{PR_i} line and show considerable spread. That is, the highest value of C_{PR} is about 0.81 and the width of the envelope of C_{PR} values for all but the $2\phi = 31.2°$ diffusers is about 0.20. This implies that for the given inlet conditions C_{PR} is strongly a function of something other than the area ratio, i.e. that performance is affected by separation.

Figure 5 is a similar plot for swirl 3. In contrast to the data for no swirl, the shaded area of the envelope has been

Table 1 Performance coefficient values for conical diffusers with and without swirl

Area Ratio, AR	Total Divergence Angle, 2φ, degrees: 4.0	6.0	8.0	12.0	15.8	31.2
1.30	0.35 0.38 0.38 0.38	0.36 0.33 0.35 0.34	0.38 0.39 0.41 0.40			
1.64	0.50 0.52 0.53 0.53	0.51 0.52 0.59 0.54	0.51 0.52 0.53 0.52	0.47 0.48 0.51 0.49	0.42 0.44 0.47 0.45	
2.43	0.66 0.63 0.66 0.64	0.66 0.71 0.74 0.73	0.63 0.66 0.74 0.70	0.54 0.64 0.70 0.66	0.48 0.64 0.66 0.65	0.33 0.28 0.39 0.31
4.48	0.77 0.92 0.93 0.93	0.77 0.91 0.92 0.96	0.75 0.89 0.93 0.91	0.62 0.82 0.89 0.85	0.55 0.64 0.80 0.69	0.35 0.24 0.38 0.28
8.27			0.80 0.94 0.98 0.96	0.67 0.86 0.95 0.91	0.60 0.76 0.90 0.83	0.35 0.34 0.46 0.35

Note in each square, the values are arranged as shown below:

Value for no swirl, from Reference 1: C_{PR} (upper left); Value for indicated swirl (1, 2 or 3), calculated from Equation 3: C_{PRS_1} (upper right), C_{PRS_2} (lower right), C_{PRS_3} (lower left).

C_{PR}	C_{PRS_1}
C_{PRS_3}	C_{PRS_2}

moved upward considerably and its spread has been reduced to 0.10. This implies a large reduction in the dependence of performance on any variable other than area ratio, for this inlet swirl.

To show the effect of swirl on performance, the data were cross-plotted to obtain the performance map for swirl 1 shown in Fig. 7, plotted on coordinates of AR-1 vs L/R_1. For comparison, the corresponding performance map from McDonald and Fox[1] with no swirl is given in Fig. 6.

The upper branch of each curve on the performance map shifts to the left with increasing swirl. The shift was largest between zero and the lowest value of swirl. The shift was still present, but to a much smaller extent, as the swirl was increased further. The effect for the line $C_{PRS} = 0.70$ is shown in Fig. 8. The right branch of the curve remains fixed or shifts slightly in the direction of smaller area ratio.

The shift of the map moves line $\alpha - \alpha$, the line of optimum performance for diffusers of fixed length ratio, to the left. Thus, a shorter diffuser may be used to produce a given C_{PR} with appropriate swirling inlet conditions. The new lines $\alpha - \alpha$ are compared to the axial inlet flow case in Fig. 9.

The optimum swirl (i.e. the swirl angle at which C_{PRS} is maximized) for a given diffuser can be obtained for some cases from the data in Table 1 by plotting the performance against the degree of swirl. Data for the $2\phi = 8°$, $AR = 4.48$ diffuser are plotted in Fig. 10 to illustrate the method. The optimum swirl angle depends on the particular diffuser, but plots like Fig. 10 indicate the range of swirl angle over which the performance is maximized.

The performance coefficient shown in Fig. 10 is still increasing with swirl angle at the highest swirl tested, indicating that optimum swirl has not been reached for this diffuser. Equipment limitations prevented going to the higher swirl

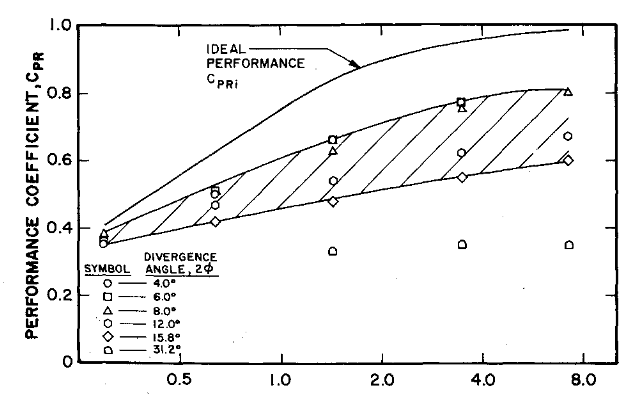

Fig. 4 Diffuser performance coefficient as a function of area ratio for axial inlet flow.[1]

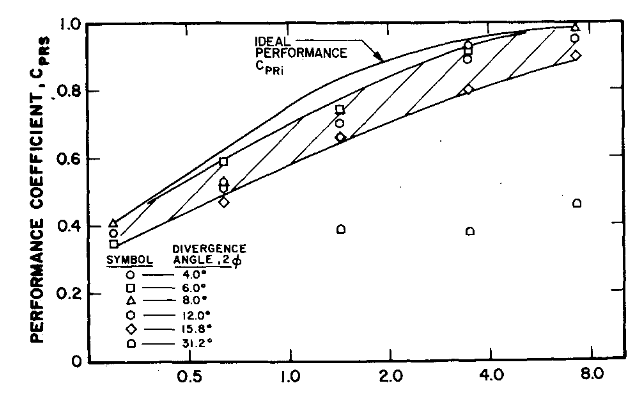

Fig. 5 Diffuser performance coefficient as a function of area ratio for swirling inlet flow (swirl 3).

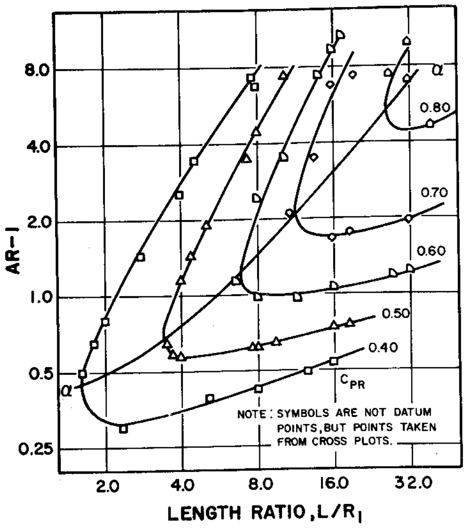

Fig. 6 Contours of constant performance coefficient for conical diffusers without swirl.[1]

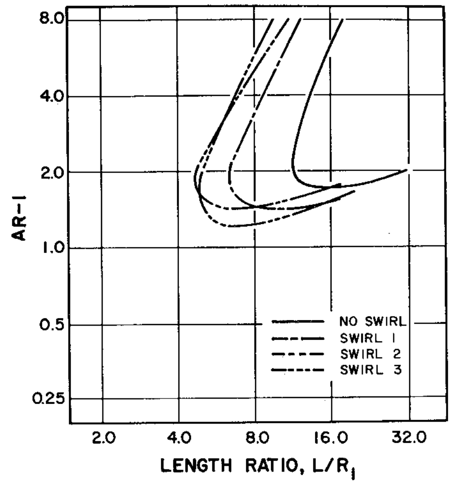

Fig. 8 Effect of swirling inlet flow on contour of constant performance coefficient, $C_{PR} = C_{PRS} = 0.70$.

generator rotational speeds necessary to produce a larger swirl angle. However, even the moderate swirls generated were sufficient to improve performance in all but the $2\phi = 31.2°$ diffusers.

At swirl angles higher than those necessary to produce optimum performance, performance will decrease. Thus performance as a function of swirl shows a definite maximum rather than an asymptotic behavior, although the range of swirl angles over which performance is close to optimum appears to be quite wide. The reason for the performance decrease at high swirl and a criterion for the onset of the decrease have been determined from previous investigations.

Harvey[10] studied the stability of a swirling flow in a straight pipe. He found that increasing the maximum swirl angle produced a stagnation bubble at the centerline of the pipe, as a result of the low pressure region created at the pipe center by the swirl. With sufficient swirl the difference between the wall and centerline static pressure becomes equal to or greater than the dynamic pressure of the axial flow. The total pressure of the centerline flow then is lower than in the surrounding flow, leading to stagnation or back flow. Harvey gave the criterion for the onset of the stagnation bubble as

$$\frac{\text{angular momentum}}{\text{axial momentum}} > 1.0$$

For solid body rotation and uniform axial flow this is exactly equivalent to saying that the maximum swirl angle is greater than 45°. However, this 45° swirl angle criterion holds only approximately for real swirl distributions. Work done in pipes by Youssef[11] gave results similar to those of Harvey.

Swirling pipe flow results also can give an indication of the limiting conditions for swirling diffuser flow. Since it was found that swirl angles increase slightly through the diffusers, the stagnation bubble should first appear at the diffuser exit and then move up to the inlet with increasing swirl. Precisely this behavior was reported by So[12] for a swirling flow in a conical diffuser with $2\phi = 6°$. Flow visualization showed the stagnation bubble and its movement upstream with increasing swirl. Unfortunately, So did not give the swirl conditions at the point when the stagnation bubble first appeared. His data showed no back flow for a maximum swirl angle of 20°, but indicated that backflow was present for a maximum swirl angle of 50°. Therefore, his work neither confirmed nor contradicted the work of Harvey.

From these considerations, one can hope that for maximum swirl angles less than 40 to 45° there will be no problems with diffuser blockage caused by a stagnation bubble or back flow along the centerline.

V. System Design

The striking increase in diffuser performance with swirl raised the question: Could swirl be used as a boundary-layer control device to improve the performance of a diffuser? The

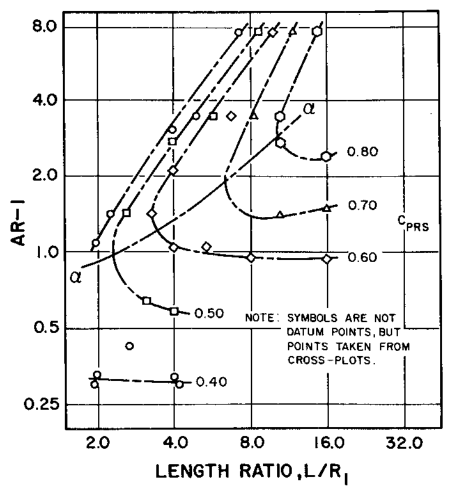

Fig. 7 Contours of constant performance coefficient for conical diffusers with swirling inlet flow.[1]

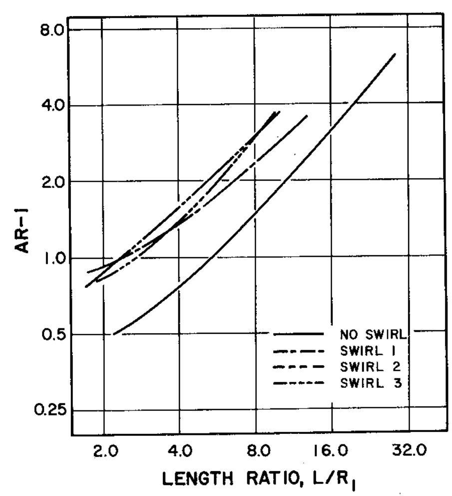

Fig. 9 Effect of swirling inlet flow on line of optimum performance at constant length ratio.

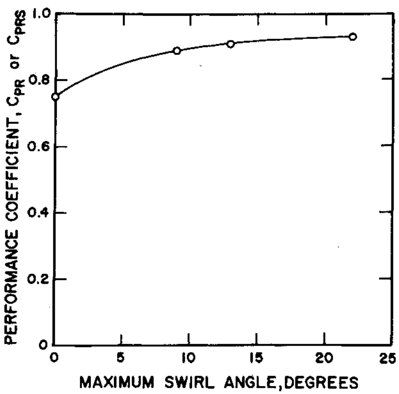

Fig. 10 Effect of inlet swirl on performance coefficient for conical diffuser with $2\phi = 8°$ and $AR = 4.48$.

answer depends on the losses caused by generating the swirl. The diffuser exit flow distribution also might be improved by the resulting reduction in separation. However, as with other diffuser performance increasing devices, complications of the system might more than outweigh the possible increase in efficiency.

VI. Conclusions

Results from a systematic series of experiments show that under some conditions swirling inlet flow can influence the flow regime and performance of a conical diffuser. Specifically: 1) The effect of inlet swirl for a given diffuser can be correlated with the flow regime present for axial inlet flow. If unstalled for axial flow, introduction of swirl has little effect; for badly stalled diffusers, addition of swirl improves performance. 2) For some of the diffusers tested, an optimum value of swirl was found which gave best performance. For units with combinations of large angle and area ratio, equipment limitations prevented reaching optimum swirl. 3) The geometry of an optimum diffuser will be different for swirling inflow than for axial inflow. 4) The results suggest that in some cases designers might consider adding swirl to improve the over-all performance of a complete flow system.

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