Effects of swirl on flow separation and performance of wide angle diffusers

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Experimental investigations have been carried out to determine whether the introduction of a circumferential velocity component can produce worthwhile improvements in the performance of, and eliminate flow separation in, wide angle conical diffusers. The swirl generator is a 24 flat-bladed, radial intake type. Systematic experimentation has been carried out for one diffuser configuration fitted with a tailpipe (16.5° and 4.4 area ratio) using varying strengths of inlet swirl and introducing the dissipated mechanical energy as the main criterion of diffuser performance. The best inlet swirl strength produced about 60% reduction of the total diffuser losses in swirl-free flow. The analysis of these results, together with information obtained from flow visualisation experiments, suggests that increasing the swirl beyond an observed threshold completely eliminated flow separation, but it also gave rise to a central zone of recirculating flow and hence additional dissipative losses. We conclude that the optimum improvement achievable in wide angle diffuser performance using swirl does not require the addition of more energy than it saves

Key words: *diffusers, fluid flow, recirculating flow, swirl*

In many industrial fluid flow situations it is often necessary to reduce the kinetic energy to increase the static enthalpy using a diffuser (eg at the outlet of pumps, fans, turbomachines and other engineering appliances). The engineer might be unable to incorporate a diffuser of optimum characteristics in such situations because of local constraints. Wider angle diffusers of shorter lengths would then be needed, and the diffuser effectiveness would be adversely affected as a result of flow separating from the walls, as any of the conventional performance criteria would indicate.

Various experimentalists have attempted to minimise or eliminate flow separation through the application of suction[1], the injection of a secondary stream[2], the introduction of vortex generators[3], or by the use of vanes[4,5] which divide the diffuser into several more efficient smaller angle diffusers. It has been claimed that the performance of a diffuser containing a separation region may be improved by the application of a circumferential component of velocity. This swirling flow will give rise to radial pressure gradients which may have beneficial effects by altering the shear stress distribution and counteracting the tendency of the flow to separate from the diffuser walls, but this must be balanced against the additional friction losses.

The review of subsonic diffuser flows by Cockrell and Kline[6] had indicated that the effects of swirl on separated diffuser flow is not well documented, but subsequently experimental work on the effects of swirl on diffuser performance has been carried out by Al-Obaidi[8]. McDonald and Fox[10] and Senoo *et al*[12]. These investigations showed that swirl had beneficial effects on the performance of conical diffusers, based on the criterion of the maximum pressure recovery coefficient.

This paper presents experimental measurements which have been obtained in a conical diffuser fitted with a tailpipe. Swirl of varying strength was introduced using a swirl generator and its effects on flow separation and diffuser dissipative losses were studied. It was observed that there existed a swirl-strength-threshold, above which the diffuser performance deteriorates.

Flow apparatus

The general layout is shown in Fig 1; the diffuser consists of an inlet duct made of perspex, a conical

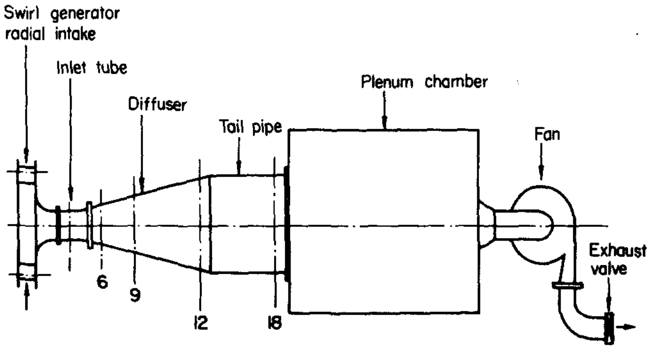

Fig 1 The flow apparatus

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Received on 24 February 1983 and accepted for publication on 15 July 1983

section with a total divergence of 16.5° and area ratio of 4.4, also made of perspex, and a tailpipe made of non-transparent material. The transparent nature of the main sections of the diffuser facilitated flow visualisation observations. A radial-intake swirl generator having 24 swirl blades was also fitted. A multistage constant-speed fan drew air through the apparatus via a plenum chamber and discharged it into the laboratory atmosphere.

Instrumentation

Measurements of time-mean velocities and yaw angle were made using a wedge-probe and Betz micromanometers (Fig 2). A control valve at the exit of the exhaust system controls the volume flow and a wall pressure tapping fitted in the exhaust duct enables volume flow rate measurements, $\dot{V}$, to be made. The diffuser was fitted with static tappings along its length and access for traversing the flow was provided by slots along the top side of the diffuser assembly. The slots could be covered by carefully fitted perspex inserts when not in use.

Traverses of total and static pressures and yaw angles were made at each of the eighteen stations. The first measurement station was in the cylindrical portion of the inlet tube, with subsequent stations placed at intervals given in Table 1.

Experimental results

The complication of data analysis for swirling flow cannot be over-emphasised. For swirl-free flow in the steady state, static pressure can be assumed constant across any cross-section, so that traverses have to be made using a suitable probe in order to evaluate the velocity and pressure of flow.

All the measurements here were made using a commercial wedge type probe. The probe was carefully calibrated for immersion depth error and the effects of one-dimensional shear on yaw angle. Guidance was sought in the literature on the effects of pitch angle, Reynolds number, probe vibration and turbulence on the measurements. It was observed that the main errors in these experiments were those due to immersion depth. Preliminary investigations in swirl-free flow at the diffuser throat where total pressure was substantially uniform indicated that assuming the wall static pressure was the true static pressure at

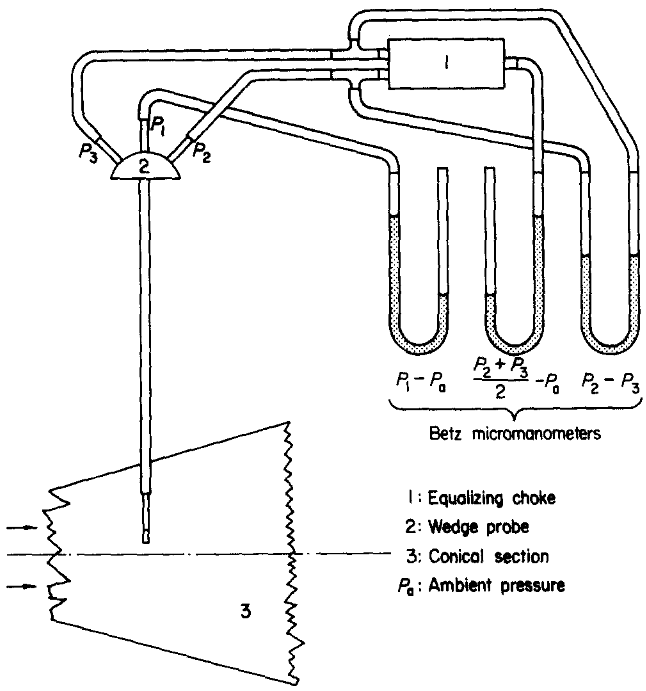


Fig 2 Wedge probe traverse arrangement

Table 1 Measurement station positions

Station no.	z, mm	R, mm
1	0.0	61.0
2	47.1	63.0
3	94.3	64.6
4	141.1	66.7
5	188.6	69.6
6	282.9	80.9
7	330.0	87.1
8	377.1	93.5
9	424.3	100.3
10	471.4	107.4
11	518.6	114.4
12	565.7	120.8
13	612.9	127.6
14	697.9	127.6
15	783.1	127.6
16	868.2	127.6
17	953.3	127.6
18	1103.9	127.6

Nomenclature

R	Radius
P	Pressure
H	Pressure head
U_z, U_r, U_θ	Axial, radial and tangential velocities
z, r, θ	Axial, radial and tangential coordinates
L	Losses
ρ	Density
μ	Viscosity
C_{PR}	Coefficient of pressure recovery
$\dot{V}$	Volume flow rate
S_N	Swirl number

Suffices

g	Gauge
so	Static condition, probe absent
0	Station zero
1	Station one
2	Station two
I	Ideal condition
e	Flow exit
at	atmospheric condition

the cross-section the indicated static pressure varied. This variation disappeared with downstream distance and at cross-sections more than $30D$ (where D is the maximum probe diameter) such errors were almost non-existent. Suitable corrections to the measured pressure and velocity consistent with the experimental observations were therefore applied. The estimated resolving ability of the wedge probe for mean velocity directions in plane flow was found to be ±1.2° and for shear flow to be ±1.5°.

The volume flow rate, $\dot{V}$, in swirl-free flow was determined by integrating the velocity distribution at the diffuser throat. The corresponding value of the pressure P_e in the exit pipe upstream of the exhaust valve was measured. This valve acted as an orifice meter, so that:

$$\dot{V} = \text{const} \times (P_e - P_{at})^{1/2} \qquad (1)$$

with the constant independent of swirl rate. The pressure difference $(P_e - P_{at})$ was kept constant throughout the investigation.

The bulk of the experimental results are presented as plots of the tangential and axial velocity vectors normalised by the mean inlet axial velocity $(\bar{U}_z)$, in swirl-free flow, and static pressure readings across the measurement stations in the form of pressure head H in mm H_2O. The flow was traversed normal to the diffuser central axis.

The amount of swirl in a flow is often represented by a parameter, S_N, called the swirl number and defined as:

$$S_N = \frac{\text{Total flux of moment of momentum across a section}}{\text{Total flux of axial momentum} \times \text{radius}} \qquad (2)$$

For 0, 3, 5 and 7° swirl blade angle settings, the corresponding inlet swirl numbers were 0, 0.0342, 0.0551 and 0.854 respectively.

The experiments were carried out at a constant volume flow rate of 0.85 m^3/s corresponding to an average inlet axial velocity of 72.44 m/s, and exit wall pressure tapping of 42.0 mm H_2O and a nominal Reynolds number of 5.2×10^5.

The measurements shown in Fig 3 reveal that for swirl-free flow the onset of flow separation occurred on the upper wall of the conical section and was located between stations 7 and 15. Flow visualisation experimentation also revealed the occurrence of an intermittent zone of flow separation within this region.

To illustrate the effects of a swirling inlet component of velocity on the overall flow pattern, profiles of axial velocity at stations 1, 8, 14 and 18 for $S_N = 0.00$ and 0.0551 respectively, and at stations 1, 14 and 18 for $S_N = 0.00$, 0.0551 and 0.0854, are plotted in Fig 4. It was observed that for 5° blade angle there was a sharp drop in core velocity right from station 1 through to station 16, but this reduction recovers as we proceed from inlet to outlet. Another remarkable feature is the increase in the values of near-wall axial velocity components as a result of the presence of swirl†, which clearly indicates that as a result of the core reduction of the flow, a corresponding increase of flow resulted in the wall region. This phenomenon had been observed by Kiya, Fukusako and Arie[11] in their work on laminar swirling flow in a circular pipe. Beyond station 11 the apparent symmetry of the flow had disappeared.

The distribution of static pressure across the diffuser for zero swirl was found to be effectively uniform at stations beyond station 7 and, as is apparent

† The near-wall flow is energised, consequently delaying flow separation.

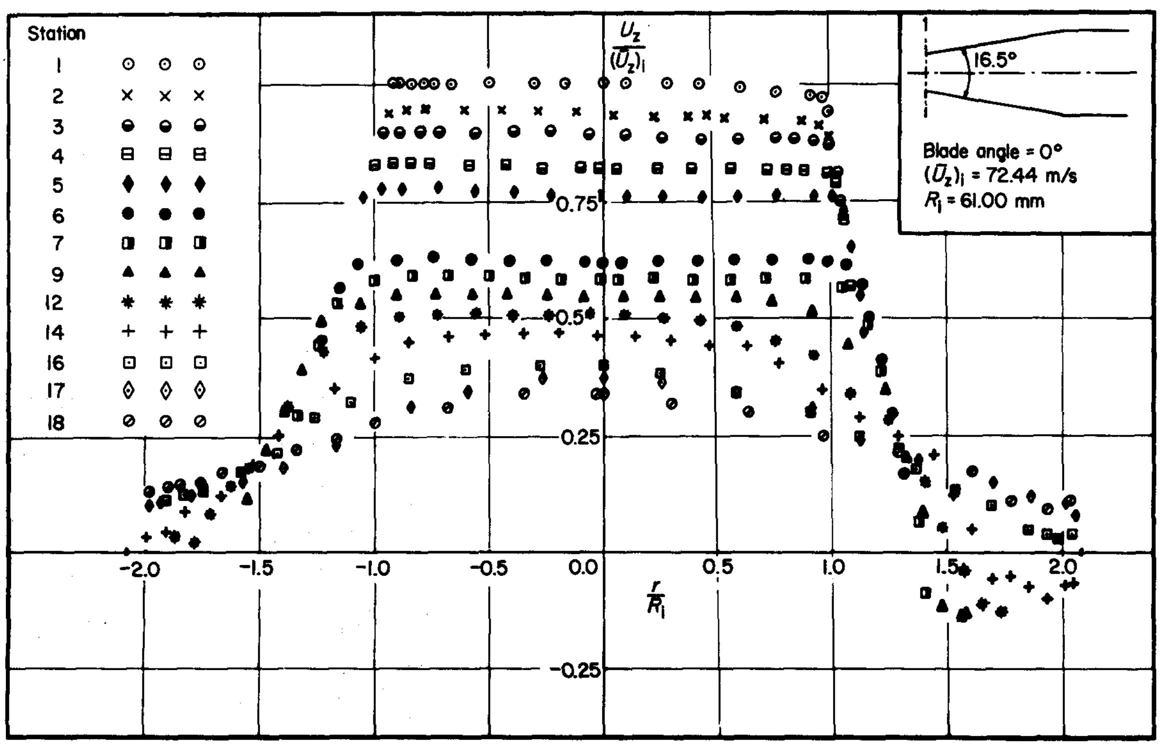


Fig 3 Experimental axial velocity distribution

from the experimental points in Fig 5, the static pressure appeared to depart from the wall value as the probe was withdrawn. This observation contradicts logical expectation, but is was felt that this could be due to streamline displacement error which seemed to get worse within regions of adverse velocity gradient. It may also be a function of duct diameter. For $S_N = 0.0551$, the static pressure distributions across stations 1, 3, 5, 6, 10 and 18 are shown in Fig 6. Across each section, except at stations beyond 10, the tendency is that the static pressure decreases towards the diffuser axis conforming to the general trend of the corresponding axial velocity profiles. This seems a phenomenon characteristic of swirling flows, and has been similarly observed by So[7], Al-Obaidi[8] and Senoo *et al*[12] in their experimental investigations.

The distributions of static pressure along the wall and along the central axis of the diffuser-tailpipe assembly are shown in Fig 7 for 0°, 5°, 7° blade angles. It can be seen that in swirl-free flow, the static pressure on the diffuser axis, after an initial high rate of increase, tends to a constant value as one proceeds downstream.

For 5° blade angle swirl, the wall static pressure distribution remained unaffected by the swirl, but the pressure on the diffuser axis suffered a sharp drop at the initial stations. This picks up gradually as one proceeds downstream. Flow visualisation experiments showed that this flow ($S_N = 0.0551$) began to separate in a region located within stations 9 and 14, a region further downstream than for swirl-free flow.

For 7° swirl blade angle ($S_N = 0.854$), flow visualisation revealed that the flow did not separate from the diffuser wall. It was observed, however, that a central recirculation zone resulted. This zone extended from the front wall A, (Fig 1), of the swirl generator to a cross-section near station 14. The zone was unstable and possessed spinning characteristics as depicted in Fig 8. It covered this region for about 70% of the time. Due to the unstable nature of the central zone, flow measurements were only possible at stations 1, 14, 15, 16, 17 and 18. Fig 9 shows the distribution of axial velocity and Figs 10 and 11 show the distribution of the corresponding tangential velocity. The trend of the inlet axial velocity within the core region confirms the presence of a central recirculation zone. The profiles in the downstream stations have a tendency to become uniform and fully developed. This event, the purpose for which the tailpipe was carefully designed, is most helpful as it facilitated the deduction of downstream boundary conditions in a calculation procedure soon to be published.

This result of a strong swirling flow in diffusers had also been observed by So[7] and Al-Obaidi[8] who noted that the central recirculation zone was obscure and difficult to map. Fig 12 shows the distribution of static pressure for 7° swirl blade angle flow. The trend at the first station shows that, as a result of the swirl, a remarkable radial pressure gradient resulted within the core of the flow. Downstream, the static pressure remained constant at all stations after station 14. The

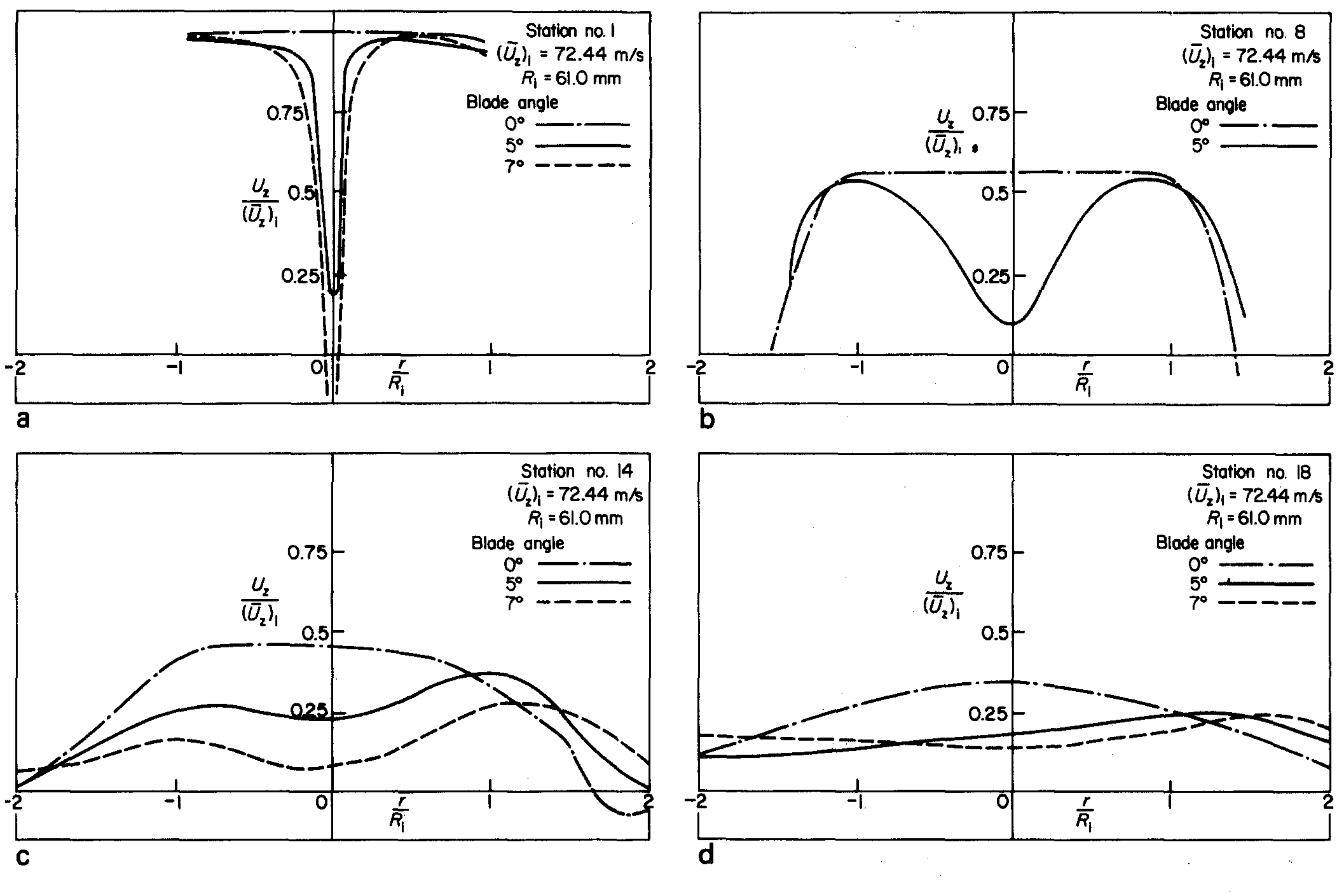


Fig 4 Experimental axial velocity

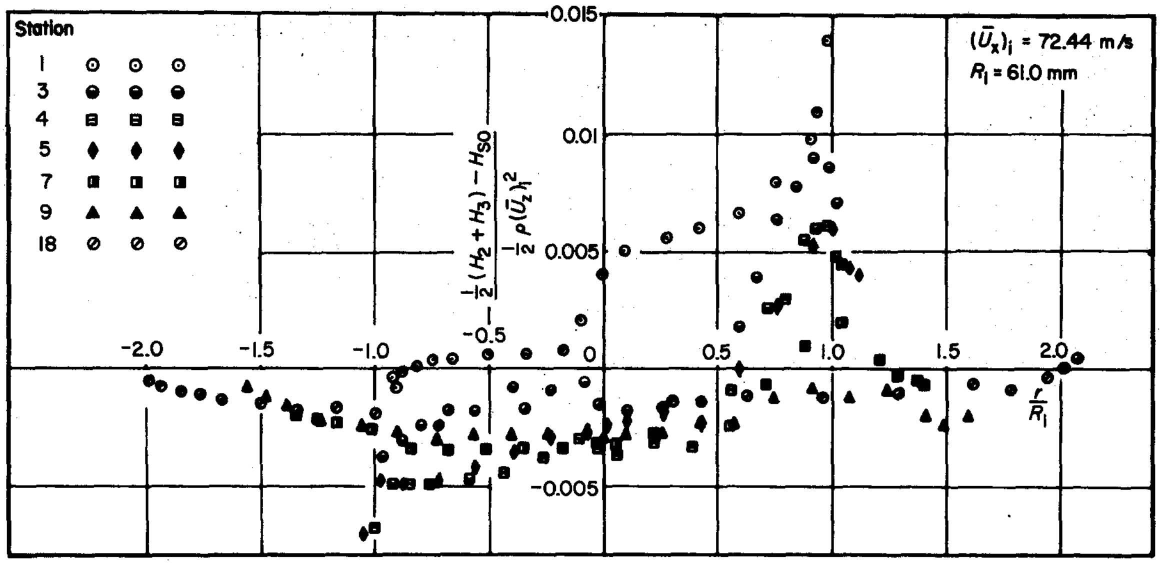


Fig 5 Immersion depth calibration

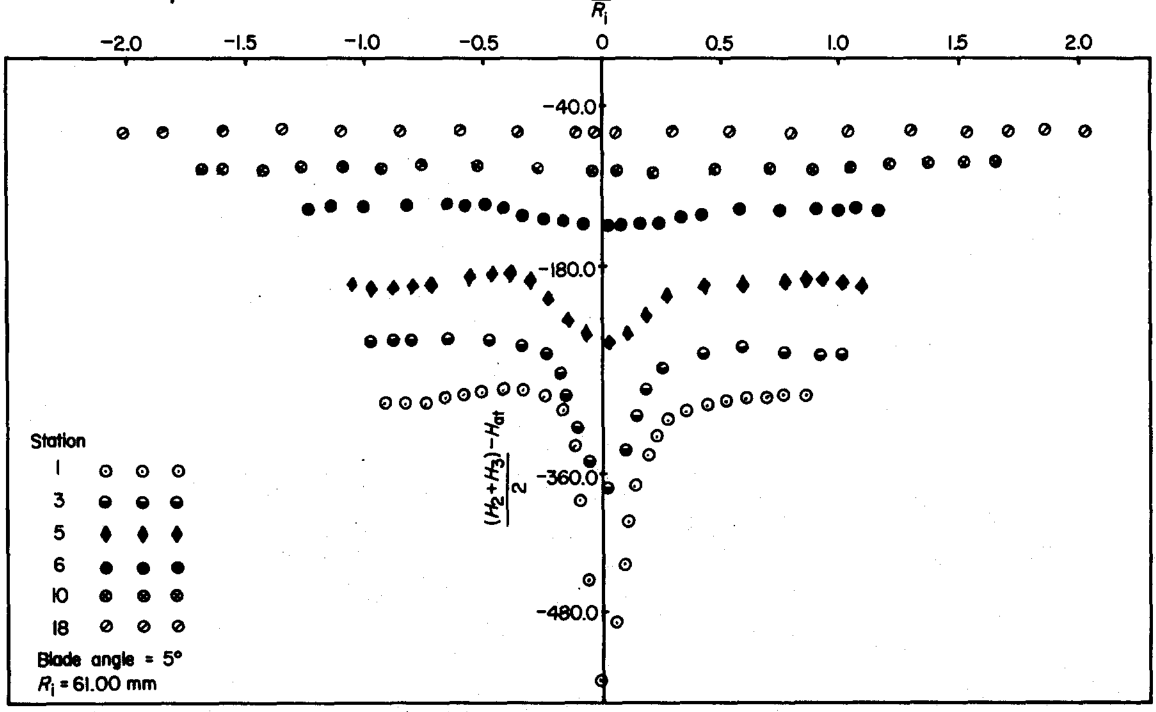


Fig 6 Static pressure measurements (mm H_2O)

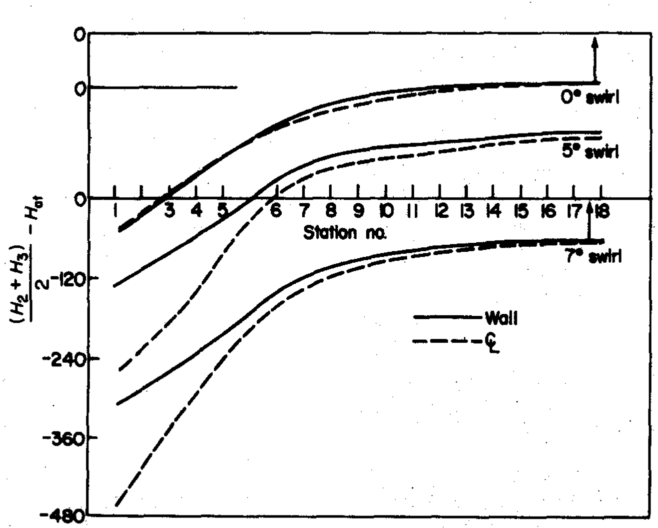


Fig 7 Static pressure distribution in a 16.5° diffuser

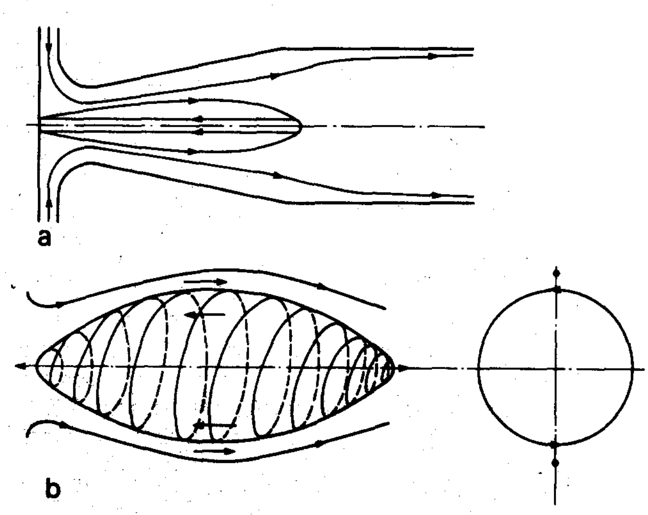


Fig 8 Flow pattern in a 16.5° diffuser (α = 7°)

distribution of the wall and centre-line static pressure is as shown in Fig 5.

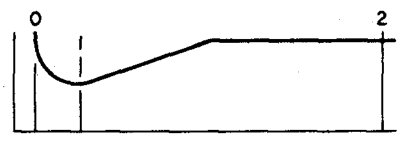
As a result of the large radial pressure gradient generated at upstream stations by swirl, a number of different ways of defining a pressure recovery coefficient are possible, depending on whether the wall pressure or an average pressure or the centre-line pressure is chosen to be representative of the inlet static pressure. Senoo *et al*[12] considered three possible definitions of pressure recovery coefficients, all of which showed that swirl had a beneficial effect. Due to the uncertainty in the choice of the most representative upstream pressure value, however, an alternative approach based on energy considerations is used here.

Diffuser performance expressed in terms of the dissipated mechanical energy is as shown in Table 2. This is deduced from the steady flow energy equation proposed and fully discussed elsewhere[9]. Table 2 shows that the introduction of 5° blade angle swirl resulted in a 46% reduction in the dissipated mechanical energy compared to swirl-free flow (an improvement), on the basis of the energy available at inlet to the diffuser above atmospheric conditions.

The application of 7° blade angle inlet swirl also resulted in a 34% reduction in the dissipated mechanical energy in swirl-free flow. However, Table 2 also shows that although the swirl is stronger than for the 5° blade angle, the resulting central recirculation zone gave rise to a net increase in dissipated mechanical energy above that for the 5° blade angle. The additional increases in wall friction, resulting from the unseparated swirling flow (7° swirl blade angle) is also thought to contribute to this dissipated energy increase between 5° and 7° blade angle swirling flows. The losses in the swirl generating device are also given in Table 2. They can be seen to be much less than the overall losses in the diffuser assembly.

Table 2 Diffuse performance in terms of dissipated mechanical energy

Blade angle, degrees	L_{0-1} % of $\int_1 \frac{u^2}{2}\,d\dot{m}$, (Exptl)	L_{1-2}, % of $\int_1 \frac{u^2}{2}\,d\dot{m}$, (Exptl)	C_{PR}, %, (Exptl)
0	0.8	23.4	75.0
3	1.3	17.34	79.0
5	2.4	15.8	80.82
7	2.7	19.35	77.66



where by applying the steady flow energy equation at sections 0, 1 and 2,

L_{0-1} = swirl generator loss,

$$\int_0 \left(\frac{P_g}{\rho}+\frac{U^2}{2}\right) d\dot{m} - \int_1 \left(\frac{P_g}{\rho}+\frac{U^2}{2}\right) d\dot{m}$$

L_{1-2} = dissipation in diffuser,

$$\int_1 \left(\frac{P_g}{\rho}+\frac{U^2}{2}\right) d\dot{m} - \int_2 \left(\frac{P_g}{\rho}+\frac{U^2}{2}\right) d\dot{m}$$

C_{PR} = pressure recovery coefficient,

$$\frac{\int_1 \frac{P_g}{\rho}\,d\dot{m} - \int_2 \frac{P_g}{\rho}\,d\dot{m}}{\int_1 \frac{U^2}{2}\,d\dot{m}}$$

C_{PR_i} = inviscid pressure recovery coefficient,

$$1-\left(\frac{A_1}{A_2}\right)^2 = 95\%$$

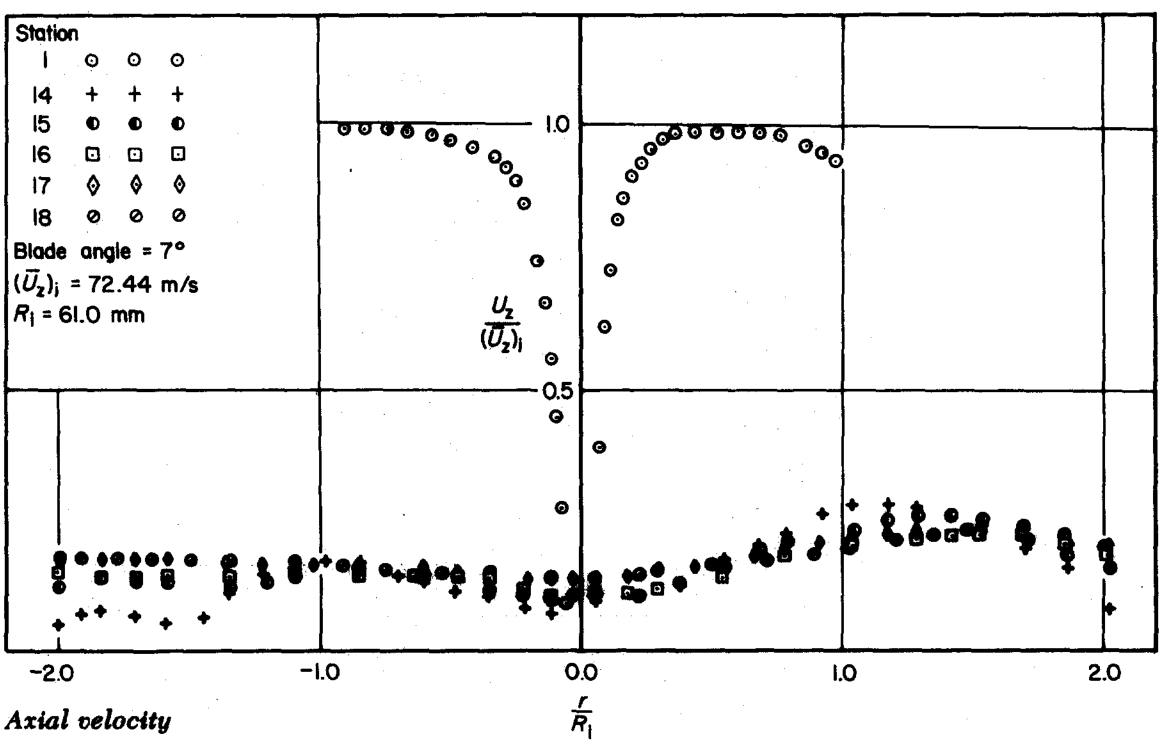


Fig 9 Axial velocity

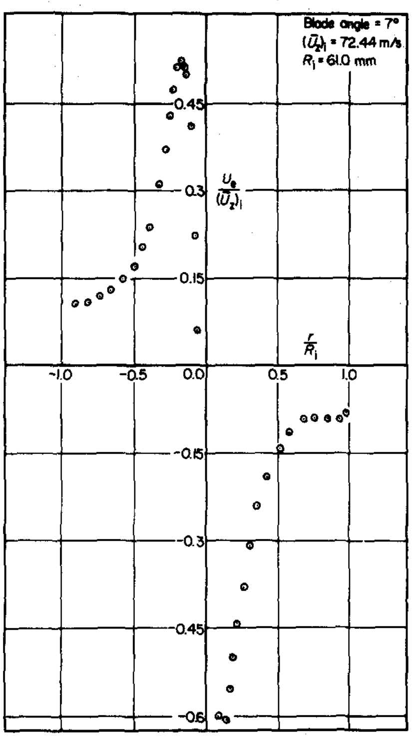


Fig 10 Swirl velocity (inlet)

Conclusions

The main conclusions which may be drawn from the experimental investigations reported here are:

- The separation of the flow in the 16.5° diffuser fitted with a tailpipe did not occur symmetrically along a single flow cross-section. Rather, it had a three-dimensional asymmetric character. The flow distribution well upstream of the region of separation was symmetrical but the separation resulted in a region of asymmetrical flow distribution well downstream of its onset. The tailpipe returned the swirl-free flow to its symmetrical nature.
- When swirl of moderate strength ($S_N = 0.055$) was applied to the inlet of the 16.5° angle diffuser, the region of the onset of flow separation in swirl-free flow was transferred some distance downstream. Remarkable variations in radial pressure also resulted with a consequent beneficial redistribution of the flow. Increasing the strength of inlet swirl ($S_N = 0.085$) eliminated the flow separation but resulted in an undesirable central zone of intermittent and recirculating flow. This central zone was complex and it possessed a spinning characteristic.
- A swirl generator-inlet pipe-diffuser combination is a simple arrangement for producing adequate velocity profiles for entry into downstream duct elements, such as pipe systems, heat exchangers or laboratory atmosphere, in a relatively short length ($4\,R_0$) of tailpipe.
- Swirl-generating flow control techniques can be successfully applied in wide-angle conical diffusers which have unstable and separated flow under normal operating conditions; the swirl flow control-technique seems to be one of the most promising for industrial applications since, under such situations, swirl is often present anyway, as for example in the draught tube fitted to a hydraulic turbine running at an off-design condition or the outlet of an axial flow fan.
- As a result of the generated swirl, it has been observed that there is a sharp increase in both the radial pressure gradient everywhere and the axial pressure gradient in the immediate upstream stations especially in the core of the flow. This observation agrees also with the works of Al-Obaidi[8] and Senoo *et al*[12]. Thus it could be misleading to define the absolute performance of a diffuser entirely on the basis of static pressure recovery.
- The energy method proposed elsewhere[9] for assessing diffuser performance has been observed to yield meaningful and logical results. It shows that the losses in swirl-free flow are reduced under conditions of optimum swirl and that increasing the

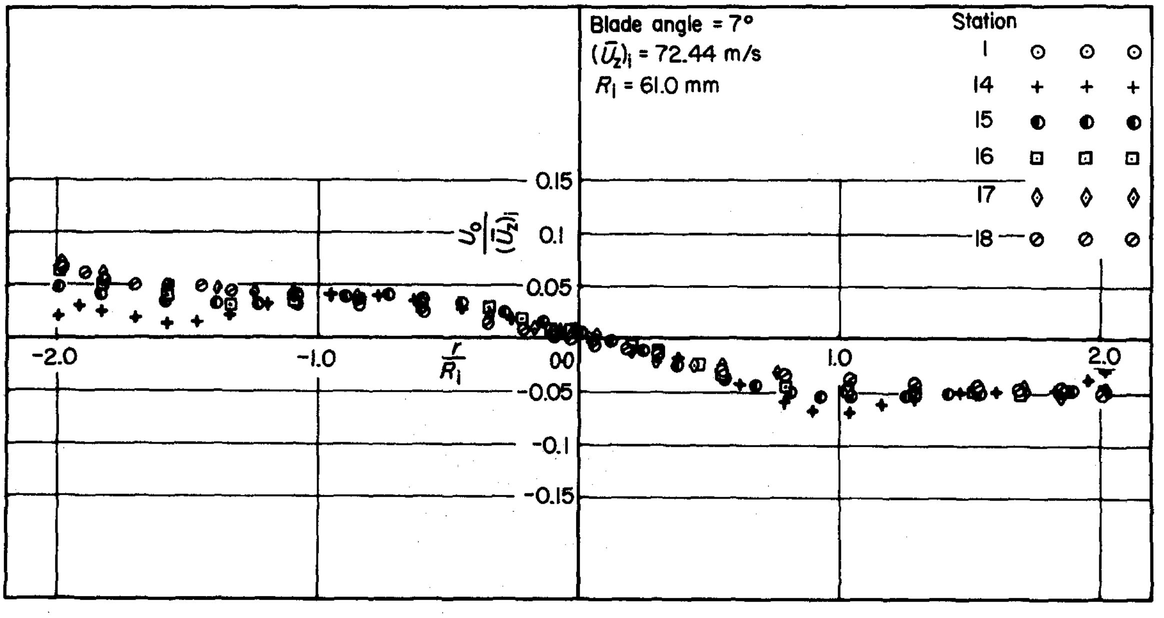


Fig 11 Swirl velocity

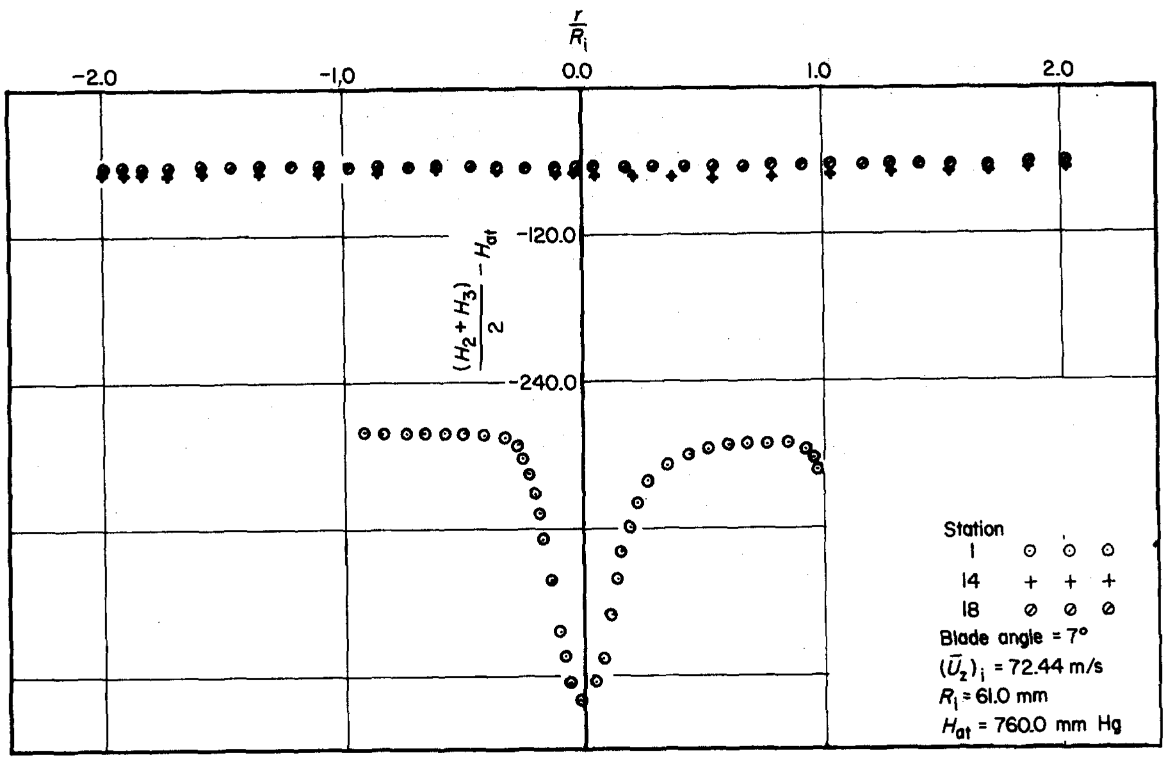


Fig 12 Static pressure (mm H_2O)

strength of swirl beyond the optimum results in additional losses. The optimum intensity of inlet swirl corresponds to a swirl blade angle setting of about 5° in the experimental set-up used.

- Finally, although the improvement in diffuser flow and performance using swirl must depend on whether the swirl present was generated or pre-existed, this investigation has revealed that, for the type of swirl generator employed, the losses incurred are much smaller than the overall losses. Hence, one could conclude that the optimum improvement achievable in wide angle diffuser performance does not require the addition of more energy than it saves.

Acknowledgements

Many thanks to the late Mr. G. Langer who supervised this work but died some months before it was published. Also, thanks to Professor W. A. Woods for reading the manuscripts and making some useful suggestions.

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