Influence of Compressor Exit Conditions on Diffuser Performance

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In this paper results are presented of an experimental investigation into the way in which compressor exit conditions influence the performance of two optimum combustor-dump diffusers. In an optimum system nearly all of the pressure rise occurs in a relatively long prediffuser which is designed to produce a symmetrical outlet velocity profile at the design flow split. One diffuser was tested downstream of a seven-stage axial flow compressor and, for comparative purposes, retested with fully developed flow at inlet. The second diffuser was tested downstream of an annular tandem cascade which could be sited at a number of positions relative to diffuser inlet. The overall performance, measured when the wakes from the outlet guide vanes had decayed considerably, was the same as that achieved with fully developed inflow. No significant penalty in overall performance was incurred even when the outlet guide vanes were sited close to the prediffuser inlet. Furthermore, reduction in the distortion and improvement in the stability of the prediffuser outlet velocity profile were obtained when the outlet guide vanes were near to the diffuser inlet.

Nomenclature

A = area of cross section
B = blocked area fraction $= 1 - 1/A \int_0^A (u/U_m)\,\mathrm{d}A$
c = blade chord length
C_p = pressure recovery coefficient
D = distance between head of flame tube and outlet of prediffuser (dump gap)
h = annulus height
L = length of prediffuser
L_e = entry length
m = mass flow rate $= \rho A U$
M = mean Mach number
n = number of outlet guide vanes
p = static pressure
P_t = total pressure
ΔP_t = loss of total pressure
q = dynamic pressure
R = radius
S = flow split $= (m_o/m_i)_4$
S^* = design flow split
t = blade pitch
u = local axial velocity
U = mass-derived mean velocity $= m/\rho A$
U_m = maximum velocity in cross section
W = width of flame tube
x = axial distance measured from trailing edge of outlet guide vanes
y = distance perpendicular to outer wall
z = distance in circumferential direction
α = velocity profile energy coefficient $= 1/A \int_0^A (u/U)^3\,\mathrm{d}A$
λ = loss coefficient
ρ = fluid density
ϕ = diffuser wall angle

Subscripts

0 = exit plane of outlet guide vane
1 = prediffuser inlet
2 = prediffuser outlet
3 = head of flame tube
4 = settling length
i = inner annulus flowfield
o = outer annulus flowfield
m = maximum value

Superscripts

$(\bar{\ })$ = mass-weighted mean value
$(\ )'$ = ideal value

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Introduction

IN the combustion system of a gas turbine engine, it is necessary to diffuse the air delivered by the compressor in order to achieve efficient combustion and avoid a large loss of total pressure. A typical annular turbojet combustor is shown in Fig. 1a. High-velocity air leaving the compressor is split into three streams: two main streams that pass to the inner and outer annuli surrounding the flame tube, and a smaller stream which feeds the primary zone. Apart from a small percentage required for turbine cooling, nearly all of the flow in the two main streams enters the flame tube via the primary and dilution ports. To achieve the same penetration in each row of dilution jets, a circumferentially uniform distribution of pressure and velocity should exist around the flame tube. Therefore, apart from minimizing the loss of total pressure, the main requirement of the diffuser system is that it should insure that a uniform flow is delivered to the annuli surrounding the flame tube.

Current combustor diffuser designs are illustrated in Fig. 1. The "faired" diffuser is widely used, but the advent of the high bypass ratio engine, in which a low gas generator flow is

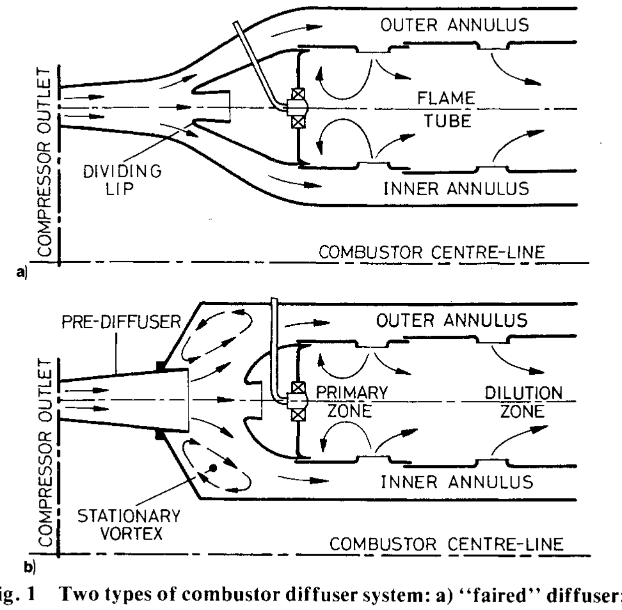

Fig. 1 Two types of combustor diffuser system: a) "faired" diffuser; b) dump diffuser.

compressed to a high pressure ratio, has led to very small annulus heights in the diffusers feeding the annuli surrounding the flame tube. Consequently, manufacturing tolerances and differential expansion and distortion during operation can produce significant variations in geometry, leading to circumferential nonuniformities in the flow at diffuser exit. Furthermore, changes in engine operating conditions produce different velocity profiles at the compressor exit, in which case the dividing lips may become misaligned with the flow and flow asymmetry, separation, and unsteadiness may occur. To avoid these problems the dump diffuser, shown in Fig. 1b, has been developed. The abrupt expansion at prediffuser exit insures good stability of flow over a wide range of operating conditions. Fishenden and Stevens[1] investigated the performance of a wide range of combustor dump diffusers. The pressure recovery and total pressure loss were measured; and the influence of prediffuser geometry, the division of flow between the two annuli surrounding the flame tube [henceforward referred to as "flow split," $S=(m_o/m_i)_4$], the distance between the head of the flame tube and the prediffuser (dump gap D) were investigated. The results indicated that the system offered good stability of flow and an overall performance comparable with that of a "faired" diffuser.

Although a considerable amount of research has been undertaken to define the optimum geometry of combustor-dump diffusers, most of the work has been carried out with fully developed flow at the prediffuser inlet. Whereas the overall blockage produced by such a flow is typical of engine operating conditions, it does not simulate the circumferential nonuniformities and unsteadiness introduced by the compressor. The flow downstream of the outlet guide vanes is that of a shear flow dominated by blade wakes and their interaction with the annulus wall boundary layers. It is also important to note that a high level of turbulent mixing has been observed by Lockhart and Walker,[2] not only in the axial direction but also in the circumferential direction, a feature which is clearly absent in fully developed inflow.

An initial attempt to simulate compressor exit conditions was made by Klein et al.,[3] who introduced an annular tandem cascade, giving zero exit swirl, at a position 0.25 of a blade chord upstream of diffuser inlet. For comparison, tests were also carried out with a boundary-layer-type inlet nonuniformity produced by an approach length of 22 inlet annulus heights. The results indicated that with the cascade at the inlet, the losses in total pressure were typically 50% higher than those measured with inlet conditions generated by a long approach length. Furthermore, the optimum geometry of the system was modified, and the wakes from the blades became more pronounced as they passed through the prediffuser. It is quite likely that the wakes may have been retarded to such an extent that local backflow was created. This kind of "internal stall" has been described by Wolf and Johnston.[4]

More recently, Adenubi[5] has investigated the flow regimes and pressure recovery of a straight core annular diffuser operating downstream of a single-stage axial flow compressor. The values of static pressure recovery coefficient compared favorably with the values predicted using the correlation due to Sovran and Klomp.[6] This suggests that design procedures normally used for conditions in which the diffuser inlet is naturally developed would be applicable to diffusers operating downstream of an axial flow compressor.

Clearly there is some conflict between the work of Klein et al. and Adenubi. However, it is difficult to compare the two series of tests because they were not conducted with the outlet guide vanes positioned at the same relative distance from the diffuser inlet. In the tests of Klein et al. the trailing edge of the outlet guide vanes was 0.25 of a blade chord upstream of the diffuser. Whereas, in Adenubi's tests the distance was about two blade chord lengths, in which case the blade wakes would have decayed considerably by the time they reached the diffuser inlet. This is the most likely explanation for the good agreement with the correlation due to Sovran and Klomp.

Irrespective of whether the wakes from the outlet guide vanes are allowed to decay in a diffuser or an entry length a total pressure loss will be incurred. The point that is really at issue is as follows: given a specified length for diffusion, is it better to allow the wakes to decay before they enter the diffuser or should the diffuser be sited immediately downstream of the outlet guide vanes? In the latter case the length of the diffuser is increased, the overall pressure gradient is reduced and possibly lower overall total pressure losses are incurred even though the wakes decay within the diffuser.

Tests have therefore been carried out on two combustor-dump diffusers with the object of assessing the influence of compressor exit conditions. One diffuser was tested downstream of a seven-stage axial flow compressor at the Aero Engine Division of Rolls-Royce (1971) Ltd., and for comparative purposes, retested with fully developed flow at inlet, henceforth this series is referred to as test series I. A second diffuser was tested downstream of an annular tandem cascade which could be sited at a number of positions relative to diffuser inlet; this is identified as test series II.

Test Facility

Test Series I

In the first series of tests a dump diffuser was mounted downstream of a multistage compressor driven by a Rolls-Royce Avon powered free turbine. The geometry of the diffuser is detailed in Table 1 and Fig. 2. The flame tube comprised a hemispherical head with parallel sidewalls. All of the flow passed to the annuli surrounding the flame tube, and no attempt was made to simulate the flow that normally enters via ports and cooling rings. The influence of flame tube porosity is the subject of a current investigation. In an attempt to avoid the regions of low pressure that can occur on the head of the flame tube at moderate dump gaps, a trumpet-shaped prediffuser was incorporated. The diffuser wall angle increased along a circular arc from 0 deg at inlet to 28 deg at outlet. In this way the reduction in the effective area of the flow path between the head of the flame tube and the vortex in the dump region, that normally occurs with a straight-walled prediffuser, was alleviated. Without the stabilizing influence of the flame tube it will be noted that, based on the flow regime studies of Howard et al.,[7] the flow in the prediffuser should be separated. Throttles were provided at the end of the settling lengths (station 4) to adjust the flow split to the design value. In all of the tests carried out downstream of the

Table 1 System geometry

Parameter	Series I tests	Series II tests
h_0, mm	28.60	38.10
h_1, mm	22.90	38.10
$L_e/(2h_1)$	34	24
$(R_i)_0$, mm	330.0	216.0
$(R_i/R_o)_0$	0.92	0.85
$(R_i)_1$, mm	333.5	216.0
$(R_i/R_o)_1$	0.936	0.85
L/h_1	2.37	3.81
2ϕ, deg	see text	12.0
W/h_1	4.05	3.5
$(R_i)_4$, mm	280.0	110.0
$(h_i)_4/h_1$	1.445	1.54
$(h_o)_4/h_1$	1.02	0.81
D/h_2	0.47	0.7
A_0/A_1	1.25	1.0
A_2/A_1	2.27	1.8
$(A_o/A_i)_4$	1.0	1.20
$(A_o+A_i)_4/A_1$	2.5	2.0
c, mm	29.21	30.48
n	162	97

compressor the distance between the trailing edge of the outlet guide vanes and diffuser inlet was maintained at 1.95 blade chord lengths. Furthermore, within this distance, the annulus area was reduced by 20% to increase the diffuser inlet Mach number.

Wall static pressure measurements were made at the exit of the outlet guide vanes, along the length of the prediffuser, and in the annuli surrounding the flame tube. For the compressor tests total pressure rakes were sited at stations 0, 1, and 4 (see Fig. 2), whereas with fully developed inflow total pressure traverses were conducted at stations 1 and 4. To obtain the radial distribution of axial velocity at stations 0 and 1, measurements of total pressure were selected at about 45 circumferential locations, extending over four blade pitches. At stations 0 and 4 the velocities were calculated on the assumption that the static pressure along each radial traverse was the same as that measured at the wall. However at station 1 the velocity profiles were obtained using the measured static pressure variation across the annulus. Measurements were also made with a Cobra probe at station 0 to establish whether the flow was free from swirl.

Test Series II

A fully annular test facility was used in which the entry length, prediffuser, and combustion chamber, were mounted vertically. The advantage of this arrangement was that, as all the inner tubes were spigotted together, they could be positioned simply by three struts in the entry flare. In this way the influence of entry length supports was reduced to a minimum. Air was drawn from the laboratory through an intake, having a contraction ratio of 8:1, into an entry length some 48 annulus heights long. Stable transition was insured by trip wires on the inner and outer walls just downstream of the intake throat. Nearly all of the rig was fabricated in plexiglass to a very high standard of accuracy (typically 0.5 m ± 0.125 mm diameter over a length of 1 m).

An annular tandem cascade, which could be sited at a number of positions relative to diffuser inlet, was incorporated in the entry length. It consisted of NACA 65-series blades with 10% thickness on circular arc camber lines. Mean pitch-chord ratio was 0.5, and the blade inlet and outlet angles were 30 deg and −6 deg, respectively. The distance between the trailing edge of the last row of blades and diffuser inlet could be varied between 3.05 and 0.05 blade chord lengths in steps of half a blade chord. The cascade was mounted in a carrier which could be rotated, using a micrometer mechanism, over a distance of six blade pitches. In this way a comprehensive set of experimental data could be obtained using one traverse station.

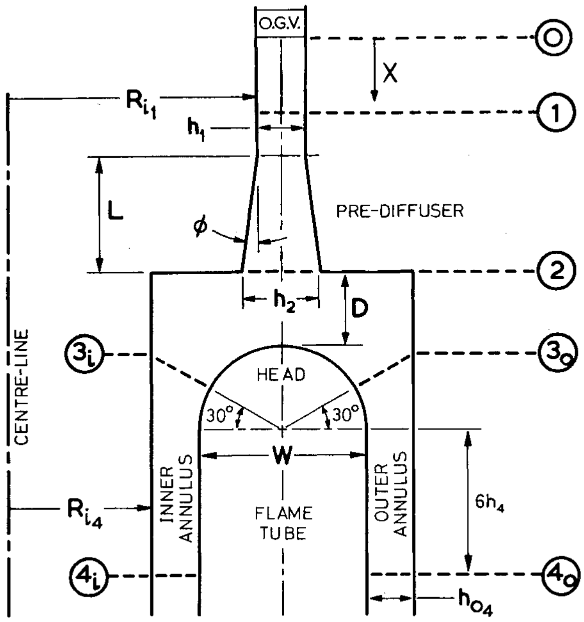

Fig. 2 Dump diffuser nomenclature.

The flow split S was controlled by a ring throttle at the exit of the outer annulus. The flame tube comprised a hemispherical head with parallel side walls, and all the flow passed to the annuli surrounding the flame tube. The geometry of the rig is detailed in Table 1 and Fig. 2. It will be noted that the geometry of the straight-wall prediffuser lies close to the optimum $C_p{}^*$ line defined by Sovran and Klomp.

Static pressure measurements were made along the walls of the entry length, prediffuser, and combustor outer casing. At each position three tappings were made, spaced equally around the surface of the inner and outer walls. Measurements were made at station 1, with a Cobra probe, to establish that the flow was essentially swirl free. Total pressure traverses were carried out at three circumferential positions, 120-deg apart, at stations 1, 2, and 4 (see Fig. 2). At station 1 traverses were carried out at circumferential locations extending over three blade pitches. The flattened total pressure tube had a wall thickness of 0.12 mm, and an opening of 0.60 mm by 1.2 mm. At station 4 the velocity profiles were calculated on the assumption that the static pressure along each radial traverse was the same as that measured at the wall. However, at stations 1 and 2 the velocity profiles were obtained using the measured static pressure variation across the annulus. Although an allowance was made for the displacement of the effective center of the pitot probe, no correction was applied to take account of the effects of turbulence. Considerable uncertainty surrounds the estimation of errors in pressure measurements when the turbulence level is high, and the indicated velocity near the walls, immediately downstream of the outlet guide vanes, may be approximately 20% too high. However, for the remainder of the flow the velocities are considered to be within 2%, and the integrated mass flows at all stations were within 3% of the value measured at diffuser inlet. The large number of pressure measurements necessitated the use of an automated data acquisition system. A Hewlett-Packard 2114B minicomputer was used to control the position of a traverse probe driven by a D.I.S.A. 52C01 stepping motor. A total of 32 readings of pressure were taken at 0.3/s intervals at each probe position. The computer then analyzed the data, and the results were presented on a teletype and as a visual display on a storage oscilloscope.

Performance Parameters

For incompressible spatially nonuniform flow the mass-weighted mean total pressure is defined as

$$\bar{P}_t = \frac{1}{m}\int_0^A P_t \mathrm{d}m = \frac{1}{m}\int_0^A (p + \tfrac{1}{2}\rho u^2)\mathrm{d}m \qquad (1)$$

where $\mathrm{d}m = \rho u \mathrm{d}A$. Equation (1) may be written as

$$\bar{P}_t = \bar{p} + \alpha \tfrac{1}{2}\rho U^2 \qquad (2)$$

were $\bar{p} = 1/m\int_0^A p dm$, U is the mass-derived mean velocity given $m/\rho A$, and $\alpha = 1/A\int_0^A (u/U)^3 \mathrm{d}A$. The velocity profile energy coefficient α expresses the ratio of the mass-weighted mean kinetic energy of a nonuniform flow to that of a uniform flow having the same mass flow rate. Designating $\Delta\bar{P}_{t_{1-2}}$ as the mean total pressure loss in the prediffuser, the energy equation for the flow may be written as

$$\bar{p}_1 + \alpha_1 \tfrac{1}{2}\rho U_1^2 = \bar{p}_2 + \alpha_2 \tfrac{1}{2}\rho U_2^2 + \Delta\bar{P}_{t_{1-2}} \qquad (3)$$

Defining the pressure recovery coefficient as the pressure rise nondimensionalized with respect to the inlet dynamic pressure, we obtain for the prediffuser

$$\bar{C}_{p_{1-2}} = (\bar{p}_2 - \bar{p}_1)/\alpha_1 \tfrac{1}{2}\rho U_1^2 \qquad (4)$$

Similarly, the total pressure loss coefficient is given by

$$\bar{\lambda}_{1-2} = (\bar{P}_{t_1} - \bar{P}_{t_2})/\alpha_1 \tfrac{1}{2}\rho U_1^2 \qquad (5)$$

Combining Eqs. (3-5) with the continuity equation, the pressure recovery of the prediffuser may be expressed as

$$\bar{C}_{p_{1-2}} = \left[1 - \frac{\alpha_2}{\alpha_1}\left(\frac{A_1}{A_2}\right)^2\right] - \bar{\lambda}_{1-2} \qquad (6)$$

In the series II tests the loss coefficient $\bar{\lambda}_{1-2}$ has been obtained from Eq. (6) using the measured values of pressure recovery and velocity profile energy coefficient. Unfortunately, the calculation of $\bar{\lambda}_{1-2}$ is very sensitive to changes in the values of α_2, α_1, and $\bar{C}_{p_{1-2}}$ and whereas the values of $\bar{C}_{p_{1-2}}$ are considered to be accurate to within 3%, the values of $\bar{\lambda}_{1-2}$ may be in error by as much as 15%.

The overall performance of the combustor-dump diffuser system is defined as

$$\bar{C}_{p_{1-4}} = [(m_i\bar{p}_i)_4 + (m_o\bar{p}_o)_4 - (m\bar{p})_1]/m_1\alpha_1 \tfrac{1}{2}\rho U_1^2 \qquad (7)$$

and

$$\bar{\lambda}_{1-4} = [(m\bar{P}_t)_1 - (m_i\bar{P}_{t_i})_4 - (m_o\bar{P}_{t_o})_4]/m_1\alpha_1 \tfrac{1}{2}\rho U_1^2 \qquad (8)$$

The values of $\bar{C}_{p_{1-4}}$ and $\bar{\lambda}_{1-4}$ are considered to be within 2% and 4%, respectively.

For the tests behind the multistage compressor the flow at diffuser inlet was compressible and the mean inlet dynamic pressure has been calculated using the expression

$$\bar{q} = \left[\int_0^A (P_t - p)\mathrm{d}m\right] m^{-1}$$

Results and Discussion

Test Series I

Inlet Conditions

The compressor was run along its design speed characteristic, and tests were performed at three mass flows to cover the operating range from near design to choking conditions. The mean conditions at the diffuser inlet are detailed in Table 2.

Table 2 Diffuser inlet conditions (test series I)

Compressor operating point	Mean Mach no., M_1	Blockage, B_1	Reynolds no., $Re_1 \times 10^{-5}$
A [near design]	0.29	0.11	7.4
B	0.35	0.125	8.6
C	0.44	0.12	10.0

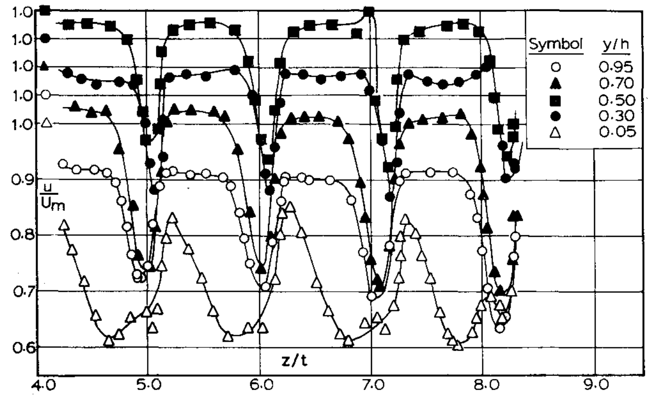

Fig. 3 Velocity distribution downstream of outlet guide vanes ($M_1 = 0.29$, $x/c = 0.305$, $U_m = 137$m/s).

The circumferential distribution of velocity over four blade pitches measured at station 0, with the compressor operating at point A is shown in Fig. 3. The local velocity was nondimensionalized with respect to the maximum velocity in the cross section. Measurements at all operating points confirmed that, within experimental error, the flow was essentially swirl free. With the compressor operating near design the blade wakes at the mid-annulus position are relatively thin and clearly defined, whereas near the walls they have a less coherent structure, lower momentum, and occupy an appreciable proportion of the blade passage. However, operation at point B produces relatively thicker wakes at the mid-annulus position. Examination of the radial traverses reveals, particularly for operation at point B, that midway between the blades the velocity profile is asymmetric. The velocity at a position 30% of the annulus height from the outer wall is higher than that measured at the same distance from the hub. This result is quite typical for this type of axial flow compressor. Radial traverses in the wakes of the outlet guide vanes indicate high velocities close to the inner wall, although the radial distortion is not as severe as that measured by Adenubi.

The distribution of velocity at diffuser inlet (station 1) is shown in Fig. 4. It will be observed that the plane of maximum velocity midway between the blade wakes has moved to a position 30% of the annulus height from the outer wall. Moreover, the wakes have spread out until they occupy the whole of the passage and the momentum of the flow near the hub is still considerably lower than that near the outer wall.

Apart from the reduction in blockage that occurs as the wakes mix out in the passage between the outlet guide vanes and diffuser inlet, it is interesting to compare the rate of decay with that measured for an isolated wake by Hill et al.[8] It will

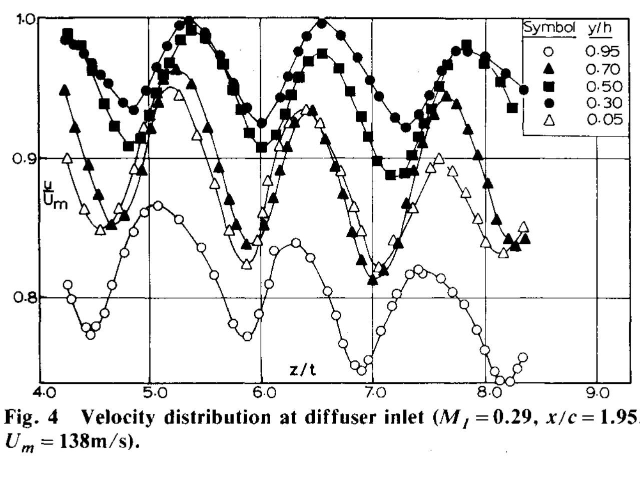

Fig. 4 Velocity distribution at diffuser inlet ($M_1 = 0.29$, $x/c = 1.95$, $U_m = 138$m/s).

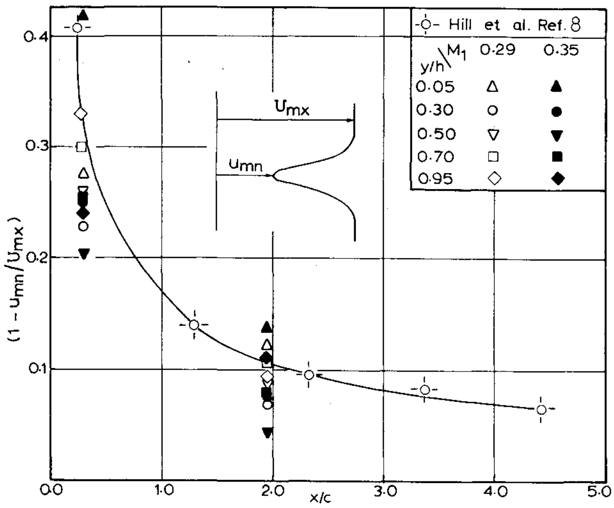

Fig. 5 Wake decay rate.

be seen from Fig. 5 that despite interaction with the annulus wall boundary layers, the rates of decay are not untypical of those measured by Hill et al. and that much of the wake decays within a length of two blade chords.

Overall Performance

The overall performance of the dump diffuser at the design flow split and dump gap is summarized in Table 3, also included is the performance obtained with fully developed flow at inlet. It will be seen that at approximately the same inlet blockage the overall loss and pressure recovery measured downstream of the compressor is, within experimental error, the same as that obtained with fully developed flow at inlet. Although the blockage fractions are almost the same, the Reynolds numbers and turbulence structures are different. A mid-annulus turbulence level of 5-6% has been recorded by Schlichting and Das[9] downstream of a multistage compressor, which is higher than the 2-3% associated with fully developed flow. Nonetheless, the result, which is in agreement with the work of Adenubi, suggests that in situations where there is considerable wake decay before diffuser inlet, the performance measured with fully developed inflow is the same as that obtained downstream of a compressor. The variation in diffuser performance with compressor operating point would be expected to occur as a result of changes in any or all of the following inlet flow parameters; boundary-layer thickness described by the blockage, turbulence structure, periodic unsteadiness associated with the rotor, circumferential and radial distortion of the velocity profile, and Reynolds number. No information is available on the way in which the total pressure loss is influenced by Reynolds number, but in the case of operation downstream of a compressor it is not thought to be a significant parameter. Also the variation in inlet blockage from 0.11 to 0.125 is not considered to be sufficiently large to be the major parameter influencing performance. However, as the compressor operating point moves away from design (point A) toward choking conditions the outlet guide vanes will operate near to the critical Mach number and at negative incidence. This leads to increased areas of separated flow at the end walls of the blade passages. Under these conditions it seems reasonable to assume that flow distortions downstream of the blades will increase. It is therefore suggested that the reduction in diffuser performance as the compressor operating point moves away from design is due mainly to the effects of increased blockage and flow distortion.

The variation of performance with flow split at constant dump gap for fully developed inflow was very similar to that measured by Goom[10] who tested a dump system having the same design flow split. In particular, the overall total pressure loss was almost constant over a wide range of flow splits. Of particular importance was the improvement in pressure distribution around the head of the flame tube. Previous work by Fishenden and Stevens[1] identified regions in which severe local reductions in pressure occurred. In the work reported here the pressure in these regions was increased considerably. This improvement is partly attributed to the use of a trumpet-shaped prediffuser.

Test Series II

Throughout the test program, which is summarized in Table 4, an almost constant dynamic pressure was maintained in the center of the annulus at a position 76-mm upstream of the cascade. This corresponded to a velocity of 25m/s and a Reynolds number based on the inlet hydraulic diameter of 1.4×10^5. The velocity profile at inlet to the cascade was symmetrical about the mean diameter and had a blockage fraction of 0.125 and an energy coefficient of 1.06. Tests were carried out for two positions of the cascade, namely 0.05 and 3.05 blade chord lengths upstream of the diffuser inlet plane.

Inlet Conditions

The distribution of axial velocity measured at the prediffuser inlet, with the cascade 0.05 blade chord lengths upstream, indicated that on the suction side of the blade, near the trailing edge, separation had occurred at the inner wall. This is due to the low Reynolds number and the influence of the wakes from the first row of blades. The radial distribution of velocity at midblade pitch was very similar to that measured in the series I tests. A thicker boundary layer was present on the hub, and the position of maximum velocity was displaced toward the outer wall. Velocity contours at diffuser inlet with the cascade some distance upstream showed that much of the distortion associated with blade wakes had decayed away. This is confirmed by the blockage fraction which has decreased from 0.17 to 0.11 (see Table 5). The increased thickness of the boundary layer on the suction side of the blade leads to the flow at diffuser inlet having a swirl angle of about 4 deg. Yawmeter measurements confirmed this value, which was calculated from the knowledge that in a distance of three blade chords the wake had moved from the trailing edge of the blade to the mid-pitch position. However, this amount of swirl will not significantly affect the results.

Owing to circumferential and radial momentum transfer reducing the momentum coefficient of the flow, there is an

Table 3 Overall performance (test series I)[a]

Compressor operating point	Mean inlet Mach no., M_I	Blockage, B_I	Reynolds no., $Re_I \times 10^{-5}$	$\bar{C}_{p\,I\text{-}4}$	$\bar{\lambda}_{I\text{-}4}$
A	0.29	0.11	7.4	0.64	0.20
B	0.35	0.125	8.6	0.57	0.27
C	0.44	0.12	10.0	0.58	0.26
Fully developed flow	0.09	0.13	1.0	0.58	0.26

[a] $S = 1.0$, $D/h_2 = 0.47$.

Table 4 Summary of tests (series II)

Test conditions	Flow split, S	Dump gap, D/h_2
Cascade at $x/c = 0.05$	0.68, 1.18, 1.33, 1.66, 1.81	0.7
Cascade at $x/c = 3.05$	0.84, 1.21, 1.47, 1.54, 1.92	0.7
Fully developed flow[10]	0.60, 1.02, 1.40, 1.90	0.7

Table 5 Overall performance (test series II)[a]

Test conditions	B_1	α_1	$\bar{C}_{p_{1\text{-}4}}$	$\bar{\lambda}_{1\text{-}4}$	α_{4_i}	α_{4_o}
Cascade at $x/c = 0.05$	0.17	1.19	0.52	0.23	1.08	1.10
Cascade at $x/c = 3.05$	0.11	1.04	0.53	0.21	1.08	1.08
Fully developed flow[10]	0.125	1.06	0.t5	0.19	1.04	1.01

[a] $S^* = 1.20$, $D/h_2 = 0.70$.

increase in static pressure between stations 0 and 1 ($\bar{C}_{p_{0\text{-}1}} \simeq 0.10$). The turbulent mixing associated with the distortion of the flow leads to an increased dissipation and a loss coefficient $\bar{\lambda}_{0\text{-}1}$ of approximately 0.025. These values are only approximate, since the calculations are very sensitive to minor errors in the estimation of the pressure rise, which is difficult to measure accurately in this region of the flow.

Overall Performance

The dependence of overall performance on flow split, for two positions of the cascade, is shown in Fig. 6. The trends are very similar to those discussed in Ref. 1, in that the overall total pressure loss does not change significantly with flow split over the normal working range. The variation of pressure recovery can be described by writing the energy equation as

$$m_1(\bar{p}+\alpha \tfrac{1}{2}\rho U^2)_1 = \left[m_i(\bar{p}+\alpha\tfrac{1}{2}\rho U^2)_i + m_o(\bar{p}+\alpha\tfrac{1}{2}\rho U^2)_o\right]_4 - m_1\Delta\bar{P}_{t_{1\text{-}4}} \quad (9)$$

Combining with Eqs. (5) and (6), we may write

$$\bar{C}_{p_{1\text{-}4}} = 1 - \frac{[(m\alpha\tfrac{1}{2}\rho U^2)_i + (m\alpha\tfrac{1}{2}\rho U^2)_o]_4}{(m\alpha\tfrac{1}{2}\rho U^2)_1} - \bar{\lambda}_{1\text{-}4} \quad (10)$$

Restricting the analysis to ideal flow conditions, we may assume uniform flow at inlet and outlet, i.e., $\alpha_1 = \alpha_{i_4} = \alpha_{o_4} = 1.0$, and no losses $\bar{\lambda}_{1\text{-}4} = 0$. Then Eq. (10) becomes

$$\bar{C}'_{p_{1\text{-}4}} = 1 - \frac{[(m_iU_i^2)+(m_oU_o^2)]_4}{m_1U_1^2} \quad (11)$$

Since $S = (m_o/m_i)_4$ and $m_1 = m_o + m_i$, it can be shown that the ideal pressure recovery coefficient is given by

$$\bar{C}'_{p_{1\text{-}4}} = 1 - \left(\frac{1}{1+S}\right)^3\left[\left(\frac{A_1}{A_i}\right)_4 + S^3\left(\frac{A_1}{A_o}\right)_4\right] \quad (12)$$

Thus, the ideal pressure recovery depends not only on the area ratios of the annuli surrounding the flame tube but also on the proportion of flow passing to each annulus. It is usual to design for the same mean velocity in each annulus, i.e. $U_{o4} = U_{i4}$, in which case the design flow split $S^* = (A_o/A_i)_4$. Equation (12) indicates a similar variation to that shown in Fig. 6 in that at off-design flow splits the pressure recovery decreases. This occurs because, if the flow to one annulus is greater than the design value, the pressure will fall and, although the pressure may rise in the other annulus, the net result is a reduction in the mass-weighted mean pressure rise. The overall performance at the design flow split is summarized in Table 5.

The performance achieved with the cascade sited far enough upstream from diffuser inlet ($x/c = 3.05$) for appreciable wake decay to occur is, within experimental error, the same as that obtained with fully developed inflow. Thus, the findings of the series I tests and those of Adenubi are confirmed. With the cascade very close, the loss is slightly higher, but the value of $\bar{\lambda}_{1\text{-}4} = 0.23$ includes the turbulent mixing loss associated with the wakes which, in an annulus of constant area, has been shown to give $\bar{\lambda}_{0\text{-}1} \simeq 0.025$. If this loss is added to the diffuser loss when the cascade is sited at $x/c = 3.05$ then $\bar{\lambda}_{0\text{-}4} = 0.235$, and the overall loss from the exit plane of the cascade is, within experimental error, the same in both cases. Therefore, no significant penalty in overall performance is incurred if the diffuser is operated with the cascade close to the inlet plane.

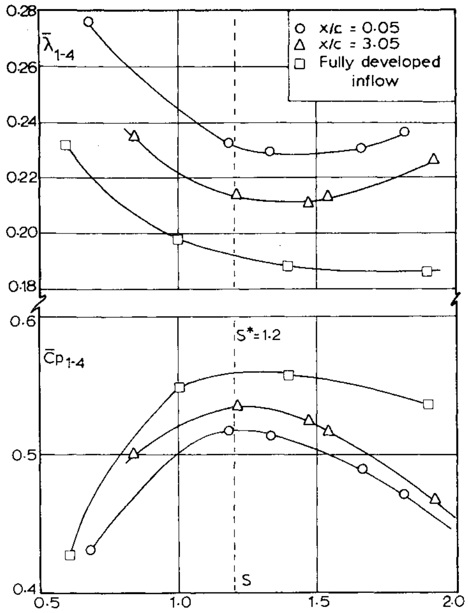

Fig. 6 Overall performance (series II tests, $D/h_2 = 0.7$).

From Table 5 it will be seen that the dump diffuser loss with fully developed inflow ($\bar{\lambda}_{1\text{-}4} = 0.19$) is increased by 20% to $\bar{\lambda}_{1\text{-}4} = 0.23$ when the cascade is sited at $x/c = 0.05$. This is much lower than the increase, typically 50%, quoted by Klein et al. who tested a combustor-dump diffuser with an annular tandem cascade sited at $x/c = 0.25$. The reason for this difference can only be established after a more detailed analysis of the prediffuser performance.

With the cascade at $x/c = 3.05$, no evidence of the wakes could be detected at the end of the settling length (station 4) although with the cascade at $x/c = 0.05$ a circumferential variation in velocity, typically 5%, was observed. Finally, it is interesting to compare the overall performance of the diffusers used in the two series when operating with fully developed inflow. Table 3 and Table 5 show that, owing to a large increase in loss, only a small fraction of the anticipated increase in overall pressure recovery, due to increasing the overall area ratio from 2 to 2.5, has been achieved. This is attributed to increased losses in the dump region brought about by having to operate at a small dump gap in order to stabilize the flow in the trumpet-shaped prediffuser.

Prediffuser

Equation (6) shows that the pressure recovery will be lowered by distortion of the velocity profile (i.e., $\alpha_2/\alpha_1 > 1$) and by losses ($\bar{\lambda}_{1\text{-}2} > 0$). Since diffusion requires a reduction in the flux of kinetic energy and (α-1) represents the excess of kinetic energy over the minimum possible at a given flow rate, any increase in α_2 above 1.0 represents a reduction in the amount of diffusion that takes place. The last term $\bar{\lambda}_{1\text{-}2}$ represents the loss of available energy due to viscous dissipation. The prediffuser performance is presented in Fig. 7. Owing to the relatively low losses, the variation in pressure

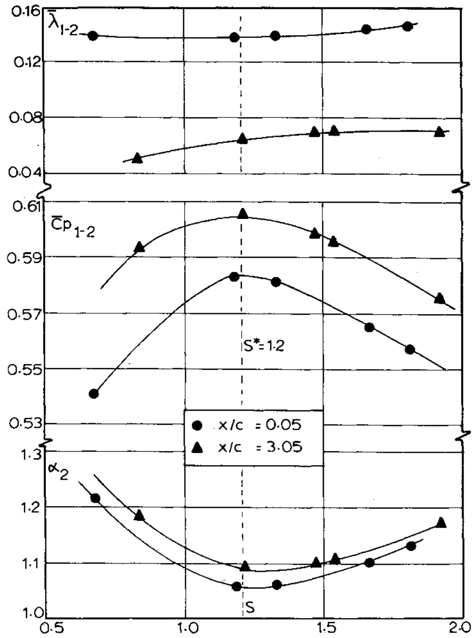

Fig. 7 Prediffuser performance (series II tests, $D/h_2 = 0.7$).

recovery with flow split is attributed to changes in the flux of outlet kinetic energy α_2. The performance at the design flow split is summarized in Table 6.

Because the mixing associated with the blade wakes increases the radial momentum transfer to the near wall regions, there is a significant reduction in the distortion of the outlet velocity profile α_2. In fact, with $x/c = 0.05$ the distortion of the inlet velocity profile is actually reduced. Comparison of the outlet velocity profile with that measured with fully developed inflow indicated that although there was still some evidence of blade wakes at the prediffuser exit, they had not become more pronounced as suggested by Klein et al. Whether the wakes decay or grow depends on the relative magnitude of the pressure and shear forces and the length of the flow path. If the axial pressure gradient is large enough, the wakes may grow rather than decay, so that a zone of stagnant or reversed flow develops. This possibility has been demonstrated experimentally by Hill et al. However, it may be important to note that, in the experiments of Hill et al., diffusion occurred in the plane of the wake. This is not the case in the present tests where diffusion takes place in a plane normal to that of the wakes, and Viets[11] has shown theoretically that in these circumstances the mixing is improved. Therefore, although the wakes have decayed in these tests, the prediffuser is an optimum design (as defined by the tests of Fishenden and Stevens); whereas if short, wide-angle diffusers of the type tested by Klein et al. are used, the wakes may become more pronounced and lead to local flow separation.

Table 6 shows that, although the prediffuser losses are increased by the mixing associated with the wakes, the pressure recovery is still improved because of the reduction in the distortion of the outlet velocity profile. It is interesting to compare the prediffuser performance with that of the system as a whole. Because of the reduction in the distortion (kinetic energy) of the prediffuser outlet velocity when the cascade is near to diffuser inlet ($x/c = 0.05$), the loss in the dump region is reduced. Thus, an increased prediffuser loss is offset by a reduction in the dump loss. Nonetheless, the overall pressure recovery is lower than that achieved in the prediffuser.

Conclusions

Experiments have been carried out to investigate the way in which compressor exit conditions influence the performance of optimum combustor-dump diffusers. Two diffuser systems have been tested, one sited at exit from a multistage axial flow compressor, the other downstream of an annular tandem cascade. Comparisons have been made with the performance obtained when operating with fully developed flow at the diffuser inlet. These tests have shown that:

1) The overall performance, measured when there is a considerable decay of the wakes from the outlet guide vanes before they reached diffuser inlet, is almost the same as that achieved with fully developed inflow.

2) Only a small reduction in overall performance is incurred even when the outlet guide vanes are positioned near to the inlet of the diffuser.

3) A reduction in the distortion and an improvement in the stability of the prediffuser outlet velocity profile are obtained when the outlet guide vanes are sited close to the diffuser inlet.

4) The use of a trumpet-shaped prediffuser improves the pressure distribution around the head of the flame tube.

Table 6 Prediffuser performance (test series II)[a]

Test	B_1	α_1	α_2	$\bar{C}_{P1\text{-}2}$	$\bar{\lambda}_{1\text{-}2}$
Cascade at $x/c = 0.05$	0.17	1.19	1.06	0.58	0.14
Cascade at $x/c = 3.05$	0.11	1.04	1.10	0.60	0.07
Fully developed flow[10]	0.125	1.06	1.38	0.54	0.06

[a] $S^* = 1.2$, $D/h_2 = 0.7$.

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