Performance of Annular Combustor-Dump Diffusers

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In this paper, results of an experimental investigation of the performance of an annular combustor-dump diffuser are presented. The geometry comprised a straight-walled prediffuser followed by a sudden expansion in which the flow divides and passes to the two concentric annuli surrounding the flame tube. Tests were carried out to investigate the influence of: prediffuser geometry; the distance between prediffuser outlet and the head of the flame tube; and the division of flow between the two concentric annuli. The system is shown to offer good performance and flow stability over a wide range of operating conditions. The way in which the presence of the flame tube improves the performance and stability of flow in the prediffuser is demonstrated. Whereas nearly all of the pressure rise occurs in the prediffuser, most of the total pressure loss occurs near the head of the flame tube. Minimum total pressure loss and a symmetrical pressure distribution on the head of the flame tube were obtained with a symmetrical velocity profile at the prediffuser outlet. The results also focus attention on the need to carefully match the geometries of the prediffuser, flame tube, and surrounding annuli.

Nomenclature

A = area of cross section
AR = area ratio, A_2/A_1
B = blocked area fraction, $1 - 1/A \int_0^A (u/U_m)\, dA$
C_p = pressure recovery coefficient
D = distance between head of flame tube and outlet of prediffuser (dump gap)
h = annulus height
H = boundary layer shape parameter, δ^*/θ (axisymmetric definitions)
L = length of prediffuser
m = mass flow rate, $\rho A U$
p = static pressure
P_t = total pressure
ΔP_t = loss of total pressure
q = dynamic pressure
R = radius
R_s = radius of dividing streamline
S = flow split, $(m_o/m_i)_4$
S^* = design flow split
u = local axial velocity
U = mass derived mean velocity, $m/\rho A$
U_m = maximum velocity in cross section
W = width of flame tube
y = distance perpendicular to wall
α = velocity profile energy coefficient, $1/A \int_0^A (u(U)^3 dA$
λ = loss coefficient
ρ = fluid density
ϕ = diffuser wall angle

Subscripts

1 = prediffuser inlet
2 = prediffuser outlet
3 = head of flame tube
4 = settling length
i = inner annulus flowfield
0 = outer annulus flowfield

Superscripts

$-$ = mass-weighted mean value
$'$ = ideal value

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Introduction

IN the combustion system of a gas turbine, it is necessary to diffuse the air delivered by the compressor in order to promote efficient combustion and avoid large total pressure losses. A typical annular turbojet combustor diffuser 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. Thus, one of the main functions of a combustor diffuser is to efficiently decelerate the air that passes to the annuli surrounding the flame tube so that sufficient pressure is available to achieve adequate penetration of the dilution jets. Failure to achieve this criterion can have serious effects on the temperature distribution at outlet from the combustor.

The "faired" diffuser shown in Fig. 1a is widely used in aircraft gas turbines. In this system, some diffusion is carried out prior to the dividing lip and then continued in the passages that feed the inner and outer annuli. At large values of compressor exit hub/tip ratio, the flow in annular diffusers can, to a first approximation, be considered as two-dimensional, and many designs have been based on the two-dimensional tests of Kline et al.[1], and Wolf and Johnston.[2] In these tests, the pressure recovery characteristics and flow regimes were thoroughly investigated over a wide range of inlet conditions. More recently, Sovran and Klomp,[3] and Howard et al.[4], have investigated the performance of a wide range of annular diffusers. The optimum performance at a fixed nondimensional length L/h_1 measured in terms of the pressure recovery coefficient was found to occur at an area ratio approximately in-

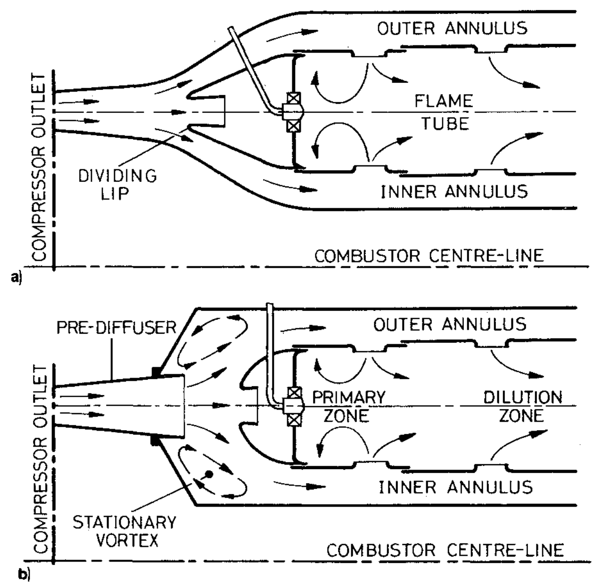

Fig. 1 Two types of combustor diffuser system. a) "faired" diffuser; b) dump diffuser.

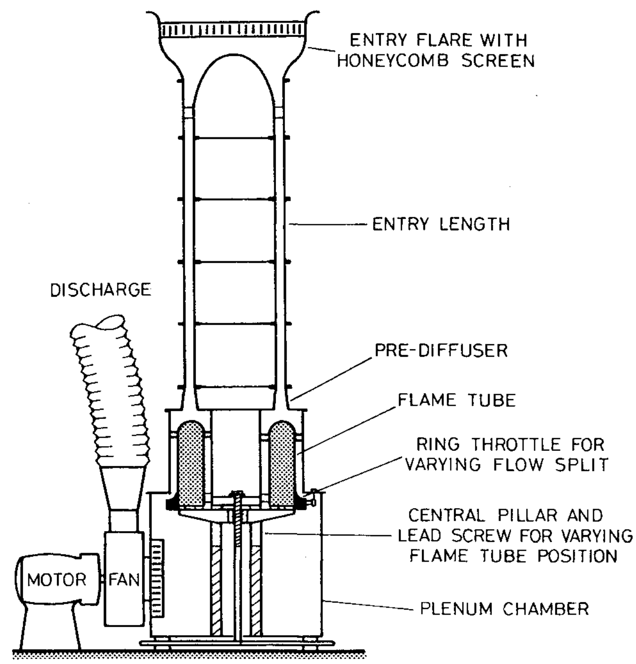

Fig. 2 Test facility

dependent of the combination of wall angles and hub/tip ratio employed.

In aircraft applications where overall length is of primary importance, the bulk of the flow in a "faired" diffuser may have to turn through an angle of up to 40° in the plane upstream of the dividing lip. Under these conditions, Stevens and Fry[5] have shown that, owing to increased turbulent mixing caused by the curvature of flow, the loss in the subsequent diffusion process is considerably increased. Additional losses will also be caused by the struts that surround the fuel feed pipes. Furthermore, the advent of the high bypass ratio engine, in which a relatively low gas generator flow is compressed to a high pressure ratio, has led to very small annulus heights in the diffusers feeding the inner and outer annuli. Therefore, manufacturing tolerances and differential expansion and distortion during operation can give rise to significant variations in diffuser geometry.

In an attempt to overcome these difficulties, particularly those associated with small annulus heights, the dump diffuser shown in Fig. 1b has been developed. Initially the flow is decelerated in a short prediffuser and then discharged into the dump region where a free-surface diffusion continues around the head of the flame tube. The basic principle of this type of design is that the presence of the flame tube can be used to modify the flow in the prediffuser. If, for instance, the head of the flame tube is brought very close to the prediffuser, it is quite possible for the flow to be accelerated at "diffuser" exit; too far away, and the diffuser will behave as though it were in isolation. By correct positioning of the flame tube, a short wide angle prediffuser can be made to operate as a diffuser of effectively smaller angle. However, if a significant amount of diffusion is still required downstream of the prediffuser, then, because of the curvature of the flow in the dump region, this will be carried out at an inherently lower efficiency. Nonetheless, a particularly stable flow will result if a stationary vortex is formed by suitable shaping of the outer wall of the combustor.

Biaglow[6] tested a two-dimensional model of a combustor incorporating a dump diffuser and showed the performance of the system to be comparatively insensitive to changes in the velocity profile at prediffuser inlet. This is an especially desirable feature since changes in engine operating conditions produce different velocity profiles at compressor exit. The influence of inlet conditions has also been investigated by Klein et al.[7] In some engine designs, the distance between diffuser inlet and compressor exit is very short, in which case the wakes and vorticity produced by the last row of blades can be expected to have a significant effect on diffuser performance. Klein et al. simulated compressor exit conditions by introducing an annular tandem cascade, giving zero outlet 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 of inlet nonuniformity produced by an approach length of 22 inlet annulus heights. The pressure recovery and total pressure loss was measured, and the influence of prediffuser geometry, the division of flow between the two annuli surrounding the flame tube‡, and the distance between the head of the flame tube and the prediffuser§ was investigated. The results indicated that pre-diffuser wall angles up to 12.5 deg could be used without incurring any significant increase in the overall total pressure loss, although the losses were higher than those of an equivalent "faired" diffuser. Whereas the loss was comparatively insensitive to changes in flow split and dump gap when operating with a boundary layer type of inlet nonuniformity, a significant variation was noted when operating with blade wakes.

The tests due to Klein et al. represent an important step in establishing the performance of dump diffusers. However, there is a need for more detailed studies in order to gain a better understanding of the flow behavior. This is particularly important in view of the large number of geometric and flow variables. Attention to date has been focused on the overall performance, but it is equally important to obtain a near-symmetrical pressure distribution around the head of the flame tube, since severe local reductions in static pressure could lead to a reversal in the direction of the cooling flows and flame tube overheating. A detailed series of tests have therefore been conducted on a range of dump diffusers to establish the influence of geometric and flow variables. Particular attention has been paid to the pressure distribution around the head of the flame tube, and to the performance of the various components.

Test Facility

The test facility is shown in Fig. 2. Air was drawn through the rig in order to avoid uncertainty associated with the inlet turbulence structure and symmetry of flow presented by a

‡Henceforth referred to as "Flow Split;" $S = (m_0/m_i)_4$.

§"Dump Gap," D.

Table 1 Prediffuser geometries

Diffuser ref. no.	A_2/A_1	2ϕ	L/h_1
1	1.4	12°	1.900
2	1.6	12°	2.850
3	1.8	12°	3.805
4	1.8	18°	2.525

"blown" system. 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 an internal contraction ratio of 8:1, into an annular entry length some 48 annulus heights long. Stable transition was ensured by trip wires on the inner and outer walls just downstream of the intake throat. Nearly all of the rig was fabricated in perspex to a very high standard of accuracy (typically 20.000 in. ±0.005 in. diam over a length of 30 in.).

The flame tube was mounted on four struts joined to a boss located on a lead screw passing down the central pillar. Adjustment of the dump gap, *D*, was achieved by rotating a wheel fixed to the lower end of the lead screw. Concentricity of the flame tube with the inner and outer walls of the combustion chamber was maintained by three struts equi-spaced around each of the annuli. 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. All the flow passed to the annuli surrounding the flame tube, and no attempt was made to simulate the flow normally entering via ports and cooling rings. The geometry of the prediffusers and the remainder of the rig is detailed in Table 1, Fig. 3. It will be noted that diffuser Nos. 1,2 and 3 lie close to the optimum Cp^* line defined by Sovran and Klomp.

Static pressure measurements were made at over one hundred positions along the walls of the prediffuser and combustor. At each position, three tappings were made spaced equally around the surface of the inner and outer walls. Total pressure traverses were carried out at three circumferential positions 120° apart, at Stations 1, 2 and 4 (see Fig. 3); and fixed rakes were mounted on the head of the flame tube at

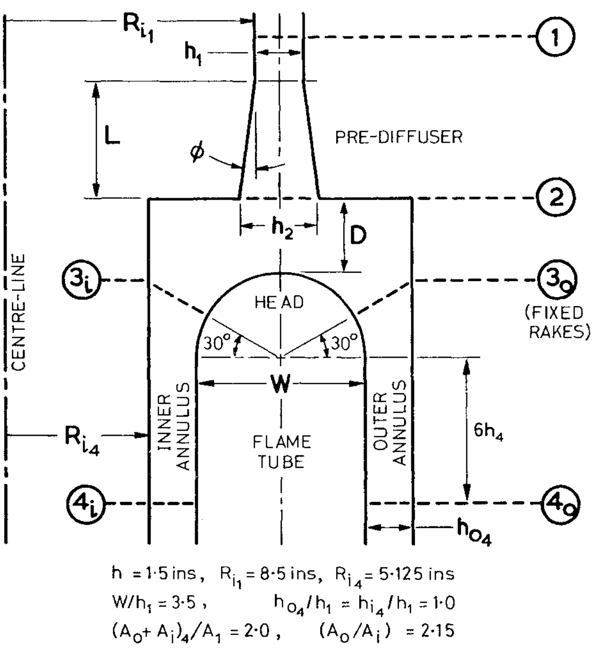

Fig. 3 Dump diffuser nomenclature.

Station 3. The flattened total pressure probes had a wall thickness of 0.005 in. and an opening of 0.040 in. by 0.015 in. All pressures were recorded on a Furness Controls micromanometer.

At Stations 1 and 4, the velocity profiles were calculated on the assumption that the static pressure along each traverse was the same as that measured at the wall. At Stations 2 and 3, the velocity profile was obtained using the measured static pressure variation across the annulus. The pressure variation was measured using a wedge probe developed by Girerd and Guienne.[8] Whereas an allowance was made for the displacement of the effective center of the pitot probe, no corrections were applied to take account of the effects of turbulence. Considerable uncertainty surrounds the estimation of errors in pitot tube measurements of mean velocity when the local turbulence level is high.[9] Nonetheless, velocity profiles taken at the three circumferential locations at prediffuser outlet exhibited excellent symmetry of flow, and the integrated mass flows were within 3% of the values measured at inlet.

Performance Parameters

Owing to the complex flow interaction at prediffuser outlet, the influence of dump gap cannot be predicted theoretically. However, the influence of flow split can be demonstrated by considering the ideal pressure recovery of a branched diffuser system.

The mass-weighted mean total pressure at a cross section is defined as

$$\bar{P}_t = (1/m)\int_0^A \left(p + \frac{1}{2}\rho\eta^2\right)dm \tag{1}$$

where $dm = \rho u dA$.
For incompressible spatially, nonuniform flow Eq. (1) may be written as

$$\bar{P}_t = \bar{p} + \alpha\frac{1}{2}\rho U^2 \tag{2}$$

where

$$\bar{p} = (1/m)\int_0^A p\, dm$$

U is the mass-derived mean velocity given by $m/\rho A$, and

$$\alpha = (1/A)\int_0^A (u/U)^3 dA$$

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 $\Delta\bar{P}_{t\,1-4}$ as the mean total pressure loss between the inlet and settling length, the energy equation for the system is

$$m_I\left(\bar{p} + \alpha\frac{1}{2}\rho U^2\right)_I = \left[m_i\left(\bar{p} + \alpha\frac{1}{2}\rho U^2\right)_i\right]_4$$

$$+\left[m_o\left(\bar{p} + \alpha\frac{1}{2}\rho U^2\right)_o\right]_4 + m_I\Delta\bar{P}_{t_{I-4}}$$

defining $\bar{C}_{p_{I-4}}\,[(m_i\bar{p}_i)_4 + (m_o\bar{p}_o)_4 - m_I\bar{p}_I]/m_I\alpha_I\,½\rho U_I^2$ and $\bar{\lambda}_{I-4} = \Delta\bar{P}_{t_{I-4}}/\alpha_I\,½\rho U_I^2$ then

$$\bar{C}_{p_{I-4}} = 1 - [m_i(\alpha\,½\rho U^2)_i$$

$$+ m_o(\alpha\,½\rho U^2)_o]_4/m_I\alpha_I\,½\rho U_I^2 - \bar{\lambda}_{I-4} \tag{3}$$

Restricting the discussion to ideal flow conditions, we may assume uniform flow at inlet and outlet i.e. $\alpha_1 = \alpha_{o_4} = \alpha_{i_4} = 1.0$ and no losses $\bar{\lambda}_{1-4} = 0$. Then Eq. (3) becomes

$$\bar{C}'_{p\,I-4} = 1 - [(m_i U_i^2)_4 + (m_o U_o^2)_4]/m_I U_I^2 \tag{4}$$

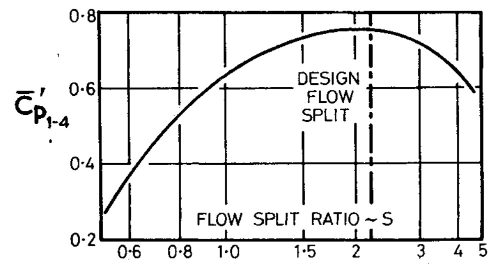

Fig. 4 Ideal pressure recovery.

Since

$$S=\left[\frac{m_o}{m_i}\right]_4=\left[\frac{U_oA_o}{U_iA_i}\right]_4$$

and $m_l=(m_o+m_i)_4$ it can be shown that the ideal pressure recovery is given by

$$\bar{C}'_{p\,l-4}=1-[(A_l/A_{i4})+S^3(A_l/A_{o4})]/(1+S)^3 \qquad (5)$$

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 $S^*=(A_o/A_i)_4$. Clearly, 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 coefficient. For the geometry under consideration, $S^*=2.15$, the variation of $\bar{C}_{p\,l-4}$ with flow split S is shown in Fig. 4.

Results and Discussion

Throughout the test program, which is summarized in Table 2, an almost constant, dynamic pressure was maintained in the center of the annulus at a position 3.0 in. upstream of the diffuser inlet plane. This corresponded to a Mach number of 0.09 and a Reynolds number, based on the inlet hydraulic diameter of $1.9\times10.^5$ Although the inlet Mach number is lower than that achieved in practice, Nelson and Popp[11] have shown that increases in Mach number as high as 0.7 have little effect on diffuser performance; however, this value is likely to be reduced as the prediffuser wall angle is increased. The inlet velocity profile was symmetrical about the mean diameter and had a blockage fraction $B_l=0.126$ and $\alpha_1=1.062$.

Overall Performance

The dependence of system performance on flow split and dump gap is shown in Fig. 5 for two of the prediffusers detailed in Table 1. It is seen that the total pressure loss coefficient at intermediate and large dump gaps is almost independent of flow split, confirming the findings of Klein et al. However, at small dump gaps, the loss is greater and does

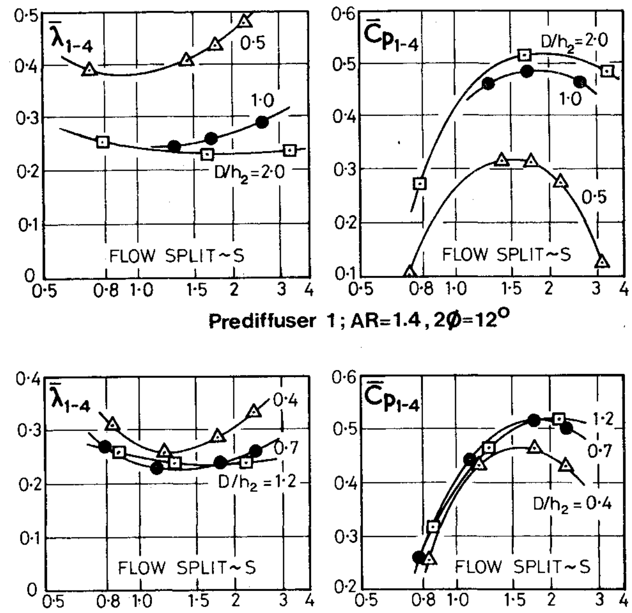

Fig. 5 Overall performance characteristics. Prediffuser 4, $AR=1.8$, $2\phi=18°$.

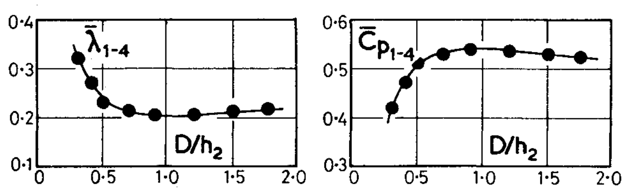

Fig. 6 Typical variation of performance with dump gap (prediffuser 3, $S=1.6$).

vary; the minimum occurring at a flow split of about 1.2. It is interesting to note that the general level of loss is similar to that recorded for a "faired" system by Stevens and Fry. This is in contrast to the high values measured by Klein et al. when operating with similar inlet conditions. The probable explanation is that the high losses obtained by Klein et al. were caused by the use of prediffusers with low area ratios, typically 1.28. The difference between the actual and ideal pressure recoveries at the design flow split is almost entirely attributable to inefficient diffusion represented by the loss coefficient $\bar{\lambda}_{1-4}$. In view of the near-constant loss characteristic and the low velocity profile energy coefficients (typically $\alpha_4=1.03$) measured in the settling length, the variation of pressure recovery with flow split is similar to that illustrated in Fig. 4. It is also worth recording that no instability of the type described by Ehrich[12] was observed.

A typical example of the way in which the dump gap affects the performance of the system is shown in Fig. 6. The rapid reduction in pressure recovery at small dump gaps is at-

Table 2 Summary of tests

Diffuser ref. no.	Nondimensional dump gap (D/h_2) and flow split ratio (S)		
1	$D/h_2=0.5$ $S=0.71, 1.42, 1.75, 2.20, 3.23$	$D/h_2=1.0$ $S=1.30, 1.72, 2.57$	$D/h_2 2.0$ $S=0.79, 1.64, 3.40$
2	$D/h_2=0.5$ $S=0.82, 1.20, 1.71, 2.48$	D/h_2 $S=0.81, 1.21, 1.54, 2.12$	$D/h_2=1.5$ $S=0.88, 1.47, 2.30$
3	$D/h_2=0.4$ $S=0.83, 1.26, 1.67, 2.36$	$D/h_2=0.7$ $S=0.78, 1.17, 1.82, 2.30$	$D/h_2=1.2$ $S=0.84, 1.37, 2.32$
4	D/h_2 $S=0.83, 1.19, 1.77, 2.25$	$D/h_2=0.7$ $S=0.78, 1.13, 1.77, 2.27$	$D/h_2=1.2$ $S=0.85, 1.27, 2.16$

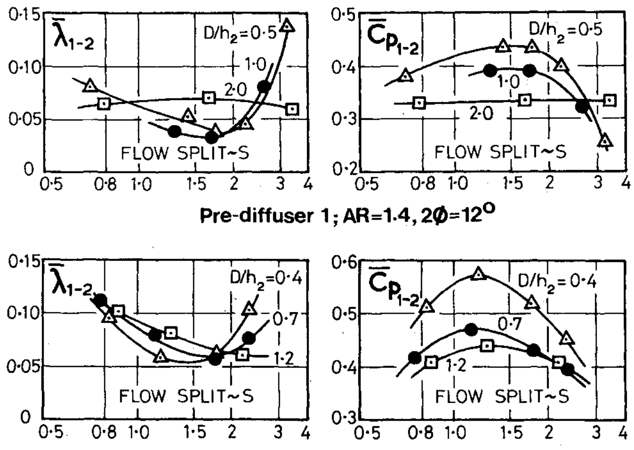

Fig. 7 Prediffuser performance characteristics. Prediffuser 4, $AR=1.8, 2\phi=18°$.

tributable to increased losses. However, the location of the region of increased loss can only be established after a more detailed analysis of the results.

Prediffuser

Writing the energy equation for the flow in the prediffuser as

$$(\bar{p}_1 + \alpha_1 \frac{1}{2} \rho U_1^2) = (\bar{p}_2 + \alpha_2 \frac{1}{2} \rho U_2^2) + \Delta \bar{P}_{t1-2}$$

Continuity gives $A_1 U_1 = A_2 U_2$ and defining

$$\bar{\lambda}_{1-2} = \Delta \bar{P}_{t1-2} / \alpha_1 \frac{1}{2} \rho U_1^2$$

then

$$\bar{C}_{p_{1-4}} = \frac{\bar{p}_2 - \bar{p}_2}{\alpha_1 \frac{1}{2} \rho U_1^2} = \left[1 - \frac{\alpha_2}{\alpha_1} \left(\frac{A_1}{A_2}\right)^2\right] - \bar{\lambda}_{1-2} \qquad (6)$$

This equation shows that the pressure recovery will be reduced by a distortion of the velocity profile (i.e. $\alpha_2/\alpha_1 > 1$) and/or by losses (i.e. $\bar{\lambda}_{1-2} > 0$). Since diffusion requires a reduction in kinetic energy flux and $\alpha - 1$ represents the excess of flux at any cross section over the miniumun possible value at a particular flow rate, any increase in α along the length of a diffuser represents a reduction in the amount of diffusion taking place. The first term, therefore, indicates a lowering of the pressure recovery due to insufficient diffusion. The second term represents the loss of available energy due to viscous effects, i.e. inefficient diffusion.

The loss coefficient $\bar{\lambda}_{1-2}$ has been obtained from Eq. (6) using the measured value 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_1 and $\bar{C}_{p\,1-2}$, which are difficult to obtain accurately owing to flow curvature and nonuniformity of pressure at prediffuser exit. Therefore, the values of $\bar{\lambda}_{1-2}$ may be in error by as much as 20%. However, no such uncertainty surrounds the value of $\bar{\lambda}_{1-4}$ presented in the previous section.

The prediffuser performance is presented in Fig. 7. It should be noted that the loss coefficient $\bar{\lambda}_{1-2}$ is very low and over a wide range of flow splits is largely independent of dump gap and prediffuser geometry. In view of this the variations in pressure recovery are almost entirely attributable to changes in the distortion of the outlet velocity profile. The reduction in distortion that occurs as the dump gap is reduced is illustrated in Fig. 8. At small and intermediate dump gaps, the head of the flame tube induces curvature in the prediffuser outlet flowfield, and therefore the pressure must be higher in the center of the annulus than at the walls. This rise in pressure, which is depicted in Fig 8, increases the diffusion

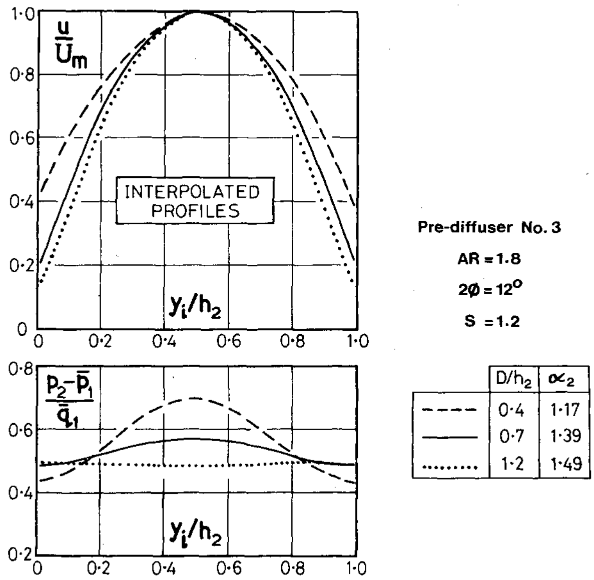

Fig. 8 Typical influence of dump gap on prediffuser outlet velocity and static pressure profiles.

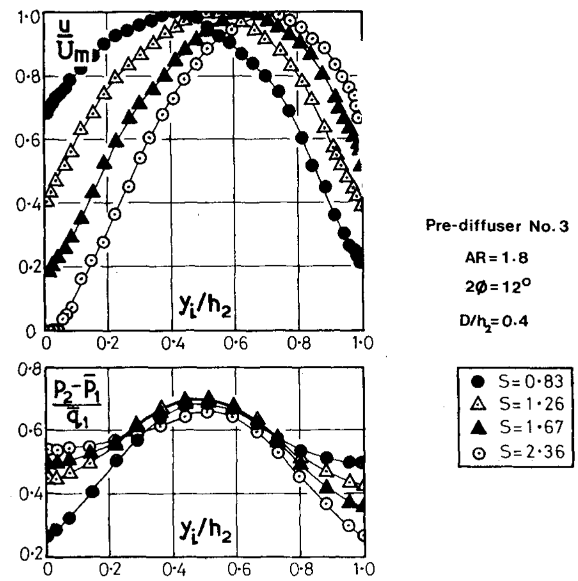

Fig. 9 Typical influence of flow split on prediffuser outlet velocity and static pressure profiles.

along the center steam tube, and continuity decrees that for a symmetrical profile ($S \simeq 1.2$) the velocity near the walls must increase.

The way in which the outlet velocity and pressure profiles change with flow split is shown in Fig 9. Operation at a flow split other than approximately 1.2 implies an asymmetric velocity and pressure profile at exit, and at a flow split of 2.26 which is near the design value, separation occurs on the inner wall as a large proportion of the flowfield is deflected towards the outer annulus. Although separation was confirmed by observations using wool tufts, no instability in the main flowfield was recorded. It was surprising to observe that the separation occurred at all positions around the circumference of the inner walls. This is in contrast to the usual behavior of separated flow in diffusers and may be associated with the stationary vortex in the dump region. The value of boundary-layer shape parameter, H, at which separation occured, was found to lie in the range 2.4 to 2.8. The conditions under which separation may be expected to occur are summarized in Fig. 10. It is seen that only by operating with small dump gaps can separation be prevented from occurring at the design flow

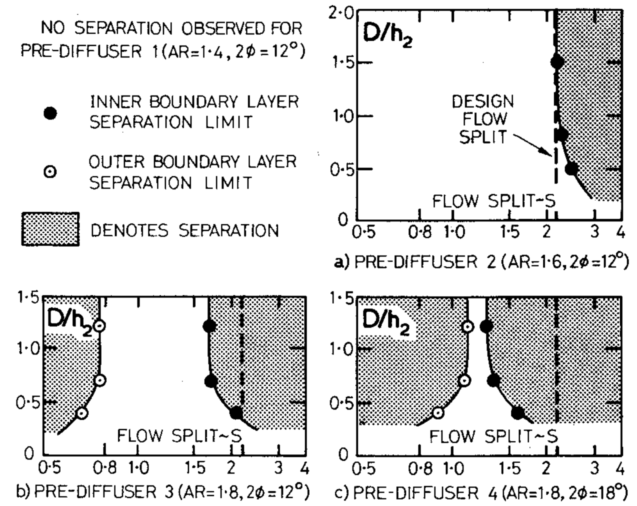

Fig. 10 Prediffuser outlet flow separation limits.

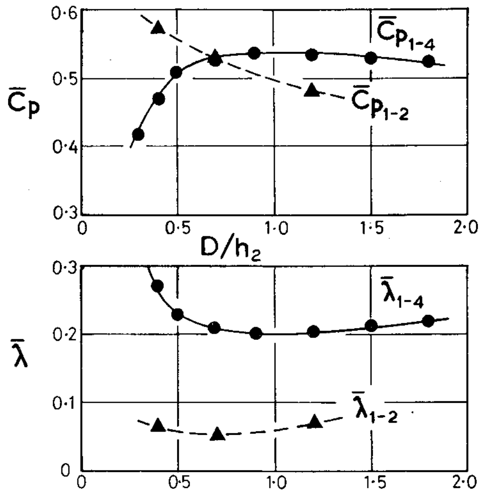

Fig. 11 Contribution of prediffuser to overall performance (prediffuser 3, $S = 1.6$).

split on the inner wall of prediffusers Nos. 2, 3, and 4. The range of flow splits over which nonseparated flow was obtained decreased with increase of area ratio at a fixed wall angle (diffuser Nos. 1, 2, and 3) and with increase of wall angle at a fixed nondimensional length (diffuser No. 4). In diffuser No. 4, even a symmetrical outlet velocity profile ($S \simeq 1.2$) was close to separation at large dump gaps.

Summarizing, the results show that efficient diffusion can be obtained over a wide range of flow splits and decreasing the dump gap improves the pressure recovery and reduces the likelihood of separation occuring. However, separation did occur when operating at the design flow split ($S^* = 2.15$). At a flow split of about 1.2, which corresponds to minimum overall loss, $\bar\lambda_{1-4}$, a symmetrical velocity profile is obtained at prediffuser outlet. The way in which the prediffuser contributes to the overall performance of the system is shown in Fig. 11. Whereas at large dump gaps there is a small increase of pressure downstream of the prediffuser, at small dump gaps, the losses in the down-stream section are high enough to cause a decrease in pressure recovery. Therefore, whereas optimum prediffuser performance is obtained at small dump gaps, the overall performance at this configuration is impaired. The losses downstream of the prediffuser at small dump gaps represent, typically, 80% of the total loss in the system, falling to about 65% at large dump gaps. Therefore, a large proportion of the losses occur in the region downstream of the prediffuser.

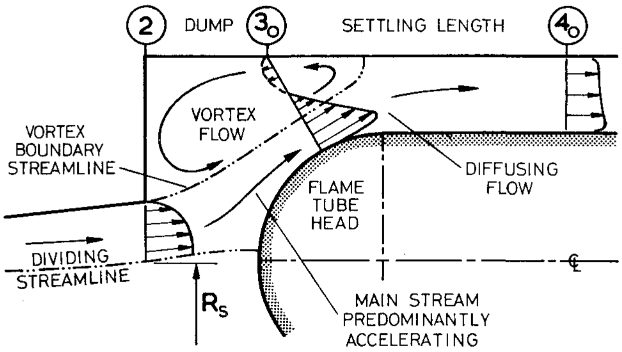

Fig. 12 Flow pattern downstream of prediffuser.

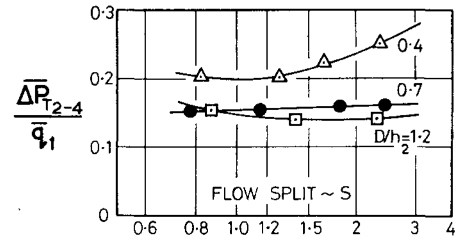

Fig. 13 Typical variation of loss downstream of prediffuser (No. 3; $AR = 1.8$, $2\phi = 12°$.

Dump and Settling Length

A typical flow pattern in the region downstream of the prediffuser is illustrated in Fig. 12. The flow is discharged into the dump region where it divides into two streams that pass over the head of the flame tube. The acceleration of the fluid into a field which is bounded by a stationary vortex produces a highly sheared velocity profile and a decrease in local static pressure on the head of the flame tube. Diffusion, assisted by centrifugal forces, then takes place as the fluid is deflected back to the axial direction. Owing to radial momentum transfer, a near-uniform velocity profile is obtained at the end of the settling length (Station 4). Figure 13 shows that the loss in the settling length and dump region is unaffected by changes in flow split.

To obtain a better understanding of the flow in this region, it is necessary to apportion the loss to the inner and outer annulus flowfields, namely

$$\bar\lambda_{2-4o} = (\Delta\bar p_{t2-4}/\alpha_2 \tfrac{1}{2}\rho U_2^2)_o = (\Delta\bar P_{t2-4}/\bar q_2)_o$$

and

$$\bar\lambda_{2-4i} = (\Delta\bar P_{t2-4}/\bar q_2)_i$$

To calculate the mass-weighted mean dynamic and total pressures in each flowfield at inlet to the dump region, the prediffuser outlet velocity profile was integrated to find the radius R_s at which

$$\int_{R_s}^{R_o} 2\pi\rho uRdR \Big/ \int_{R_i}^{R_s} 2\pi\rho uRdR = (m_o/m_i)_4 = S \qquad (7)$$

then

$$\bar P_{to} = \frac{2\pi}{U_o A_o}\int_{R_s}^{R_o}\left(p+\frac{\rho u^2}{2}\right)uRdR$$

The variation of the loss coefficient in each field with flow split is shown in Fig. 14. The decrease of loss coefficient with increase of flow is mainly attributed to a decrease in the amount of diffusion being attempted. In fact, at low flow splits, an overall acceleration is achieved in the inner flowfield. Therefore, at low flow splits, a high loss in the

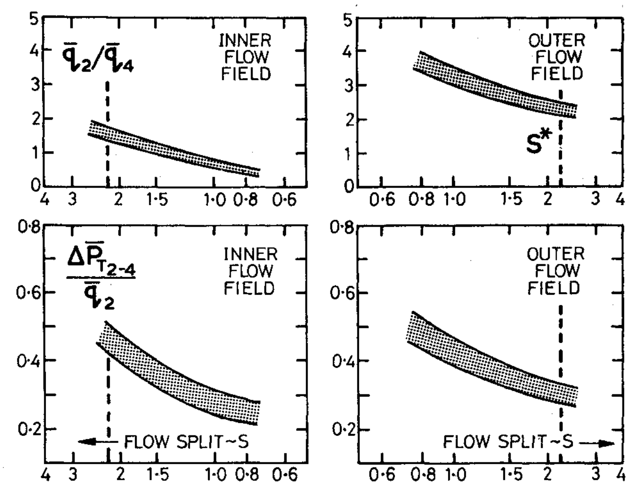

Fig. 14 Dynamic pressures and losses downstream of prediffuser.

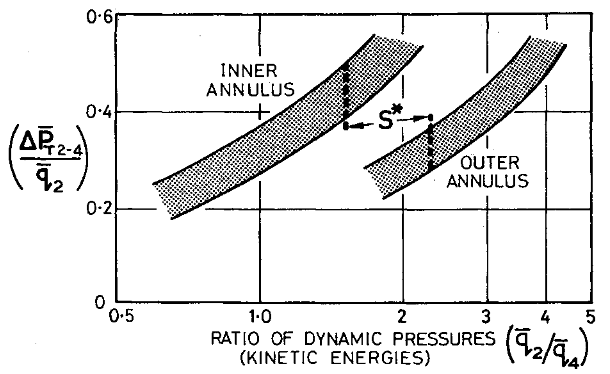

Fig. 15 Correlation of pressure loss with amount of diffusion.

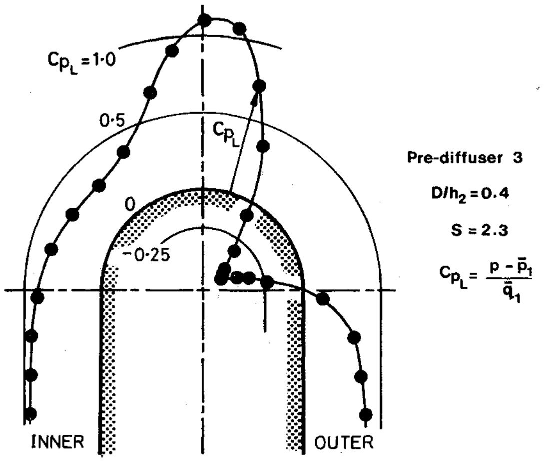

Fig. 16 Static pressure distribution on flame tube head.

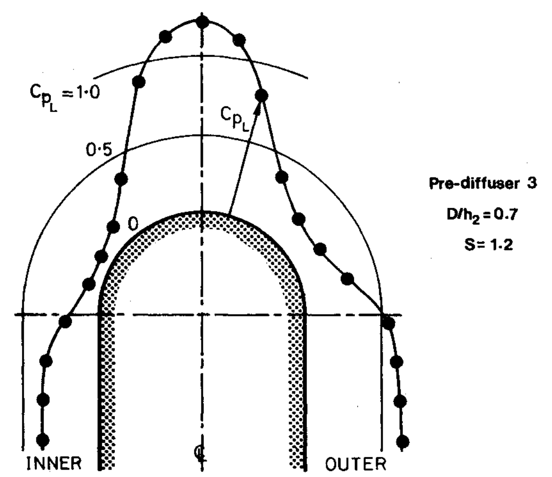

Fig. 17 Static pressure distribution on flame tube head.

outer annulus is offset by a low loss in the inner annulus and vice versa. Consequently, the *mass-weighted* total pressure loss is almost unaffected by changes in flow split (see Fig. 13). Owing to the complex nature of the flow in this region and the practical limitations imposed on the instrumentation, it is only possible to discuss in qualitative terms the various factors responsible for the high losses. Nonetheless, sufficient measurements were taken to establish that nearly all of the loss in the region between the outlet of the prediffuser and the settling length occurs downstream of the minimum pressure point on the head of the flame tube. Moreover, most of this loss is associated with the region between the center of the annulus and the wall of the flame tube.

The reason for the high loss is found in the unstable nature of a turbulent flow in which uR¶ decreases with R, as described by Prandtl[13] in an extension of his mixing-length theory. For flows over curved surfaces, the centrifugal "force" is largely balanced by a normal pressure gradient. Particles moving outward across the mean direction of flow into regions of lower mean velocity should retain some memory of their previous higher mean velocity; their individual centrifugal forces will be greater than the new mean normal pressure gradient, resulting in a net destabilizing force which manifests itself in a large increase in Reynolds stresses. Taking the center of curvature of the flow as Station 3 to be the center of the semicircle forming the head of the flame tube,analysis of the velocity profile at Station 3 indicated that apart from the wall boundary layer, the bulk of the profile is unstable. The flow will also retain some imprint of the turbulence structure at prediffuser outlet and, therefore, the turbulent mixing at Station 3 could be enhanced if the flow had a high initial turbulence level.

¶u and R measured relative to the center of curvature.

As the flow enters the dump region and is deflected towards the outer walls of the combustion chamber, the flow is stable because uR increases with R. It is estimated that at least half the loss in this region can be associated with the energy required to sustain the stationary vortices. Finally, it is important to note that the flow in the outer annulus is in the direction of an increasing radius and, therefore, a considerable amount of diffusion can occur circumferentially, i.e., normal to the plane of the velocity profile; whereas in the inner flowfield, diffusion can only occur in the plane of the velocity profile. Viets[14] in a theoretical investigation, has shown that the mixing rate is improved by diffusing normal to the plane of the velocity profile.

The losses in the two flowfields, at moderate and large dump gaps, are compared in Fig. 15. The difference in the reduction of kinetic energy at the design flow split is due to the asymmetry of the prediffuser outlet velocity profile. At the design flow split, separation on the inner wall at the exit of the prediffuser causes a high level of mixing in that portion of the flow which passes to the inner annulus surrounding the flame tube. This increased level of mixing is thought to be responsible for the higher losses in the inner flowfield. At small dump gaps, the curvature and local acceleration of the flow is even greater, and the losses are higher than those shown in Fig. 15.

To summarize, it is thought that the principal factors determining the losses in the dump and settling length regions are: (1) the amount of diffusion being attempted; and (2) the radius of curvature of the flow which is determined by the size and shape of the head of the flame tube and the dump gap.

Pressure Distribution on the Head of the Flame Tube

Apart from the attainment of minimum loss, it is also important to obtain a symmetrical pressure distribution around the head of the flame tube. The pressure distribution at a small dump gap and near design flow split, is shown in Fig. 16. It is seen that the flow accelerates from the stagnation point to a position about two-thirds the distance around the head before diffusing into the surrounding annuli. In the minimum pressure region, the normal pressure gradient is consistent with flow over a convex surface. Although the pressures in the annuli surrounding the flame tube are about the same, the pressure distribution around the head is asymmetric and areas exist in which the external static pressure can be lower than the pressure within the flame tube. Any holes placed in these areas would allow a reversal in the direction of flow to occur causing flame tube overheating. The asymmetric pressure distribution is caused by the radial distortion of the prediffuser outlet velocity profile (see Fig. 8).

In view of this, it would be expected that a near-symmetrical pressure distibution should be obtained with a symmetrical prediffuser outlet velocity profile ($S \simeq 1.2$). Figure 17 shows that this is indeed the case. Unfortunately, the areas of the annuli surrounding the flame tube are based on the need to achieve equal velocity (pressure) at the design flow split of 2.15, whereas at the optimum flow split of 1.2, the pressures are unequal. Therefore, by adjusting the areas surrounding the flame tube to suit a flow split of 1.2, i.e. $(A_o/A_i)_4 = 1.2$, a stable, low loss system with a symmetrical pressure distribution around the flame tube can be achieved. Subsequent tests by Goom[15] have confirmed this finding.

Canted Prediffuser

If the dividing streamline in the dump region is assumed to lie along the same radius as the center line of the pre-diffuser and flame tube, then the optimum flow split is given by Eq. (7) with $R_s = [(R_i + R_o)/2]_2$. Assuming a symmetrical velocity profile at prediffuser outlet and $(R_i/R_o)_2 = 0.772$, Eq. (7) gives $S = 1.1$, which is very close to the experimental value of about 1.2. If a design flow split other than about 1.2 is required, and a symmetrical profile is to be retained, it is necessary to alter the relative positions of the prediffuser and flame tube by canting the prediffuser or adjusting mean diameter of the flame tube.

Conclusions

Low speed tests have been carried out to investigate the performance of a combustor-dump diffuser system. These tests have shown that:

1) The system, which offers good flow stability over a wide range of operating conditions, has an overall performance comparable to that achieved by a "faired" diffuser.

2) The location of the flame tube has a marked effect on the performance and stability of flow in the prediffuser.

3) Nearly all of the static pressure rise occurs in the prediffuser, but most of the total pressure loss occurs in the dump and settling length region.

4) The principal determinants of the total pressure loss are: the amount of diffusion being attempted downstream of the prediffuser; and the radius of curvature of the flow which is determined by the size and shape of the flame tube and the dump gap.

5) The geometry of the prediffuser, flame tube, and surrounding annuli need to be carefully matched if a satisfactory pressure distribution on the head of the flame tube is to be obtained at flow splits other than optimum.

It is emphasized that the above conclusions are based on tests conducted with near fully-developed flow at inlet. The way in which conditions at exit from an axial flow compressor affect the performance of combustor-dump diffusers is the subject of a current investigation.

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