The Influence of Blade Wakes on the Performance of Combustor Shortened Prediffusers

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Results of an experimental investigation of the influence of blade wakes on the performance of several combustor prediffusers are presented. Two curved and two straight wall diffusers of different length, area ratio, and turning angle were tested with a single-stage compressor sited at a number of positions relative to diffuser inlet. Only a small increase in loss was incurred when the wakes from the outlet guide vanes were allowed to decay within the prediffuser. Distance along the mean streamline was found to be the most significant parameter influencing the decay of the wakes. In outwardly curved diffusers, a distance of about four blade chord lengths was required for these effects to be minimized.

Nomenclature

A = area of cross section
B = blocked area fraction
C = radial clearance between blade and hub or casing
c = blade chord length
C_p = pressure recovery coefficient
D = diameter
D_h = hydraulic diameter
h = radial height of annulus
L = mean diffuser axial length
m = mass flow rate
p = static pressure
P_t = total pressure
R = radius
ΔR = radial height of annulus at inlet
R_e = Reynolds number based on hydraulic diameter
s = blade spacing
t = maximum blade thickness
u = local axial velocity
u_s = average axial velocity in spanwise direction
u_c = wake centerline velocity
U = maximum velocity in cross section
U_s = maximum value of u_s
U = wake edge velocity
x = axial distance from diffuser inlet
y = distance normal to surface
z = spanwise coordinate
α = velocity profile energy coefficient $= 1/A \int_0^A (u/U)^3 \mathrm{d}A$
β = angle between meridional streamline and axial direction
δ^* = displacement thickness
θ = momentum thickness
λ = loss coefficient
ρ = fluid density
τ = shear stress
$\emptyset$ = diffuser wall angle

Subscripts
1 = diffuser inlet
2 = diffuser outlet
i = inner wall
o = outer wall

Superscripts
$\sim$ = mass-weighted mean value
$-$ = mean 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 the necessary conditions prior to combustion and avoid large losses in total pressure. Ideally the flow should be decelerated within a relatively short distance, since engine weight is proportional to axial length. Furthermore, current energy efficient engine designs[1] indicate a large increase in diameter from compressor exit to turbine inlet; consequently the diffuser is required not only to diffuse the flow but also to turn it radially outward. Such constraints add to the difficulty of designing combustor diffusers to give a flow which is both stable over a wide range of operating conditions and insensitive to manufacturing tolerances. Apart from minimizing the loss in total pressure, an additional requirement of the diffuser system is that it should ensure that a near uniform flow is delivered to the annuli surrounding the flame tube. Failure to achieve this criterion can have detrimental effects on combustor exit temperature distribution, emission levels, and liner integrity

A large number of parameters influence the performance of a dump diffuser-combustor system.[2] The main geometric prediffuser variables are the area ratio, the length to inlet annulus height ratio ($L/\Delta R_I$) and, for curved wall systems, the degree of turning (β) However the dump gap, shape, and porosity of the combustor head, as well as the mass flow split between the two annuli, essentially determine the overall loss of the system. In addition, the flow is influenced by the shape of the inlet velocity profile, turbulence structure, circumferential nonuniformities generated by the compressor, and, in particular, by the wakes from the outlet guide vanes.

Klein et al.[3] presented experimental results of the effects of flow distortions generated by an annular tandem cascade on the performance of a short dump diffuser-combustor inlet.

The distance between the cascade and the diffuser inlet plane could be varied and the overall loss was found to increase substantially when this distance became small In most of the tests, minimum loss was obtained when the cascade was at a distance upstream of about two blade-chord lengths

Stevens et al have also investigated the way in which compressor exit conditions influence the performance of optimum combustor dump diffusers The experiments showed that the overall loss to the flame tube annuli measured when there was a considerable decay of the blade wakes prior to diffuser inlet, was almost the same as that achieved with fully developed inflow Of particular importance was the finding that when the guide vanes were close to the diffuser entrance a reduction in the distortion of the prediffuser outlet velocity profile was obtained; furthermore, only a small increase in overall loss was incurred Although Klein et al suggest that the conflicting results may be correlated using a radial, rather than a total, blockage factor at prediffuser inlet, the present authors consider that the explanation lies in the development of the flow in the prediffuser and, in particular on the behavior of the blade wakes

In the experiments of Klein et al the ratio of the length of the prediffuser L to the chord of the outlet guide vanes c was $L/c = 0$ 59, which is a considerably smaller value than that used in most aircraft applications At representative dump gaps, the values of total blockage factor at prediffuser outlet suggest that the blade wakes did not decay significantly Furthermore, doubts must be expressed concerning the use of annular tandem cascades to simulate compressor exit flows, since they do not simulate the unsteadiness associated with the upstream rotors The wakes from the preswirl vanes may also affect the behavior of the flow

Experiments have, therefore, been conducted using a single stage axial flow compressor, which can be sited at a number of positions relative to the inlet of four combustor prediffusers, which show the influence of variables such as those mentioned above

Experimental Apparatus

The test facility, manufactured in Plexiglass, is shown in Fig 1 The entry length, diffuser and settling length 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 integral flare and nose bullet and down an annular approach length 40 hydraulic diameters long, ensuring fully developed flow at inlet to the single stage compressor Details of the compressor stage are given in Table 1 The rotor was driven by a shaft coupled to an electric motor mounted in the plenum chamber.

Diffusers 1 and 2 both had straight walls their outer wall angles being 10 and 6 62 deg respectively, and their inner walls being of constant diameter Diffusers 3 and 4 had outward curving inner and outer walls their center lines formed a circular arc of radius $R = 10\ 65\Delta R_1$ The diffuser geometries are detailed in Fig 2

An almost constant dynamic pressure was maintained in the center of the annulus at a position 0 076 m upstream of rotor inlet This corresponded to a velocity of 57 m/s and a Reynolds number based on the inlet hydraulic diameter of 2×10^5

Static pressure measurements were made at positions upstream of rotor inlet and along the length of the diffuser At each position three tappings were made, spaced equally around the surfaces of the inner and outer walls In addition, measurements were made of the radial pressure and velocity distribution at a number of stations along the length of the diffusers, using a miniature five-hole combination probe This probe consisted of a cluster of five tubes of 0 25 m bore; its stem being 1 73 mm diam To facilitate a series of circumferential traverses across a distance of two blade spacings, the outlet guide vanes were mounted in a carrier which could be rotated and accurately positioned using a micrometer mechanism

The positioning of the probe and the digitizing and recording of the analogue signals from the pressure transducers was controlled by a DEC LSI 11 microcomputer

Performance Evaluation

The definitions of total pressure loss and static pressure recovery coefficient are

$$\tilde{\lambda}_{1\ 2} = (\tilde{P}_{t_1} - \tilde{P}_{t_2})/\alpha_1 \tfrac{1}{2}\rho\bar{u}_1^2$$

and

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

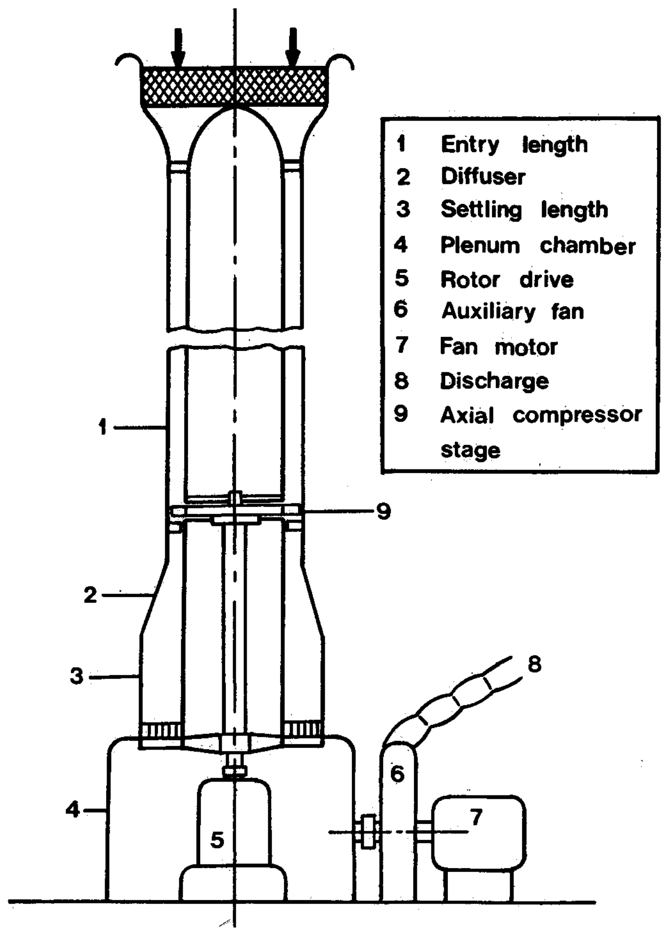

Fig 1 Test facility

Table 1 Stage geometry

Blade chord length c mm	19 56
Blade height h_1 mm	25 40
Hub tip ratio, $(R_i/R_i)_1$	0 83
Rotor blade tip clearance mm	0 20
Outlet guide vane tip clearance mm	0.10
Space chord ratio s/c	0.50
Thickness chord ratio t/c	0 10
Reaction	50%
Blade inlet angle deg	30
Blade outlet angle deg	−6
Stage axial clearance, mm	20 30

where $\tilde{p}_t = \tilde{p} + \alpha \frac{1}{2} \rho \tilde{u}^2$ Writing the energy equation across the diffuser as

$$\tilde{p}_1 + \alpha_1 \tfrac{1}{2} \rho \tilde{u}_1^2 = \tilde{p}_2 + \alpha_2 \tfrac{1}{2} \rho \tilde{u}_2^2 + (\tilde{p}_{t_1} - \tilde{p}_{t_2}) \qquad (1)$$

and from continuity $A_1 \tilde{u}_1 = A_2 \tilde{u}_2$ Then Eq (1) may be written as

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

The mass weighted mean values of dynamic and static pressure were based on data from circumferential traverses extending over two blade spacings at 17 radial locations The mass flow at various stations was compared with the value obtained from a calibrated measurement taken three annulus heights upstream of the compressor rotor Whereas the mass flow at diffuser exit was only 2% higher than the reference value, at inlet the discrepancy rose, in some cases to +6% The influence of turbulence on pressure measurements and the fact that measurements taken over two blade spacings were averaged over the entire annulus are considered to be the most likely sources of error To satisfy continuity and retain the same velocity ratios u/U corrections were applied equally to both total and static pressure at each point in the flow

Unfortunately, the calculation of $\tilde{\lambda}_{1-2}$ is very sensitive to changes in the values of α_2 and $\tilde{C}_{p_{1-2}}$; because of this the value of $\tilde{\lambda}_{1-2}$ may be in error by as much as ±15% The values of $\tilde{C}_{p_{1-2}}$ are considered to be within ±5%

Results and Discussion

Inlet Conditions

Figure 3 shows the axial velocity contours in a section comprising two blade spacings when the trailing edge of the guide vanes is at the inlet of Diffuser 3 Measurements confirmed that, within experimental error, the flow was essentially swirl free The vortex located at the tip of the blades between the suction surface and the hub can be clearly seen; also, a much smaller secondary flow region is present near the blade root adjacent to the outer casing These differences in the regions of secondary flow may be due to the pressure field associated with the curvature of the flow at diffuser inlet Since the flow is deflected radially outward, the pressure on the inner wall must be greater than that on the outer casing Thus the adverse pressure gradient near the tip, or the suction surface of the blades, is increased leading to an earlier separation of the flow The axial velocity contours measured at diffuser inlet with the guide vanes one blade chord length upstream, are depicted in Fig 4 It will be seen that the vortices have increased considerably, and near the hub have moved away from the wall Also, high velocity regions have moved towards the outer casing Figure 5 compares the radial variation of the axial velocity u_s obtained

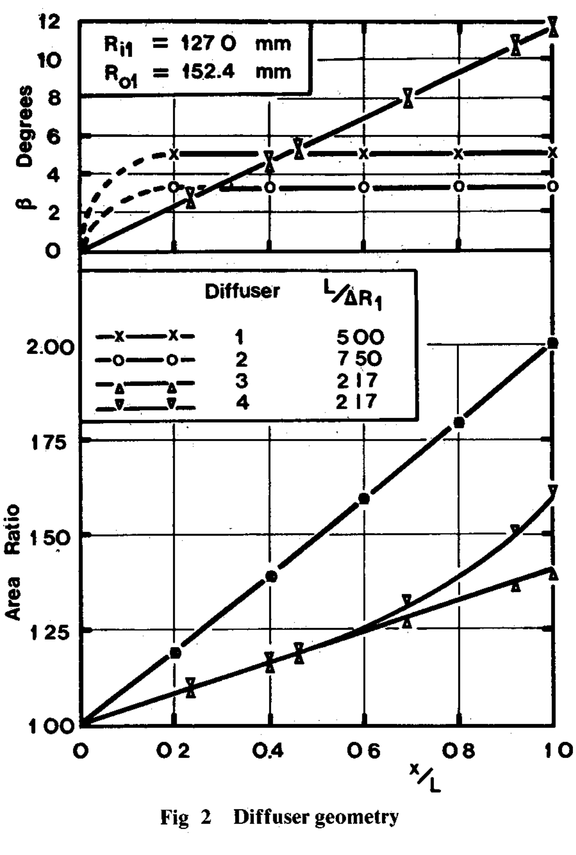

Fig 2 Diffuser geometry

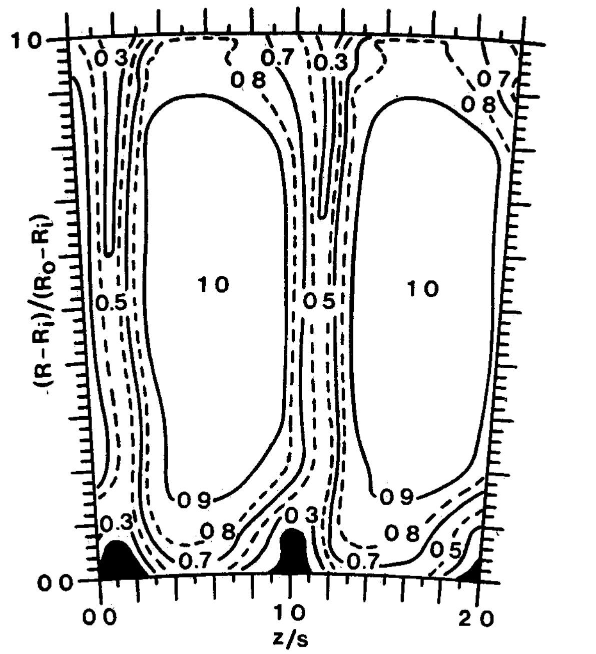

Fig 3 Axial velocity ratio contours at inlet of Diffuser 3, $x/c = 0\ 0$

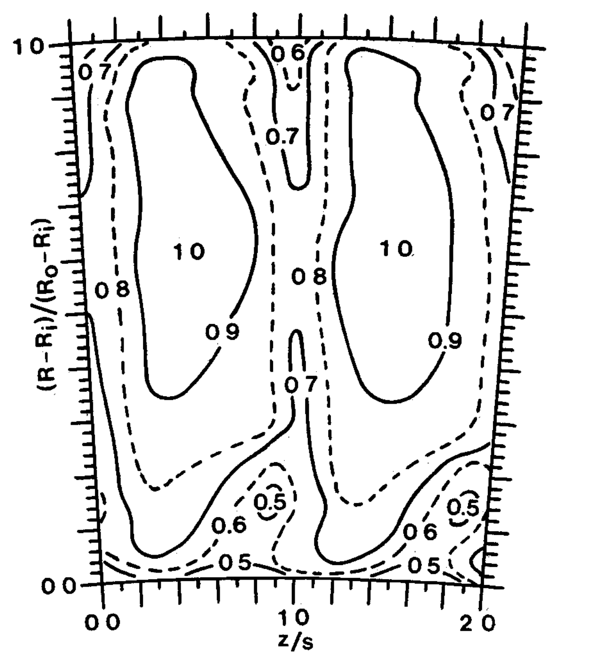

Fig 4 Axial velocity ratio contours at inlet of Diffuser 3, $x/c = -1\ 0$

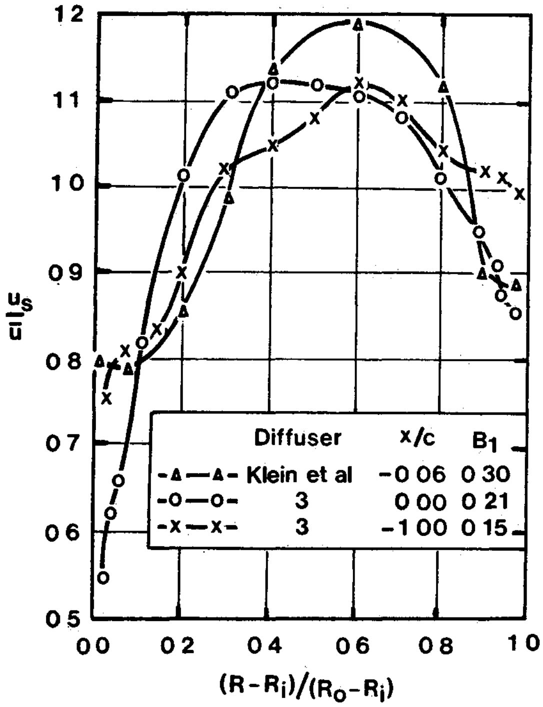

Fig 5 Axial velocity profiles at diffuser inlet

by averaging circumferentially over two blade spacings, with the data due to Klein et al The large vortex near the hub can be seen to have resulted in a lowering of the axial momentum in that region When the guide vanes are at $x/c = 1\ 0$, the energy transfer brought about by the mixing associated with the vortices at root and tip increases the average momentum in the inner wall regions; nonetheless, the blockage fraction is still greater than that associated with fully developed flow ($B_l = 0\ 105$). Similar trends have been observed by Klein et al , although owing to the different blade geometry, the values of blockage were higher

Straight Wall Diffusers

The development of the axial velocity contours at various stations along the length of Diffuser 1 is described in Ref 5 and typical results are shown in Fig 6 Owing to the vigorous mixing associated with the wakes they decay rapidly and by $x/L = 0\ 39$ most of the wakes have disappeared and only slight circumferential nonuniformities are present

Using a similar approach to that proposed by Stratford [6] the flow may be described using conventional boundary layer techniques. The main difference is the need to take account of circumferential variations in axial velocity The momentum thickness θ may then be defined by

$$\rho U^2 \theta = \int_{A_b} (U-u)\, dm/2\pi R \qquad (3)$$

where U is the maximum velocity in an area A_b extending over one blade spacing Similarly, the displacement thickness δ may be defined as

$$\delta^* = (A_b - A_b')/2\pi R \qquad (4)$$

where A_b' is the area which the flow would occupy were it all at the local maximum velocity U

Applying the momentum equation to the flow in the shaded area as in Fig 7 and assuming that there is no potential core

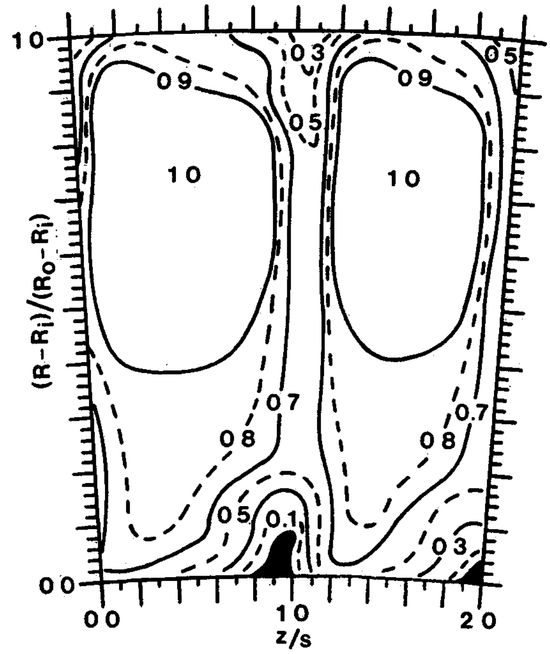

Fig 6a Axial velocity contours, Diffuser 1, $x/c = -0\ 125$, $x/L = 0\ 0$

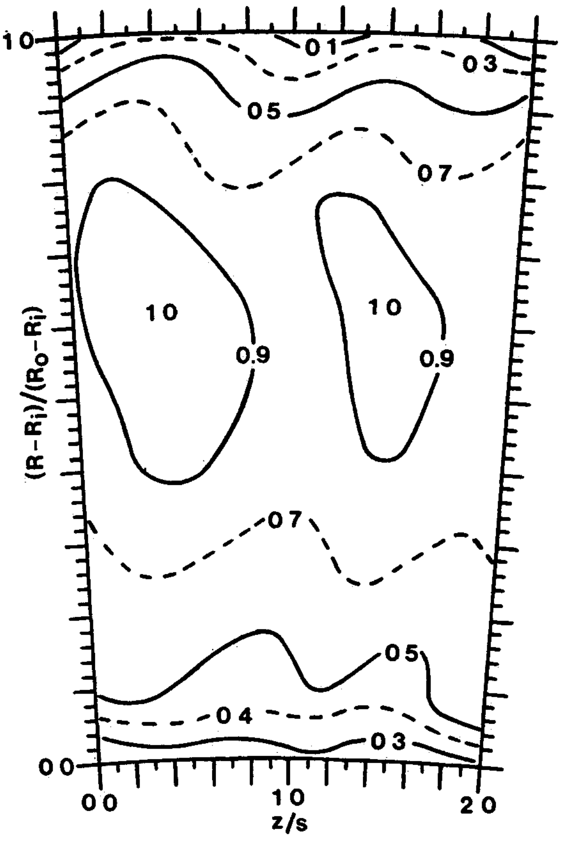

Fig 6b Axial velocity contours Diffuser 1 $x/c = -0\ 125$, $x/L = 0.39$

then

$$\int_{A_{2b}} u_2 \mathrm{d}m_2 - \int_{A_{1b}} u_l \mathrm{d}m_l - (m_{2b} - m_{1b}) \frac{U_l + U_2}{2} = -(p_2 - p_l)\left(\frac{A_{1b} + A_{2b}}{2}\right) - \tau_\omega \pi (R_l + R_2)(x_2 - x_l) \qquad (5)$$

The last term on the left side of Eq (5) takes account of any mass transfer between the two layers Substituting Eqs (3) and (4) and noting that

$$A_b' = \int_{A_b} (u/U)\,\mathrm{d}A$$

yields the momentum difference equation

$$\rho R_2 U_2^2 \theta_2 = \frac{\tau_w}{2}(R_1+R_2)(x_2-x_1) + \frac{\Delta p}{2}(R_1\delta_1^* + R_2\delta_2^*)$$

$$+ \frac{1}{2\pi}\left[\left(m_{2_b}U_2 - m_{1_b}U_1\right) + \frac{\Delta p}{2}\left(\frac{m_{1_b}}{\rho U_1} + \frac{m_{2_b}}{\rho U_2}\right)\right]$$

$$- \frac{1}{2\pi}\left[(m_{2_b} - m_{1_b})\left(\frac{U_1+U_2}{2}\right)\right] + \rho R_1 U_1^2 \theta_1 \qquad (6)$$

where $\Delta p = p_2 - p_1$

A typical comparison of the measured values of $\rho R_2 U_2^2 \theta_2$ with the experimentally determined right side of Eq (6) is given in Fig 8 The wall shear stress was approximated using a Clauser plot of the circumferentially averaged velocity profiles (u_s/U_s) Any marked divergence between the two sides of the equation is generally attributed to the effects of asymmetry That the comparison is, within experimental error, very good is seen as further evidence that the data is free from significant asymmetry

Typical wake velocity profiles at three radial positions along the length of Diffuser 1, with the outlet guide vanes close to inlet, are shown in Fig 9 The results are presented as the ratio of the local velocity to the maximum value U_z recorded along a circumferential traverse Near the hub the wake occupies a large proportion of the blade spacing and is asymmetric, this asymmetry being a reflection of the different pressure gradients on the pressure and suction surfaces of the blade. At midannulus the size and momentum defect of the wakes is much smaller The behavior of wakes in pressure gradients has been studied by Hill et al ,[7] who showed that if the adverse pressure gradient is large enough, the wake may grow, rather than decay Whether the wakes grow or decay depends on the relative magnitudes of the pressure and shear forces, although when the wakes are thin and the shear stresses are high, a considerably larger pressure gradient is required for the wake depth to increase The decay of the blade wakes in Diffusers 1 and 2, with the guide vanes near to inlet, is depicted in Fig 10 It is interesting to note that despite the high initial adverse pressure gradient, the high level of mixing is responsible for a rapid decay of the wakes during the initial stages of diffusion However, the rate of decay, particularly on the outer wall, is less than that of an isolated wake in zero pressure gradient, and so the pressure

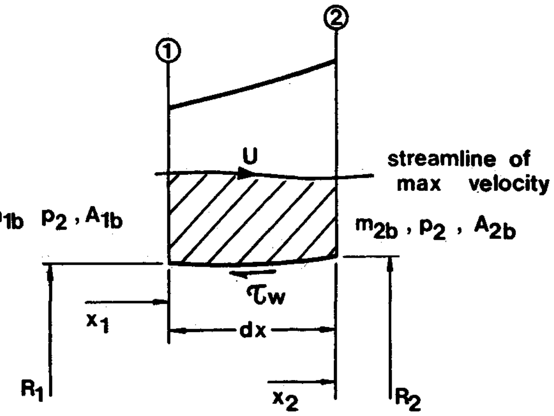

Fig. 7 Inner wall boundary layer control volume

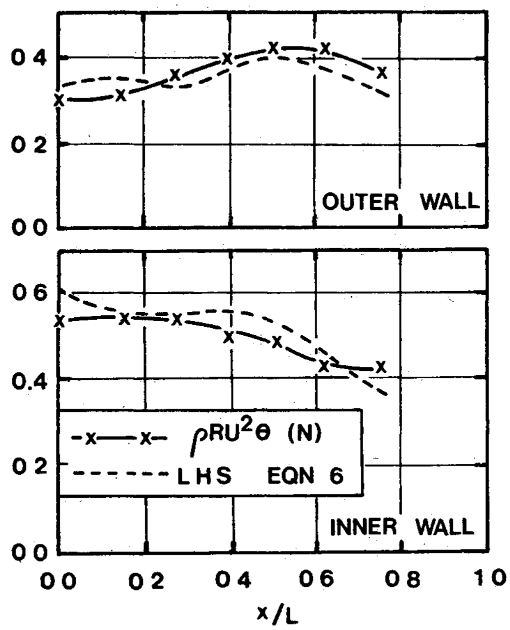

Fig 8 Balance of momentum, Diffuser 1

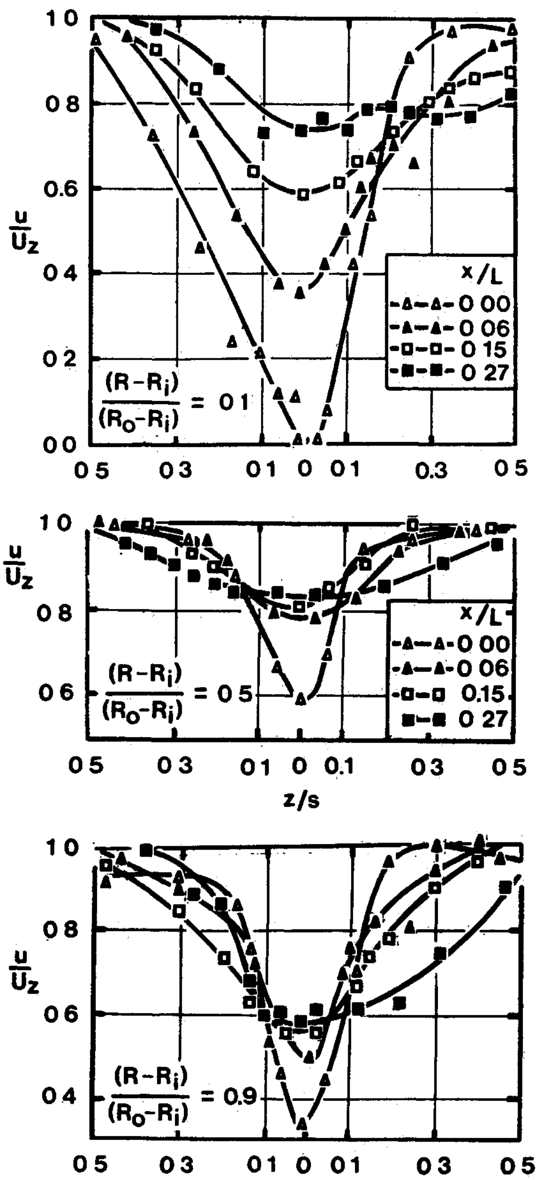

Fig 9 Wake velocity profiles Diffuser 1

field clearly influences the flow mechanism Unfortunately, owing to the influence of the wakes, wall pressure measurements and estimates of gradients are unreliable close to the inlet of the diffusers Previous tests with fully developed inflow,[8] however, indicate that due to the outward turning of the flow the pressures on the inner wall are higher than those on the outer casing Consequently, the initial pressure gradient along the outer wall is larger, as indicated in

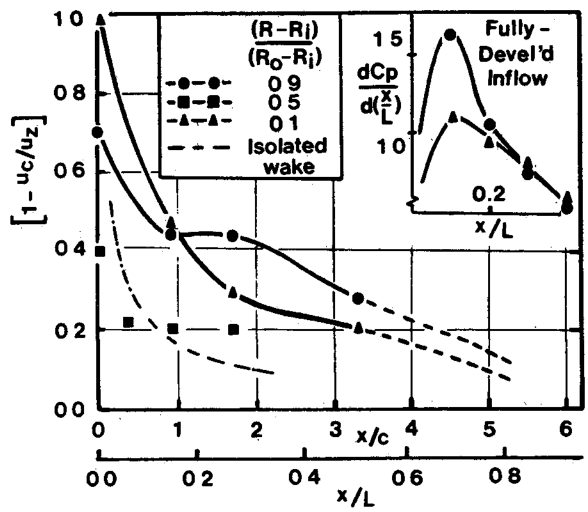

Fig 10 Variation of relative wake depth, Diffuser 1, $x/c = -0\ 125$

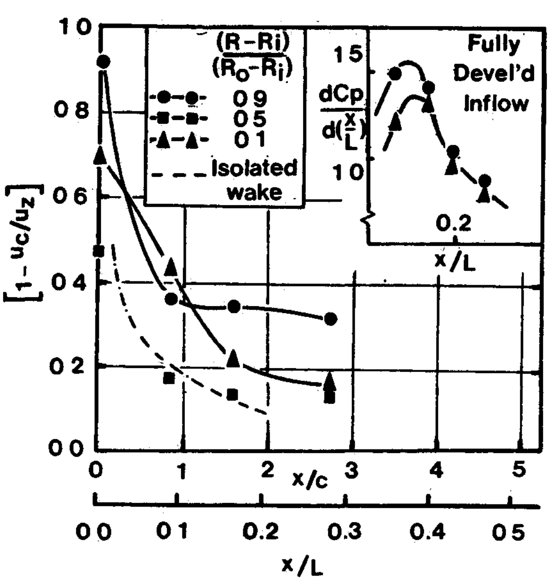

Fig 11 Variation of relative wake depth, Diffuser 2, $x/c = -0\ 125$

Fig 11, and the higher values of wake depth along the outer wall are, therefore, attributed to the higher initial pressure gradient Values of wake depth have not been quoted for the latter stages of diffusion because, as illustrated in the contour plots, only small circumferential nonuniformities are present and no clear wake pattern can be distinguished

The radial variation of the average velocity in the spanwise direction u_s at the outlet of Diffuser 1 is shown in Fig 12 together with the results obtained with fully developed inflow With the guide vanes near to inlet there is a significant im provement in the flow on the outer wall, but, owing to the lower momentum at inlet, there is a deterioration on the inner wall Nonetheless, no separation was observed and the stability of the outlet flow was improved, confirming the findings of an earlier investigation [4]

The overall performance of Diffusers 1 and 2 with the guide vanes at various positions is presented in Table 2, with data obtained for fully developed flow at inlet being included for comparison Because of the higher kinetic energy at inlet (α_I) when the guide vanes are at $x/c = 0\ 125$ the potential pressure recovery of both diffusers is increased significantly. This effect is enhanced in Diffuser 1 by the reduction in the distortion of exit velocity profile (α_2) whereas in Diffuser 2 any improvement in $\tilde{C}_{pI-2}$ is offset by increased losses If results for $x/c = -0\ 125$ are compared with those for fully developed inflow, the losses in both diffusers ($\tilde{\lambda}_{I-2}$) are seen to increase by about 0 04 It is important to recognize that not all of this increase is attributable to the diffuser, since in many engines the loss associated with the decay of the blade wakes occurs prior to diffuser inlet Tests reported in Ref 4 suggest that the loss coefficient associated with wake decay in a parallel passage is about 0 025, and so the increase in diffuser loss coefficient due to the presence of the wakes is quite small

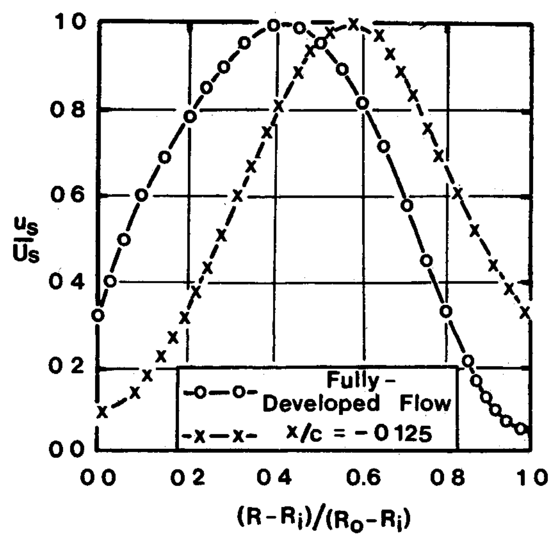

Fig 12 Outlet axial velocity profiles Diffuser 1

Table 2 Performance of Diffusers 1 and 2

Diffuser	OGV located at x/c	$-0\ 125$	$-3\ 125$	$-6\ 125$	Fully developed flow
Diffuser 1,	$(u/U)_I$	0 77	0 85	0 86	0 895
$L/\Delta R_I = 5$	α_I	1 14	1 05	1 04	1 045
	α_2	1 55	1 72	1 76	1 74
	$\tilde{C}_{pI-2}$	0 54	0 51	0 50	0 51
	$\tilde{\lambda}_{I-2}$	0 12	0 08	0 08	0 077
Diffuser 2,	$(u/U)_I$	0 78		0 86	0 895
$L/\Delta R_I = 7\ 5$	α_1	1 15		1 04	1 045
	α_2	1 39		1 45	1 41
	$\tilde{C}_{pI-2}$	0 59		0 58	0 60
	$\tilde{\lambda}_{I-2}$	0 11		0 07	0 06

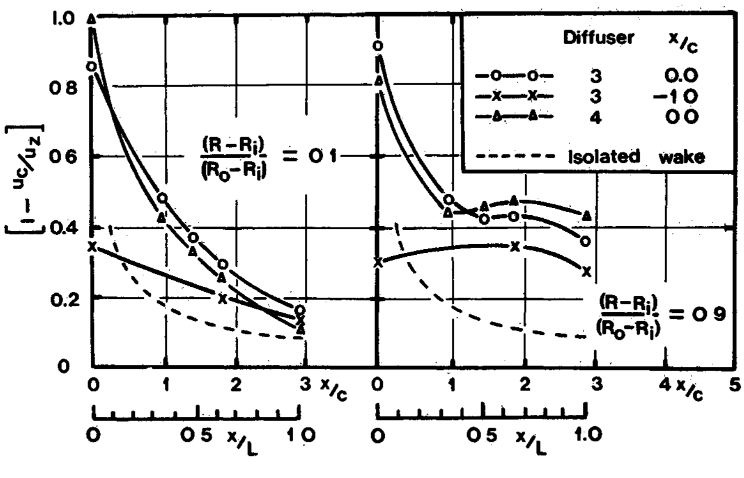

Fig 13 Variation in relative wake depth.

Table 3 Overall performance				
Diffuser	3	3	3	4
OGV located at x/c	0	$-1\ 0$	Fully developed flow	0
$(u/U)_1$	0 79	0 85	0.895	0 79
α_1	1 17	1 06	1 06	1 17
α_2	1 18	1 17	1 16	1 29
$\bar{C}p_{1-2}$	0 39	0 38	0 39	0 46
λ_{1-2}	0 08	0 055	0.035	0 11

It is interesting to note that when the guide vanes are sited further upstream the performance is similar to that obtained with fully developed inflow, confirming the finding of other investigations.[4,9]

Curved Wall Diffusers

Apart from velocity surveys at inlet and exit, traverses were also carried out at $X/L = 0\ 34$, 0 46, and 0 65 The decay of the relative wake depth in both diffusers is presented in Fig 13. Whereas in Diffusers 1 and 2 radial pressure gradients are only present at inlet, because curvature continues along the length of Diffusers 3 and 4, radial pressure gradients persist throughout. Consequently the wall pressure gradients will be dissimilar Figure 14 depicts the values obtained when operating with fully developed inflow. Whereas the pressure gradient on the inner wall falls continuously, along the outer wall, it is maintained at a relatively high value

That the decay of the wakes near both walls is strongly influenced by the pressure gradient is illustrated by Fig 12 Also shown are the results obtained when the guide vanes are sited one blade chord length upstream of the inlet. Although during the initial stages of diffusion the wakes decay, the extent to which the pressure gradient arrests this process can be estimated by comparing the depth of wake U_c/U_z at a distance one blade-chord length from inlet with the value at inlet when the guide vanes are at $x/c = -1\ 0$ In the latter case, the wakes are decaying in almost zero pressure gradient Near to the inner wall, with the guide vanes at inlet the value of u_c/U_z at $x/c = 1\ 0$ is about 0 55, compared with a value of 0 65 at inlet with the blades at $x/c = -1\ 0$ Near to the outer wall the comparable values are 0 55 and 0 70 That the retarding effect is greater near the outer wall is seen as further evidence of the influence of the pressure gradient, which is higher in this region. Although the wakes continue to decay along the inner wall of Diffuser 4, on the outer wall down stream of $x/L = 0\ 4$ they actually grow for a short distance Figure 13 indicates that throughout this region of increased diffusion the pressure gradient along the outer wall remains at a high value It is important to note that in all the diffusers

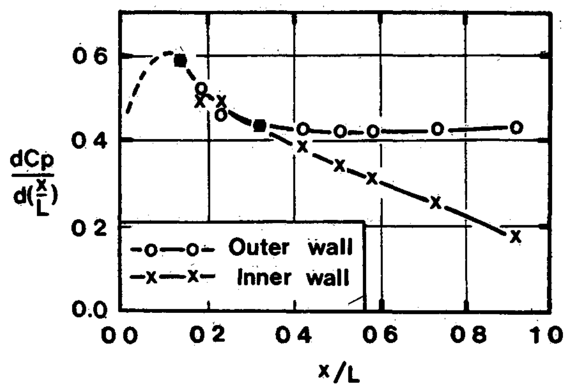

Fig 14 Static pressure gradient, Diffuser 3 fully developed inflow.

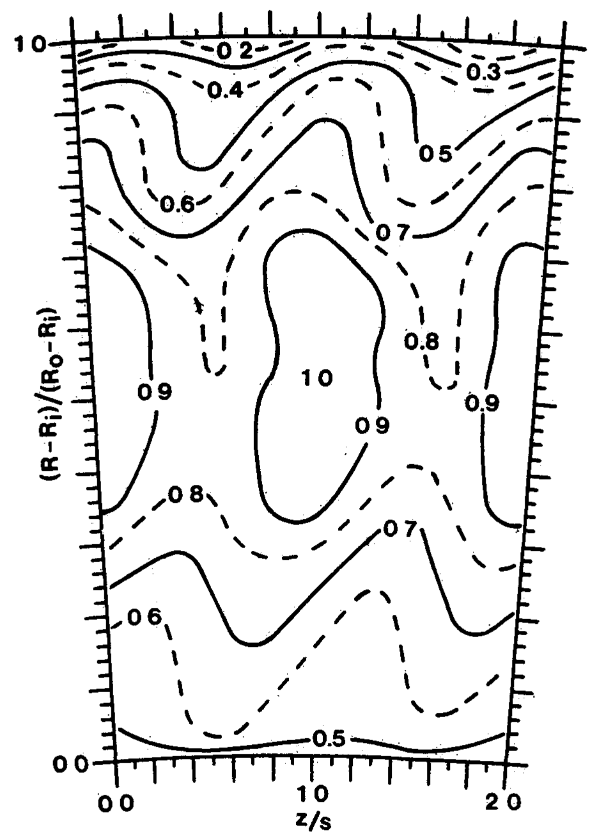

Fig 15 Axial velocity contours at exit of Diffuser 3, $x/c = 0\ 0$

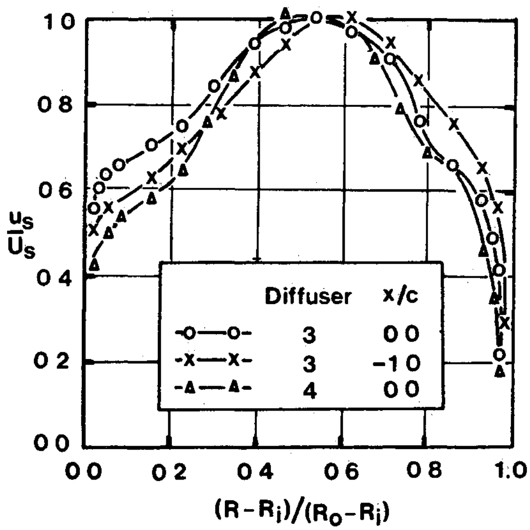

Fig 16 Axial velocity profiles at diffuser exit

tested a distance of about four blade chord lengths is required for the wakes near the outer wall to decay significantly It is for this reason that in the work of Klein et al minimum losses were achieved with the guide vanes sited at $x/c = -2\ 0$

The velocity contour plot at exit of Diffuser 3 with the guide vanes at inlet is illustrated in Fig 15 Although the blade wakes are more pronounced near the outer wall and low velocity regions are evident no separation could be detected The radial variation of the average velocity in the spanwise direction u_s is presented in Fig 16 Whereas the mixing associated with the decay of the blade wakes has increased the average velocities near the inner wall on the outer the presence of the wakes lowers the average velocity

The overall performance of the two diffusers is presented in Table 3 The performance of Diffuser 3 with the guide vanes at inlet and at $x/c = -1\ 0$ is within experimental error the same The potentially increased pressure recovery associated with the higher inlet kinetic energy (α_1) when the guide vanes are at $x/c = 0$ is counteracted by the losses due to the presence of the blade wakes. Increasing the area ratio to 1 6 (Diffuser 4) produces a 15% rise in pressure recovery despite the higher outlet kinetic energy (α_2)

Conclusions

Tests conducted to investigate the influence of blade wakes on the performance of combustor prediffusers have shown that:

1) Only a small increase in total pressure loss is incurred when the wakes from the outlet guide vanes are allowed to decay within the prediffuser

2) For outwardly curved prediffusers a distance of about 3 to 4 blade chord lengths is required for the wakes near the outer wall to decay significantly. Consequently if very short prediffusers are used it will be necessary to place the guide vanes upstream Increasing the outward curvature of the flow will accentuate these effects

3) When the outlet guide vanes are sited very close to the entrance of an optimum length diffuser there is an im provement in the stability of the outlet velocity profile compared with the results obtained with fully developed inflow

4) The radial pressure gradient associated with the cur vature of flow at diffuser inlet may, if the outlet guide vanes are very near lead to early separation from the blade sur faces Considerable care must be exercised in the aerodynamic integration of the diffuser inlet with the outlet guide vanes

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