ABSTRACT
Detailed flow measurements at the inlet to a centrifugal compressor vaneless diffuser are presented. The mean velocities and six component stress tensor are used to determine the turbulence production terms which lead to total pressure loss. Four regions in the flow are identified as potential sources of loss - the blade wake, the shear layers between passage wake and jet, the thickened hub boundary layer and the interaction region between the secondary flow within the blade wake and the passage vortex. The blade wakes generate most turbulence, with smaller contributions from the hub boundary layer and secondary flows, but no significant contribution is apparent from the passage wake shear layers.

NOMENCLATURE
L - Radial distance from impeller outlet
q - Turbulent kinetic energy
r,θ,z - Cylindrical co-ordinate system
R₀ - Impeller outlet radius
U_r, U_θ, U_z - Radial, Tangential and axial mean velocity components
U_r', U_θ', U_z' - Radial, Tangential and axial r.m.s fluctuating velocity components
y, Y_θ - Tangential coordinate in measurement plane
z - Axial coordinate in measurement plane
z₀ - Axial diffuser width in measurement plane

INTRODUCTION
A major objective of the turbomachinery designer is to minimise the aerodynamic losses which result both within the turbomachine blade passages and downstream as non uniformities in the flow are mixed out. Moore and Moore (1987) showed that, for a turbine cascade, a major contributor to loss production within the turbulent boundary layers and passage vortex is the conversion of mean kinetic energy to turbulent kinetic energy and hence to total pressure loss due to turbulent dissipation. Such a conversion process is only possible where shear is present in the mean flow eg within the boundary layers or within the secondary flow regions associated with a passage vortex. Most turbomachines discharge a complex 3-d flow exhibiting several flow features which will ultimately result in turbulent dissipation and loss. The designers problem is in identifying features which are the major contributors to the loss such that he can attempt to modify his design to reduce their effect.

The centrifugal compressor vaneless diffuser is a good example for the study of losses due to turbulent dissipation. The geometry is simple, but because the flow entering the diffuser is highly non uniform with strong secondary flows, turbulent dissipation leads to relatively high losses. The purpose of this paper is to study the flow mechanisms which lead to the losses within a low speed centrifugal compressor vaneless diffuser.

EXPERIMENTAL PROCEDURE
A schematic of the low speed centrifugal compressor rig used in the study is shown in Figure 1. The impeller was a De Havilland Ghost impeller, the geometry of which is given by Johnson and Moore (1980). In the current study, the original radial outlet section was replaced to provide a 30° backswept outlet angle as detailed in Farge and Johnson (1990). The vaneless diffuser has straight walls and a constant cross sectional area. The geometry operating conditions and measurement stations are summarised in Table 1.

Instrumentation and Measurement Technique
A triple hot wire probe was used to measure the velocities within the diffuser. A single wire (Dantec 55P11) was aligned...
circumferentially with a double wire (Dantec 55P61) arranged with each wire in the axial radial plane and at 45° to both the radial and axial directions. This mutually perpendicular arrangement of wires was therefore capable of resolving the axial, circumferential and radial velocity components and the directional sign of the axial component. It was assumed that the radial and circumferential components remained positive throughout the flow. There was no evidence in the results of either of these components reducing to zero and hence this assumption was justified. The hot wires were connected to three constant temperature hot wire anemometer bridges. The wires were then calibrated in two stages in a wind tunnel following the procedure of Jorgensen (1971) and detailed by Pinarbasi and Johnson (1994a).

An optical shaft encoder provided a pulse for every 1/3° of impeller rotation. This was used to trigger the simultaneous sampling of the three anemometer voltages through a Microlink data acquisition unit. Readings from 57 measurement points spanning one of the 19 impeller passages were logged on each of 230 consecutive impeller revolutions. The triple wire sensor was traversed in the axial direction in order to provide a mesh of data points for each measurement plane.

Analysis of Results
The 230 readings obtained at each measurement point for each of the three anemometers were used to compute the mean velocity components and Reynolds stress tensor using the calibration coefficients. The flow rate was also computed by numerical integration of the radial velocity component over each of the measurement planes. The maximum deviation of this flow rate from the mean for all stations was 4.8% which gives an indication of the overall experimental accuracy.

EXPERIMENTAL RESULTS
Detailed mean velocity and Reynolds stress results through the diffuser for the design flowrate have been presented previously by Pinarbasi and Johnson (1994a), (1994b) and also for off design flowrates by Pinarbasi and Johnson (1994c), (1994d). Here, results will only be presented at station 1, although mean velocity results from stations 2 and 3 have been used to evaluate the radial gradients of mean velocity using finite difference techniques.

Mean Velocities
The mean velocities measured at station 1, which is 12 mm downstream of the impeller blade trailing edge, are shown in Figure 2. The contours represent the radial component whereas the remaining components in the axial and tangential directions are depicted by an arrow at each measurement point. The flow pattern is typical of the 'jet-wake' discharge flow characteristic of centrifugal machines. The passage wake is located in the shroud/pressure side quarter of the passage with its core at y/y_e=0.35 with the jet filling most of the remaining three quarters of the passage cross section. The wakes from two consecutive impeller blades result in the low velocity bands on the pressure side (y/y_e=0) and suction side (y/y_e=1) of the passage.

Strong secondary flows exist in the blade wake region, but within the passage the cross velocity arrows depict an anticlockwise circulation of the bulk flow which is also typical of backswept centrifugal impellers (Krain (1988) and Farge and Johnson (1990)). Shear within the flow resulting from the boundary layers, the blade wakes, the passage wake and the secondary flows will thus lead to the production of turbulence which will ultimately dissipate resulting in total pressure loss. The regions in the flow of high shear can also be identified by considering the three components of vorticity. The vorticity is computed from the mean velocities using second order finite differences with an estimated accuracy of ±10%. Figure 3
Figure 3 Radial, Tangential and Axial components of dimensionless vorticity

shows the radial component of vorticity associated with the secondary flows observed in Figure 2. The vorticity is positive (anticlockwise) over the majority of the cross section with the highest values occurring in the passage wake core ($y/y_0=0.35$, $z/z_0=1$) and where the passage wake interacts with the blade wake on the shroud ($y/y_0=1$, $z/z_0=1$). Clockwise rotation is observed though on the suction side of the passage at $y/y_0=0.95$ due to shear between the secondary flow from shroud to hub within the blade wake and the secondary flow in the opposite direction associated with the passage vortex.

The axial component of vorticity shows very high levels in the blade wake, particularly on the suction side. High levels are also associated with the shear gradients at either side of the passage wake ($y/y_0=0.15$, $z/z_0=0.7$ and $y/y_0=0.7$, $z/z_0=0.7$). The tangential component of vorticity also shows high levels in the middle of the passage where high shear is present between jet and wake. However, the highest levels occur in the pressure side/hub corner within the thickened hub boundary layer observed in Figure 2.

Turbulent Kinetic Energy

The turbulent kinetic energy

shown in Figure 4 indicates peak levels of 7% within the blade wake and at the core of the passage wake. Significant levels are observed in the thickened hub boundary layer near the pressure side corner and also in the secondary flow interaction region close to the suction surface. The level within the jet is considerably lower between 0.5% and 1.5%.

Stress Tensor

The three direct stress and three shear stress terms in the stress tensor are shown in Figures 5 and 6. The direct stresses are presented as non-isotropy factors. Thus negative values correspond to below average contributions from the direct stress to the turbulent kinetic energy and positive values to an above average contribution. Figure 5 shows that the turbulence is highly non-isotropic within the blade wake, where the direct stresses are primarily in the radial and tangential directions. This might be expected since the vorticity (Figure 3) is in the axial direction. Although other regions of high non-isotropy exist, it is observed that the turbulence is essentially isotropic throughout the passage wake.

The three Reynolds stress components are presented in Figure 6 and discussed in detail by Pinarbasi and Johnson (1994b). These results are similar to measurements made downstream of a backward centrifugal impeller by Ubaldi et al (1993). High levels are observed in the blade wake and also in the regions of high shear in the thickened hub boundary layer near the pressure surface corner and in the interaction region between the passage vortex and the secondary flow within the blade wake. There are thus some similarities with the corresponding components of vorticity in Figure 3. The most significant difference is that the high levels of vorticity associated with the shear layers around the passage wake do not lead to significant levels of Reynolds stress. This identifies an important difference between the turbulence within the passage and blade wakes. The passage wake has its shear origins within the impeller and hence its shear
Figure 5 Non-isotropy factors

layers are more fully developed than those associated with the blade wakes which are only generated at the impeller exit. The authors therefore believe that the high levels of turbulent kinetic energy within the passage wake (Figure 4) are due to low frequency meandering in the passage wake position rather than high frequency turbulence. Hathaway et al (1992) have also suggested that meandering in the passage wake position occurs within their low speed centrifugal compressor impeller. The wake position is sensitive to the impeller passage flow rate (Johnson and Moore (1983)) and to the strength of the tip leakage jet (Farge, Johnson and Maksoud (1988)) and hence relatively small variations in flow rate and blade loading will cause significant variations in the passage wake position.

Turbulence Production

Reynolds (1895), in his famous paper which defined Reynolds stresses, derived an expression for turbulence production

\[
\frac{\rho u'^2}{U_2^2} - \frac{2}{3} \frac{u'^2}{3}
\]

Figure 6 Shear stress components

\[
\frac{\rho u'^2}{U_2^2} + \frac{2}{3} \frac{u'^2}{3}
\]

This expression has recently been used by Moore et al (1994a, b) to determine the origins of the turbulence generated within the tip vortex of a turbine cascade.

Moore et al (1994b) grouped these nine terms into (using the current notation) a direct stress term

\[
-\rho \mu' \frac{\partial u'}{\partial \phi} - \rho \frac{\mu'}{r} \frac{\partial u'}{\partial \theta} - \rho \frac{\mu'}{r} \frac{\partial u'}{\partial z}
\]

and three shear stress terms.
These components are shown in Figure 7 and contribute 22%, 39%, 22% and 17% respectively to the total turbulence production. The normal stress term contributes primarily to the turbulence production within the blade wake. The first of the shear stress terms is the strongest contributor to turbulence production within the blade wake although significant contributions are also made from the remaining shear stress terms. Figure 7 also shows quite clearly how the first and third shear stress terms generate turbulence within regions of primary velocity shear in the blade wake and boundary layers whereas the middle term, which is only dependent on the secondary velocities $u_r$ and $u_z$, generates turbulence only in regions of secondary velocity shear. These results suggest that the blade wake is responsible for about three quarters of the turbulence production with most of the remainder coming from the secondary flows. It should be noted that the relative contributions to turbulence production will be dependent on the impeller flow. The strength of the secondary flows at the impeller exit and the thickness of the blade trailing edge will be of particular significance.

If losses in centrifugal compressor diffuser flows are to be predicted accurately, the turbulence production terms within CFD codes need to be correctly modelled. The non-isotropy of the turbulence exhibited by the current results suggests that turbulence models which assume isotropy (e.g. k-epsilon and Baldwin-Lomax) are unlikely to be adequate for this purpose. It therefore follows that full Reynolds stress turbulence models will be required if reliable prediction of the details of turbulent production is to be achieved within diffuser flows.

CONCLUSIONS

1. Four regions of high shear rate are identified within the flow - within the blade wake, between the passage wake and jet, within the thickened hub boundary layer and between the blade wake secondary flow and passage vortex. Each of these regions is associated with high turbulent kinetic energy and with high levels in at least one component of vorticity.

2. Significant Reynolds stresses are generated in all these regions except between the passage wake and jet. The authors believe that these observations suggest that the passage wake is meandering in position.

3. The turbulence is strongly non-isotropic within the blade wake, but remains close to isotropic throughout the passage wake.

4. The turbulence production terms show how most of the losses can be attributed to mixing out of the blade wakes. Significant losses do occur however in the suction side shroud corner region due to interaction of the secondary flow within the blade wake with the passage vortex and within the hub boundary layer. There is no evidence of significant turbulence production within the passage wake.

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