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Wedge type vaned diffuser flow measurements in a low speed centrifugal compressor

Ümit Nazl1 Temel, Adnan Öztürk and Ali Pinarbasi*

Department of Mechanical Engineering, Engineering Faculty, Cumhuriyet University, 58140, Sivas, Turkey

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This study establishes effectiveness of vaned diffuser in comparison with vaneless diffusers in centrifugal compressor. Hot wire measurements at three inter-vane positions for mean velocity, turbulent kinetic energy and flow angle distributions in vaneless space and vane region of a low speed centrifugal compressor are presented. A low speed compressor with a 19 bladed backswept impeller and 16-wedge vanes diffuser was used. The flow entering diffuser closely resembles classic jet-wake flow characteristic of centrifugal impeller discharges. A strong upstream influence of diffuser vanes, which resulted in significant variations in flow quantities between vane-to-vane locations, was observed. Circumferential variations due to passage and blade wakes rapidly mixed out in vaneless space, although some variations were still discernible in vaned region. Impeller blade wakes mixed out rapidly within vaneless space than in an equivalent vaneless diffuser. Although, flow is highly non-uniform in velocity at impeller exit, there was no any separation from diffuser vanes. Presence of vanes accelerated mixing out process. Therefore, use of twisted vanes in diffuser would be beneficial in reducing losses.

Keywords: Compressor, Diffuser flow, Vaned diffuser

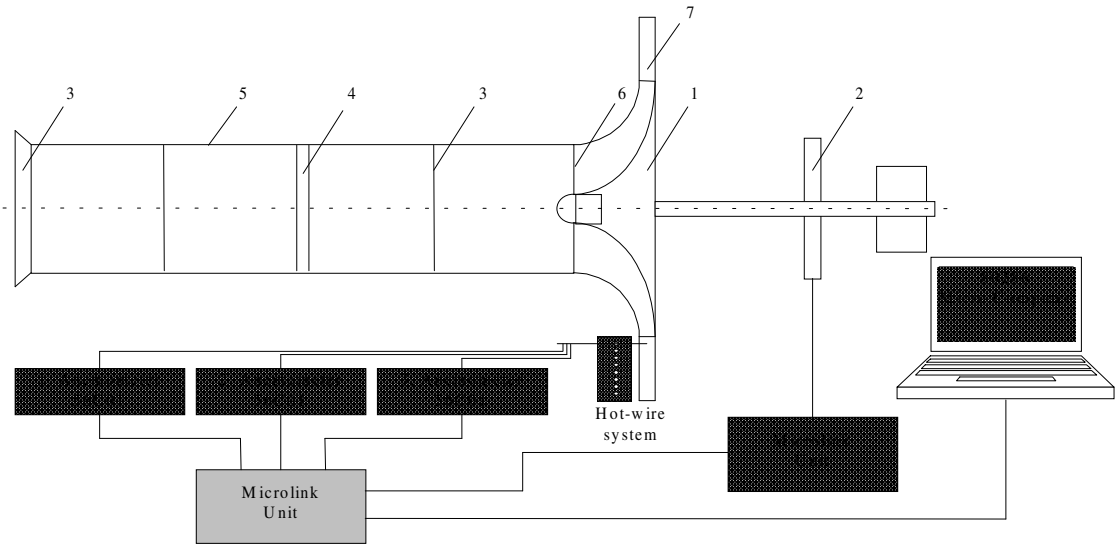
Introduction

Vaned diffusers are used in wide variety of compressor applications. Yoshinaga *et al*[1] used vaned diffusers to improve stage efficiency (4%) in comparison with vaneless diffuser for same diffuser outlet radius. Inoue & Cumpsty[2] observed that circumferentially averaged mean radial velocity profile in axial direction for vaned diffuser inlet was almost identical with that for vaneless diffuser. Krain[3] observed that in a flat straight channel diffuser with a splitter blade impeller an unsteady flow in vaned diffuser entrance region was highly distorted. These studies indicate that flow is highly distorted near leading edge of vane. This study presents measurement of detailed flow in a vaned diffuser in comparison with vaneless diffuser used in other studies[4-6].

Experimental Details

In this study, low speed centrifugal compressor test rig (Fig. 1) was used. In original impeller geometry[7], radial outlet section was replaced to give a 30° backswept outlet angle[8]. Vaned diffuser (Fig. 2) contained 16-wedge type blades mounted at 45° to give zero incidence angle at leading edge. Vaneless space ratio (1.1) was selected as a typical value[2,3,9]. Vane design gives a constant cross sectional flow area through vaned part of diffuser and hence absolute Mach number (value, 0.1) is constant throughout. Measurement locations [L (radial distance from impeller outlet) / R_o (impeller outlet radius)] for different stations (S) were as follows: S1, 0.02; S2, 0.08; S3, 0.15; S4, 0.21; S5, 0.27; S6, 0.33; S7, 0.39; and S8, 0.45. Geometry and operating conditions for an impeller were as follows: inlet blade radius at hub (R_h), 88.75 mm; inlet blade radius at shroud (R_s), 283.75 mm; outlet radius (R_o), 454.6 mm; backswept blade angle (β), 30°; number of blades (N), 19; outlet blade span (b), 72.3 mm; rotating speed (n), 500 rpm; mass flow rate (m), 0.1311; and mean absolute outlet flow angle (λ), 45°. Geometry and operating conditions for a vaned diffuser were as follows: wedge angle, 22.5°; vane angle, 45°; number of vanes, 16; inlet/outlet area ratio, 1; diffuser throat width, 178.5 mm; diffuser inlet diam, 912.5 mm; diffuser outlet diam, 1524 mm; channel length, 208.75 mm; vane span, 71.3 mm; vaneless space radius ratio, 1.1; and length width ratio, 0.37. This design was chosen for comparison with vaneless diffuser with constant cross sectional area.

*Author for correspondence
Tel.: +90 346 2191339; Fax: +90 346 219 11 75
E-mail: pinarbasi.ali@gmail.com



1-Impeller 2-Pulley Driven by motor 3-Screens 4-Honeycomp 5-Inlet Duct 6-Seal 7-Diffuser

Fig. 1—Schematic of centrifugal compressor test rig

Measurement Method

Measurements were made with hot wire anemometry using three orthogonal wires [single wire Dantec (55P11) oriented in circumferential direction, and double wire probe Dantec (55P61) oriented in axial and radial planes]. A shaft encoder was used to control simultaneous sampling of output voltages, which were converted into corresponding wire cooling velocities using King's law as

$$E^2 = A + BU_e^c \qquad \ldots(1)$$

where, E, voltage; U_e, effective cooling velocity; A, B & C, calibration coefficients of each wire.

Directional sensitivity of wires was undertaken to determine relationship between effective wire cooling velocities and orthogonal velocity components. Each of the instantaneous velocity components, according to Jorgensen[10], are written as

$$\begin{aligned} U_{1e}^2 &= k^2U_n^2 + U_t^2 + h^2U_b^2 \\ U_{2e}^2 &= h^2U_n^2 + k^2U_t^2 + U_b^2 \\ U_{3e}^2 &= U_n^2 + k^2U_t^2 + h^2U_b^2 \end{aligned} \qquad \ldots(2)$$

where U_n, U_t and U_b are normal, tangential and binormal velocity components relative to wire. Coefficients h and k were determined by changing wire orientation at fixed wind speed in wind tunnel. Probe was rotated through a range of ±50° yaw and 20° pitch angles. Errors in measured velocity and flow direction resulting from Eqs (1) and (2) were estimated as ±1 m/s and ±5° respectively. Major component of this error results from the assumption of constant coefficients in Eqs (1) and (2). After calibration, measurements were taken at 14 axial measurement locations on each passage cross section for different diffuser locations and flow rates. At each position (at 1/3° interval of shaft rotation), readings were taken for each of 230 shaft revolutions. Mesh of data points is thus 8x14x57 in radial, axial and tangential directions respectively. In case of vaned diffuser, measurements were repeated at three vane-to-vane diffuser positions (10%, 50% and 90%).

Results and Discussion

Mean velocity and Reynolds stress components are statistically calculated considering 230 consecutive impeller revolutions at each measurement point. Mean velocity ($\bar{u}$) and Reynolds stress components ($\overline{u_i'^2}$) were determined through Eqs (3) and (4), respectively.

$$\bar{u}_i = \frac{1}{230}\sum_{m=1}^{230} u_i \qquad i = r, \theta, z \qquad \ldots(3)$$

$$\overline{u_i'^2} = \frac{1}{230}\sum_{m=1}^{230} (u_i - \bar{u}_i)^2 \qquad i = r, \theta, z \qquad \ldots(4)$$

Turbulent kinetic energy (q) is defined as

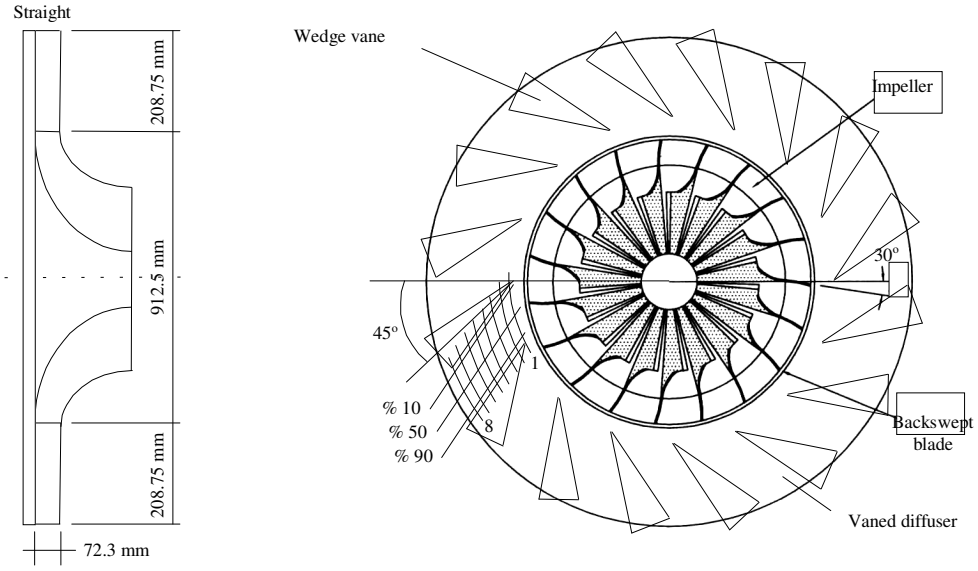


Fig. 2—Diffuser details

$$q = \frac{1}{230}\sum_{m=1}^{230}\frac{\overline{u_r'^2 + u_\theta'^2 + u_z'^2}}{2U_T} \qquad \ldots(5)$$

where U_T is blade velocity at impeller outlet; u, u_r, u_z, tangential, radial and axial mean velocity components. u_θ', u_r', u_z' tangential, radial and axial r.m.s fluctuating velocity components.

An error analysis indicated that uncertainties in mean velocity components were: Reynolds stress, $\pm 0.1\% \pm 1$ m/s; and turbulent kinetic energy, $\pm 0.05\% \pm 1$ m/s. Majority of this error results from assumption of constant coefficients in Eqs (1) and (2).

Station Results

Results were considered for four critical stations located at diffuser vaneless space (S1), entry region (S2), vaned part of diffuser (S3), and exit section of diffuser (S8), just represented at mid vane position (50%) diffuser despite of collected data at tree vane-vane position in here. Mean velocities were presented as contour diagrams, where PS and SS were represented with pressure side and suction side, and z/z_0 and y/y_0 represented with nondimensional axial and circumferential coordinates respectively. Arrows represented secondary velocities.

Station 1 (S1)

S1 is located within vaneless space close to impeller exit. In mean velocity results for S1 (Fig. 3a), flow rate adjacent to vanes is reduced with a corresponding increase in flow rate midway between vanes. Peak level of radial velocity midway between vanes is thus 3 m/s higher than close to vanes. Strong cross flows are observed within impeller passage wake in the pressure side (PS) shroud quarter because wake is increasing in size due to decrease in velocity within it. Wake is less marked at 50% vane-to-vane location, which suggests that pressure in this region may be lower at this location. Vanes also moderate circumferential variation (Fig. 3b) in absolute flow angle $[tan^{-1}(u_\theta / u_r)]$. Variation is generally less than 10° near vanes, whereas 30° variations are observed at mid vane position. Turbulent kinetic energy distributions (Fig. 3c) clearly show a substantial increase from 1% in majority of passage to between 6 and 8% within blade wake, which is however less than that observed for a vaneless diffuser downstream of same impeller[4]. This is because blade wake is weaker in vaned diffuser as it mixes out more rapidly downstream of impeller blade. Enhancement of mixing out is believed to be due to periodic unsteadiness induced by diffuser vanes.

Station 2

S2 is located within vaneless space close to vane leading edges. Highest flow velocities are observed at mid vane-to-vane location (Fig. 4a). Deficit in velocity in passage wake is little altered from S1, but wake has spread across shroud wall, particularly at two near vane positions. This is because of upstream effect of vanes in

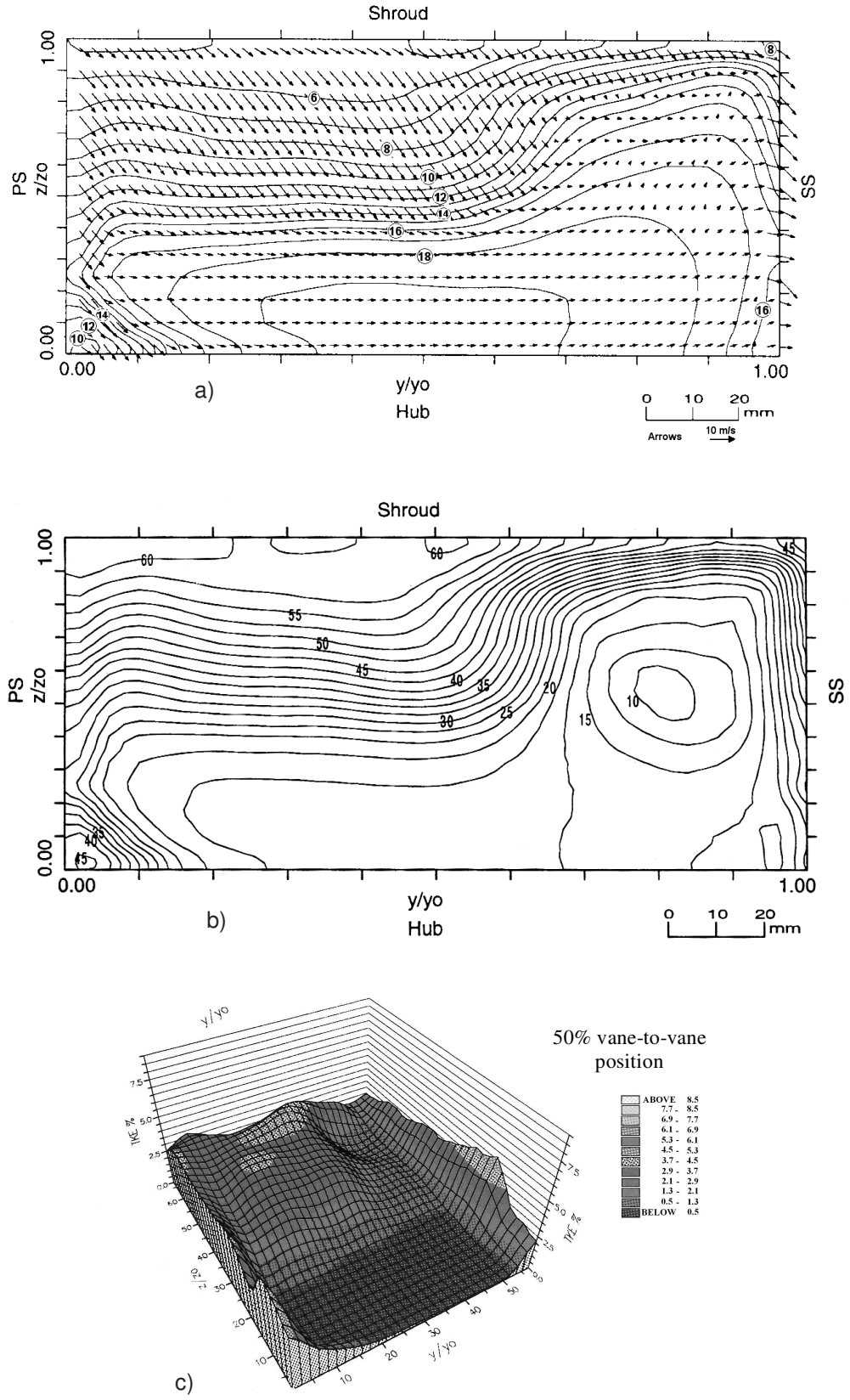
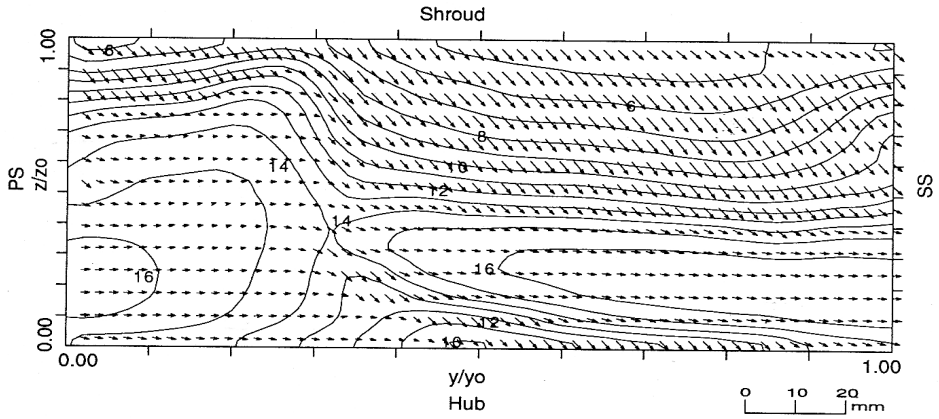
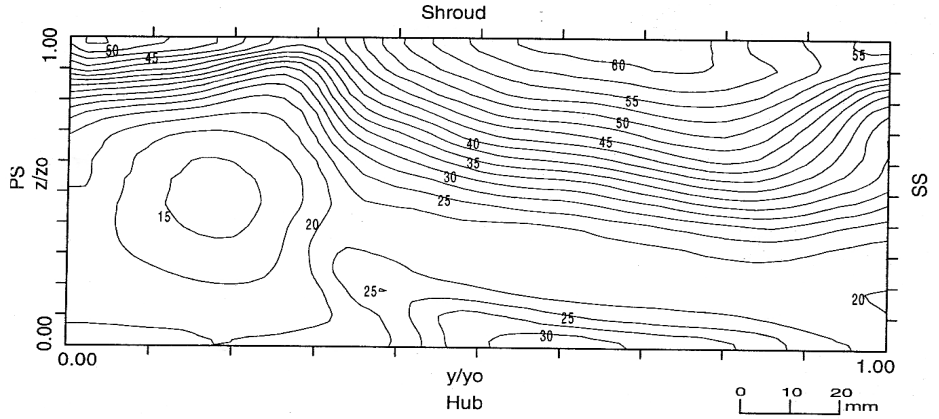
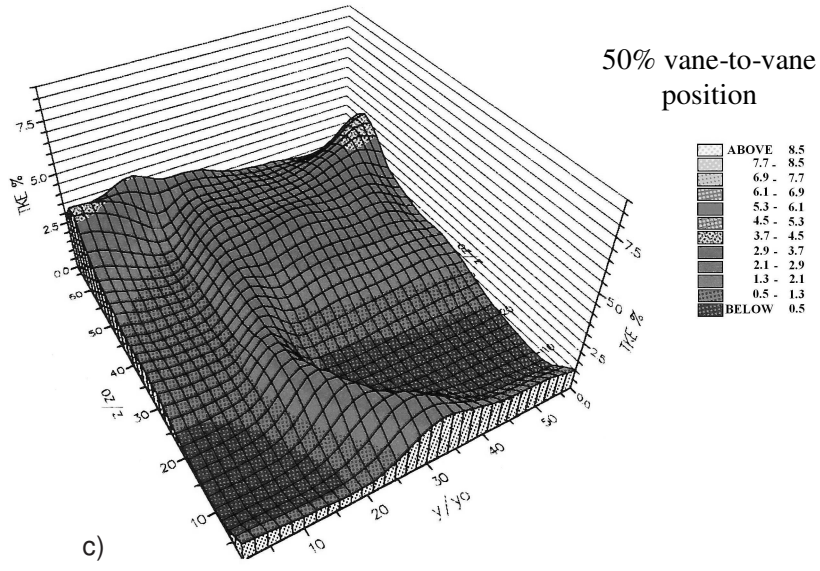


Fig. 3—Station 1 (50% vane-to-vane position): a) Mean velocities; b) Flow angle; and c) Turbulent kinetic energy



a)





c)

Fig. 4—Station 2 (50% vane-to-vane position): a) Mean velocities; b) Flow angle; and c) Turbulent kinetic energy

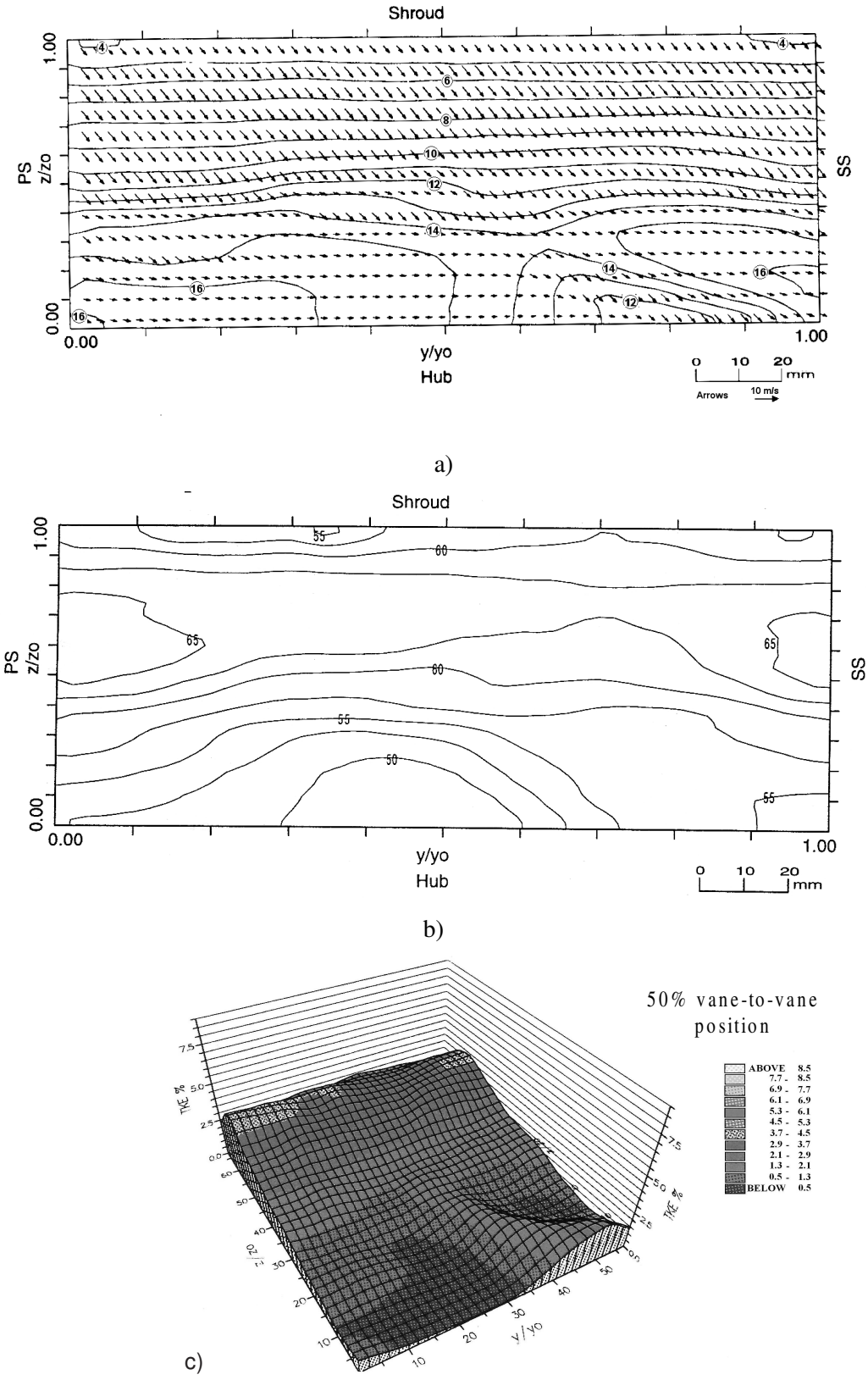
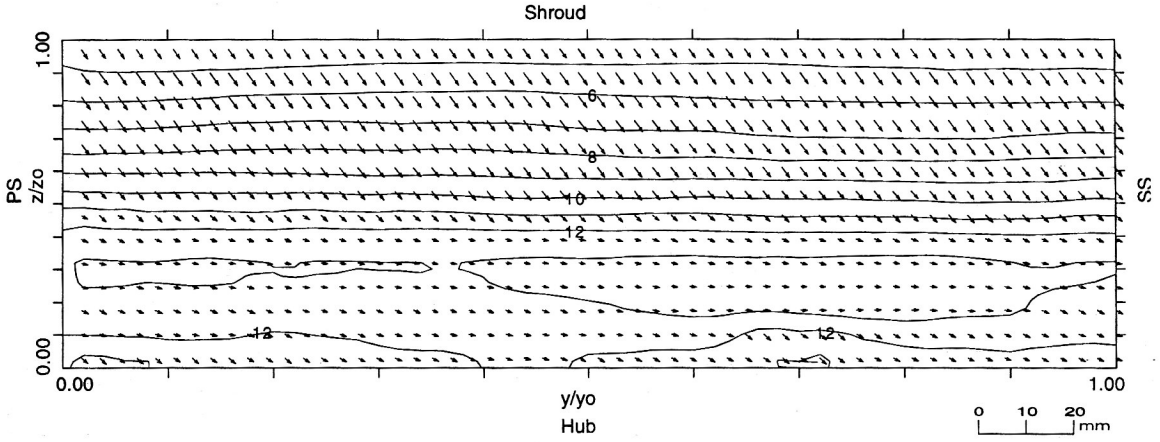
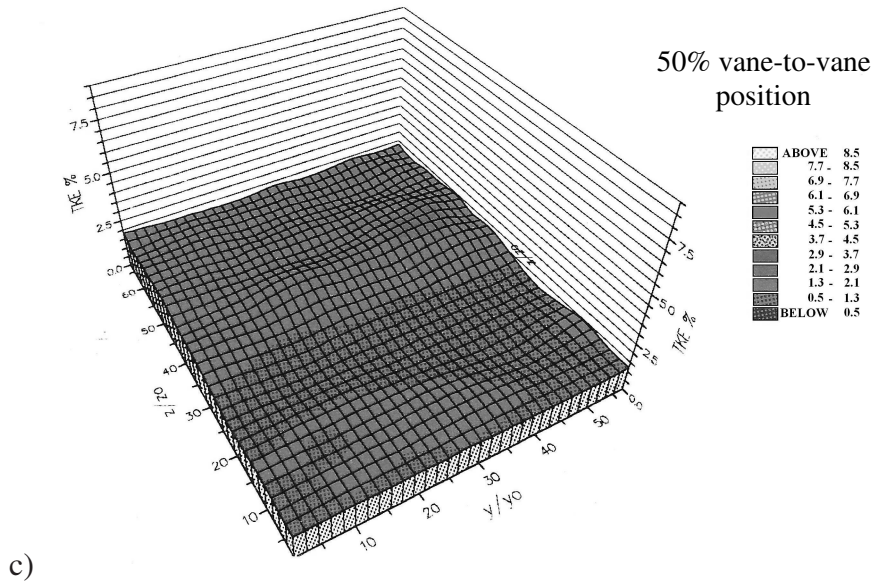


Fig. 5—Station 3 (50% vane-to-vane position): a) Mean velocities; b) Flow angle; and c) Turbulent kinetic energy

moderating circumferential variations in tangential/radial flow angle (<10°). There is however significant variation in flow angle in axial direction and so some twisting of leading blade edges would be beneficial (Fig. 4b). Passage wake has moved across the shroud because of tangential velocity between S1 and S2. Blade wake has also moved in this direction. This is due to increase in radial/tangential flow angle due to decrease in radial velocity. Close proximity of vanes also has a significant moderating effect on secondary velocities, which are largest at mid vane position. Turbulent kinetic energy diagram (Fig. 4c) shows how level within blade wake has decreased substantially from S1. Increase in width of blade wake is also clearly depicted. However, little change is observed in kinetic energy within passage wake. These observations are similar to those made for vaneless diffuser[4], where high level of kinetic energy was only associated with high levels of Reynolds stress in blade wake. High levels of kinetic energy in passage wake



a)



c)

Fig. 6—Station 8 (50% vane-to-vane position): a) Mean velocities; and c) Turbulent kinetic energy

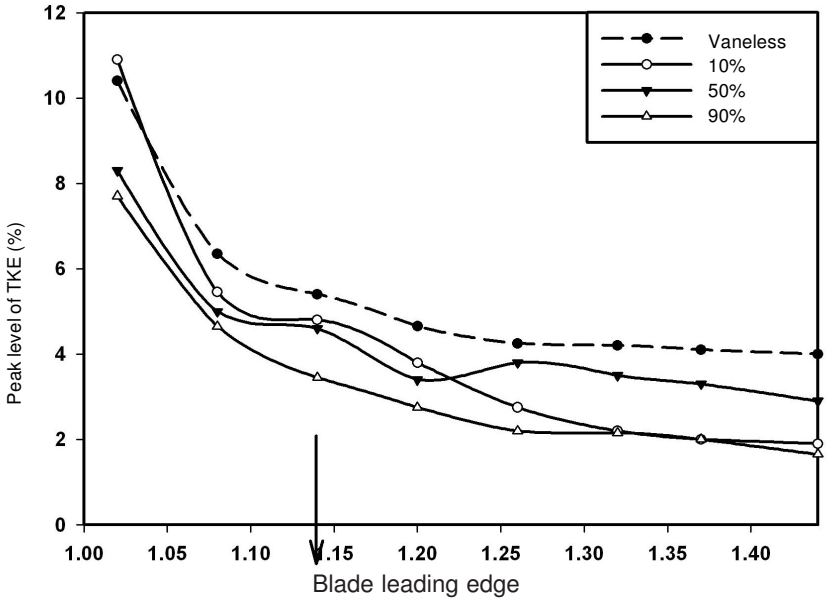


Fig. 7—Blade wake decay in the vaned and vaneless diffusers

were attributed to low frequency meandering of wake position.

Station 3

S3 is located within vaned part of diffuser. Circumferential variations in velocity (Fig. 5a) are now generally small. Maximum deficit in velocity is within blade wake at mid vane position, where strongest secondary velocities are also observed. Krain[3] and Inoue[11] also support these observations. Flow angles (Fig. 5b) indicate little circumferential variation and variations in axial direction have also moderated from S2 due to guiding effect of vanes. Flow does have a significantly different direction at mid vane position. Under kinetic energy diagram (Fig. 5c), levels within passage wake reduced slightly from S1.

Station 8

Mean velocities (Fig. 6a) show only negligible variations in circumferential direction. Axial component of velocity is also small except near shroud at mid vane position. Cross velocities appear to be associated with thickening shroud boundary layer, which is substantially thicker at mid vane position than near the vanes. Turbulent kinetic energy level (Fig. 6c) is more or less uniform across passage with a modest increase in shroud boundary layer at mid vane position.

Blade Wake Decay

Decay of wakes is commonly quantified using velocity deficit. Highly non-uniform flow means that velocity deficit cannot be determined accurately. For this reason, peak turbulent kinetic energy measured within blade wake is used as a measure of strength of blade wake (Fig. 7). Blade wake decays more rapidly for current vaned diffuser than in vaneless diffuser[4].

Conclusions

Measurements show that flow at diffuser inlet exhibits similar non-uniformities observed previously at impeller discharge. Presence of vanes in a diffuser significantly influence flow in vaneless space. Velocities are increased in mid vane position and decreased close to vanes. Mixing out of blade wakes is enhanced. Circumferential variations in velocity are rapidly mixed out near vane positions although some variations persist at mid vane position. Large variations in flow angle are observed at impeller exit, and although this does not appear to lead to separation of flow from diffuser vanes. Twisting of vane leading edge in axial direction is recommended to minimize flow separation.

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