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Internal Flow Investigation of a Centrifugal Pump at the Design Point

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***Abstract*:** The unsteadiness of the flow at the leading edge of a vaned diffuser represents a source of low efficiency and instability in a centrifugal pump. Furthermore, the internal flow of the impeller can be affected by asymmetric downstream conditions, which results in extra flow unsteadiness and instabilities. The improvement of machine performances can only be achieved if there is a progress in the comprehension of the nature of the complex flow that develops at the gap between the rotor and the stator.

This paper presents the results of LDA investigation of the internal flow of a centrifugal pump equipped with a backswept impeller and a vaned diffuser. The data taken in phase with the impeller rotation are presented as animated frames reconstituting a temporal evolution of the flow in radial planes at the diffuser inlet.

***Keywords*:** centrifugal pump, rotor stator interaction, internal flow, LDA.

1. Introduction

During the design of a turbomachine, the flow is considered steady and uniform at the entry of each element. For a centrifugal pump with a vaned diffuser, satisfying this assumption requires a large interface between the rotor and the stator so that the mixing process of the flow leaving the impeller can take place. Otherwise, the unsteady flow that enters the diffuser represents a source of low efficiency. Furthermore, the internal flow of the impeller can be affected by asymmetric downstream conditions, which results in extra flow unsteadiness and instabilities.

A number of authors have treated the problem of the interaction of the impeller and its surrounding. Miner et al. (1989) used a laser velocimeter to measure velocities within the impeller and the volute of a centrifugal pump and found that the relative velocity components distribution varies with the circumferential angle relative to the volute cutwater. Liu et al. (1994) has also used LDA for the internal flow investigation and found that the asymmetric volute alters the relative flow, the flow rate from each impeller passage varied with the volute circumferential position by up to 25 percent at an off-design flow rate. Inoue and Cumpsty (1984), Sideris and Van Den Braembussche (1987) and Arndt et al. (1989, 1990) have been concerned with the action of the diffuser. For internal flow studies, a large number of the experimental investigations that revealed the presence of a jet-wake structure at the discharge of centrifugal rotors was concerned with compressors as done by Eckardt (1975, 1976), Johnston and Eide (1976), Johnson (1978), and more recently Rohne and Banzhaf (1991) and Ubaldi et al. (1993). Krain (1988) found a velocity profile that differed widely from the jet-wake type flow.

During this study, LDA measurements revealed the presence of a jet-wake flow structure. The location and the extension of the wake seem to be affected by the proximity of the diffuser vanes.

2. The SHF Impeller

The test impeller, shown in Fig. 1, is a low specific speed (w_s=0.577) shrouded impeller. It has seven backswept blades with an exit angle of 22.5 degrees relative to the tangential direction. The main geometric data of the impeller and its operating conditions are resumed in Table 1. For optic access, the shroud was made out of clear Plexiglas and a clear window was realized on the casing. The impeller was run with a vaned diffuser and a spiral casing of industrial type. The diffuser is a straight wall constant width with six vanes (Fig. 1). The main dimensions of the diffuser are listed in Table 1.

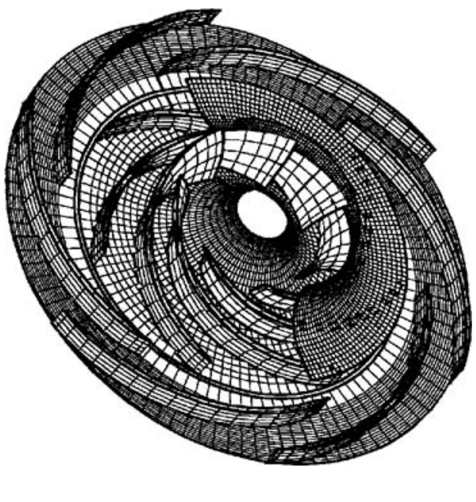


Fig. 1. Test impeller and vaned diffuser.

Table 1. Impeller and diffuser characteristics.

impeller parameter				**description**
R_1	=	98.25	mm	inlet blade radius at the shroud
R_2	=	177.25	mm	impeller exit radius
b_2	=	26.7	mm	blade height
Z	=	7		blade number
b_2	=	22.5	Deg	blade angle
Q_n	=	0.0774	m³/s	design flow rate
N	=	1188	rpm	rotational speed
f	=	0.118		design flow coefficient
y	=	0.481		design head coefficient
w	=	0.577		specific speed
diffuser parameter				
R_3	=	182	mm	diffuser inlet radius
R_4	=	199.3	mm	leading edge radius
R_5	=	258	mm	diffuser exit radius
b_3	=	28.1	mm	diffuser width
Z	=	6		vane number
a_3	=	12	Deg	vane angle

3. Test Rig

Experiments were performed on a centrifugal pump test facility consisting of a closed rig equipped for the overall performance characterization of the machine. Water enters the impeller through a straight suction pipe of 1.4 m in length. The net flow rate traversing the pump is measured by an electromagnetic flow meter with a precision of 0.2 percent at the actual experimental conditions. The impeller is driven by a variable speed DC motor of 45 kW power at 1500 rpm mounted in balance mode for torque measurement.

4. Measuring Equipment

Measurements of velocity distribution at the impeller discharge and in the diffuser were obtained by using an LDA system. The light source is a 5 W laser operating in multiline mode in order to operate with the blue (488 nm) and green (514.5 nm) wavelengths. Modular optics are used to derive a three beam configuration exiting a 310 mm focusing lens. The LDA system was used in a back-scatter mode. The optical components, including the laser and the photomultipliers, were mounted on a three axis traversing system to place the probe volume at the location of interest (Fig. 2).

To relate the velocity measurement to the angular position of the impeller, an optical encoder is used to synchronize measurement with machinery. The encoder fixed on the pump shaft gives the position of the measuring point in the blade-to-blade plane with a resolution of 3600 angular readings for one shaft revolution.

For each test point in the axial direction, an average of 10000 data points were taken. The sampling rate was arbitrary and depended on the nature of the LDA signals. The data processing system, consisting of two burst signal analyzers with built-in synchronization inputs for cyclic phenomena, is connected to a PC. A software uses encoder pulses to phase-sort the velocity samples versus the phase angle. The data is also phase-averaged to compute the mean and RMS velocities.

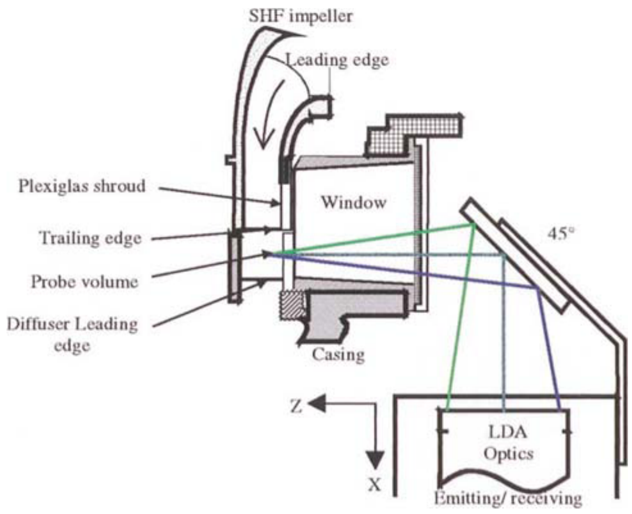


Fig. 2. Optic access to the measuring sections.

5. Discussion of the Experimental Results

In order to study the interaction of the impeller and the diffuser, the flow was investigated at 7 radial planes located at different angular positions in the diffuser as given on Table 2 and shown on Fig. 3. For each plane, the flow was investigated at eight radial sections in the impeller and the diffuser (Table 2), for each radii, sixteen points were explored.

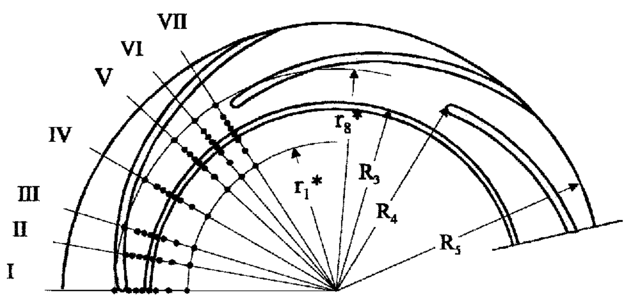


Fig. 3. Locations of the measurement planes within the impeller and the diffuser.

Table 2. Measuring sections.

angular position in the diffuser pitch	a (deg)		r/R_2
I	0 (or 60)	in the impeller	0.8182
II	10		0.9092
III	17		0.9784
IV	30	in the diffuser	1.0179
V	43		1.0451
VI	50		1.0846
VII	57		1.1298
			1.1980

The laser Doppler velocimeter used for internal flow investigation is two dimensional, only the velocity components in the radial and circumferential directions were measured; the velocity in the axial direction was not measured and was supposed to be negligible.

The absolute velocity is presented as a function of the angular position within the impeller channels and the axial distance z/b. The shroud and the hub are located at z/b=0 and z/b=1 respectively. Figures 4 and 5 represent the flow field evolution at the nominal flow rate. The reported results correspond to the measurements obtained at the impeller mid height (z/b=0.5) and three sections corresponding to r/R_2= 0.818, 0.909 and 0.978.

In these figures, the same velocity distribution repeated over seven periods corresponds to the flow in the seven impeller channels; it illustrates the periodic nature of the flow at the impeller exit.

In Fig. 4 corresponding to the radial plane located at a=0 degree, a significant evolution of the flow structure is present along the impeller passages. At r/R_2=0.818, the flow structure is the same as expected for a potential flow having deep velocity gradients between the suction and pressure sides of the impeller passages. On the pressure side, we notice a high tangential velocity component Cu, while the radial component Cr is low. In the next section (r/R_2=0.909), both components reach a minimum at the channel center. In fact, the flow pattern starts

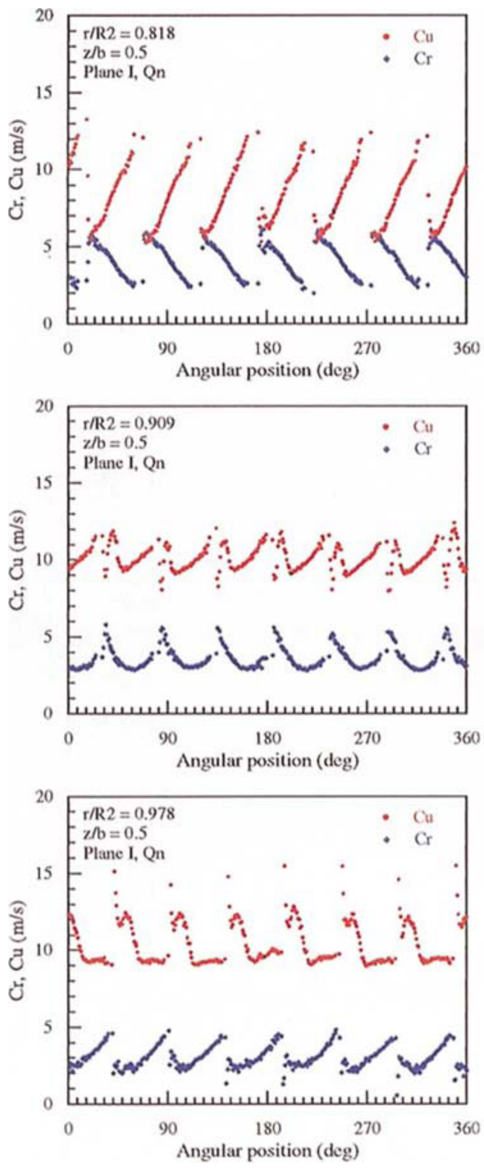


Fig. 4. Fluctuation of the flow velocity at the impeller channels, a=0°, z/b=0.5.

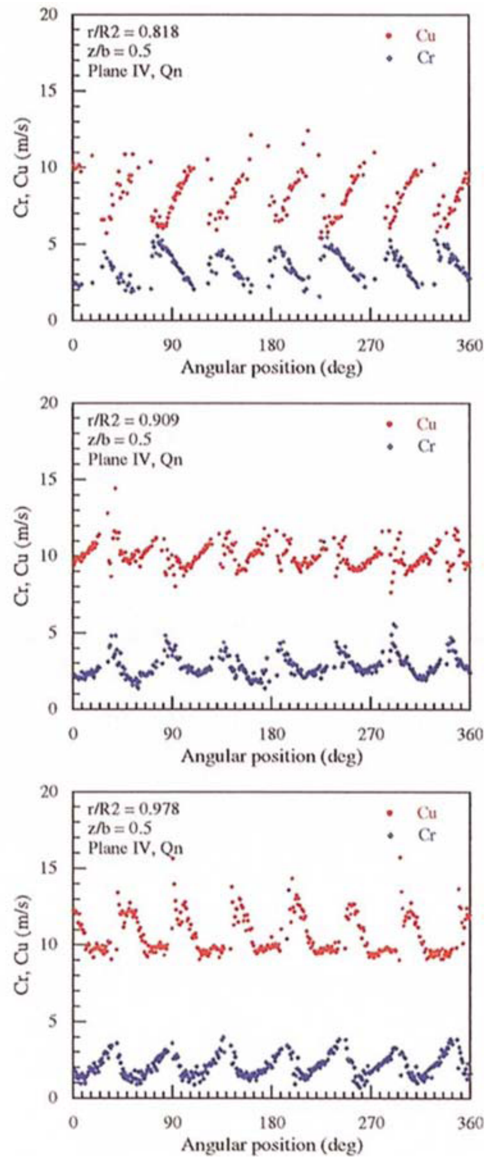


Fig. 5. Fluctuation of the flow velocity at the impeller channels, a=30°, z/b=0.5.

to deviate from the theoretical model. This is particularly observed near the impeller exit (r/R_2=0.978) where the velocity gradient is inverted. The radial velocity is now increasing from the suction side to the pressure side. One should also notice the important velocity fluctuations registered in the proximity of the suction side. The tangential component is more uniform over a large area of the passage near the pressure side.

At a=30° (Fig. 5), the flow in the impeller preserves obviously its characteristics of periodicity and unsteadiness. At the inner radius, the potential flow lost a little of its intensity and very significant local fluctuations come to accentuate the level of turbulence as the impeller passages reach this part of the diffuser pitch.

The proximity of the leading edge of the diffuser (a=0°) causes an acceleration of the flow and a decrease of turbulence level.

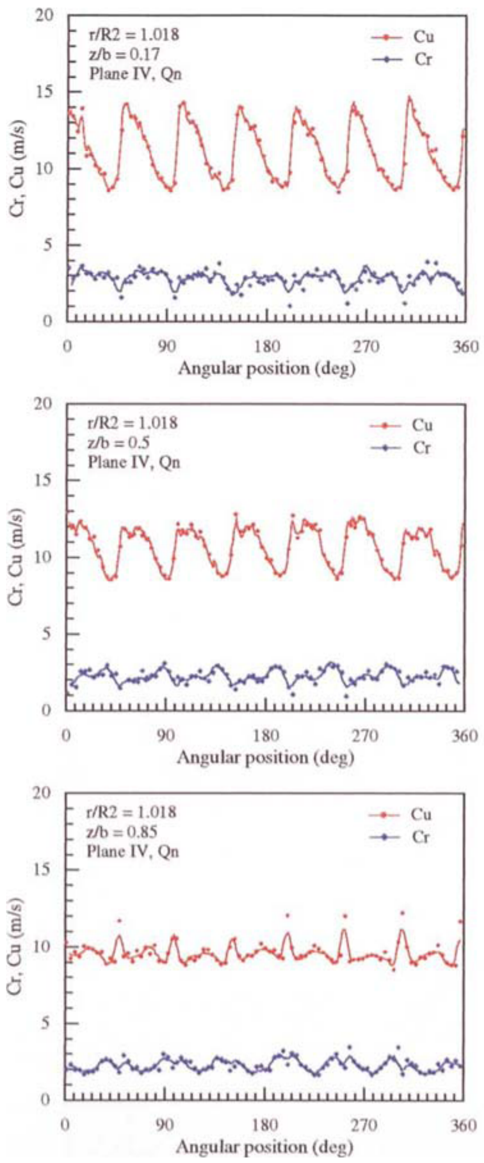


Fig. 6. Fluctuation of the flow velocity at the impeller discharge, a=30°, r/R_2=1.018.

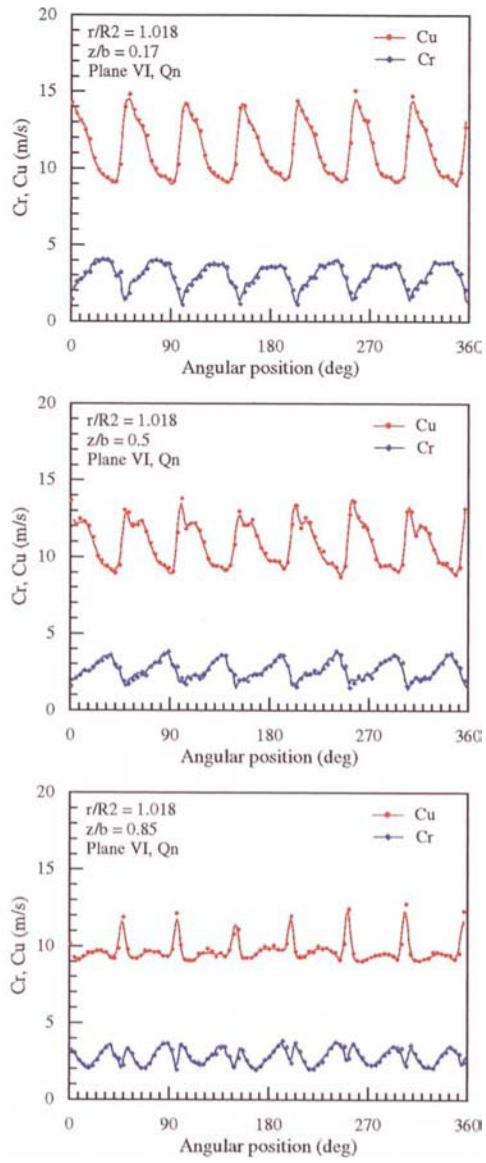
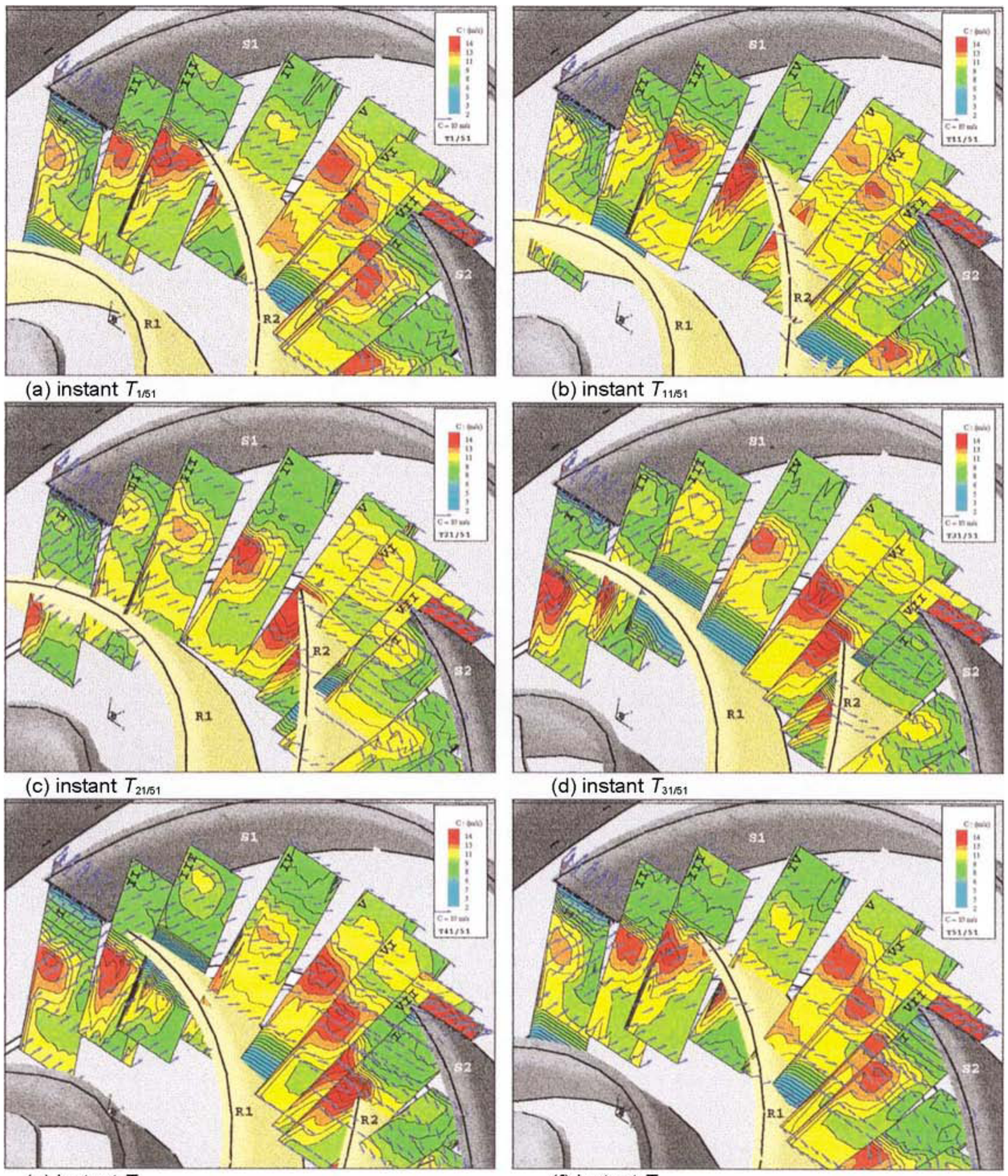


Fig. 7. Fluctuation of the flow velocity at the impeller discharge, a=50°, r/R_2=1.018.

Figures 6 and 7 report the results obtained at the impeller discharge at two radial planes (IV, VI) thus at different distances from the diffuser leading edge. We notice that the periodicity of the velocity profiles is deeply modified between the two sections, whereas the mean values remain constant.

On Fig. 8, the results are presented for a single impeller passage. The flow field represents an average flow obtained from the seven impeller channels. The passage was divided into segments of 1 degree angular width. For each segment, the mean velocity was obtained and the results are shown as constant velocity contours. The frames represent, at different instants (T_1, T_{11}, T_{21} T_{31}, T_{41} and T_{51}), the impeller blades R_1 and R_2 rotating from left to right in front of the diffuser vanes S_1 and S_2. The time period between the passage of two successive impeller blades, in front of a diffuser vane, corresponds to 51 instants.

The frames show a very deep evolution of the flow structure as it approaches the impeller discharge. At the inner radii, a large part of the channel is occupied by parallel constant velocity contours corresponding, thereby, to the potential flow distribution with a positive gradient from the suction side to the pressure side. This invinscid behavior of the flow is only altered in a small section of the channel confined to the suction side/shroud surface. In this region, the flow has a high absolute velocity, a deep velocity gradient is registered. As the fluid is achieved to the trailing edge, the location of the high velocity fluid moves from the suction side/shroud corner to the suction side of the impeller. The flow is therefore organized in a jet-wake structure as reported by previous research works



(a) instant $T_{1/51}$ (b) instant $T_{11/51}$

(c) instant $T_{21/51}$ (d) instant $T_{31/51}$

(e) instant $T_{41/51}$ (f) instant $T_{51/51}$

Fig. 8. Time resolved flow field frames.

on centrifugal machines. In this structure, the wake corresponds to the high absolute velocity flow and the jet is characterized by a low absolute velocity.

In the impeller, the flow structure remains unchanged when the distance between the measurement section and the leading edge of the diffuser is modified. At the design point, the flow within the impeller is hardly affected by the presence of the vaned diffuser as noticed by Toussain et al. (1998).

At the diffuser inlet, the mixing of the jet-wake structure depends on the proximity of the diffuser vanes. In the first half of the diffuser pitch (planes I to IV), the wake core moves towards the channel mid height. In plane IV, the flow is almost uniform when it reaches r/R_2=1.084, which indicates that the mixing process of the complex flow coming out of the impeller has finished. On the contrary, the jet-wake is still present in the second half (planes V to VII). The more important radial velocity, as seen in Fig. 7, diffuses the jet-wake structure farther away of the impeller exit. Consequently, it retards the mixing process.

The flow has a negative incident when approaching the leading edge of the diffuser (planes VII and I). On the pressure side of the vanes, an important velocity is registered, indicating that the flow rushes to enter into the diffuser channels. In the suction side, a flow slow down prevents the flow structure leaving the impeller to go further in the radial direction.

6. Conclusion

This detailed investigation of the internal flow within a centrifugal pump, at its design point, has permitted to study the effect of a vaned diffuser on the flow inside the impeller. From the actual results, we can conclude that the impeller-diffuser interaction is limited to the impeller exit and it does not have any upstream influence on the flow. The frontier between the rotor and the stator seems to be not easily crossed. The mixing process of the flow at the impeller discharge is affected by the presence of the diffuser vanes. The first half of the diffuser pitch is characterized by an early mixing of the flow. Whereas, in the second half, the flow entering the diffuser channel is still presenting its periodicity due to the impeller. This result indicates, therefore, that the diffuser performances may be affected by the complex flow coming out the impeller. Further investigations in the diffuser and at different operating conditions are projected to better understand the rotor-stator interaction.

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