The Internal Flow Investigation of a Centrifugal Pump

by

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ABSTRACT

This paper provides the results of a detailed flow investigation within a centrifugal pump equipped with a vaned diffuser. The measurements made with a laser-Doppler velocimeter were carried out at the impeller design point. In a previous paper concerning the same machine, El Hajem (1998) found a jet-wake structure developing at the impeller exit. During the actual study, measurements were obtained in the impeller and the diffuser at different measuring planes relative to the diffuser vanes. Results are presented as animations reconstituting a temporal evolution of the flow at the diffuser inlet.

Unsteady velocity measurements obtained in phase with the impeller angular position gave access to the flow inside the impeller channels where three sections were explored. For each section, results were obtained as a function of the position of impeller blades relative to the diffuser vanes. Thus time resolved details of the flow could be examined for a better understanding of the complex unsteady flow existing between the two interacting blade rows.

The analysis of the impeller flow field indicates the presence of a complex, unsteady and periodic flow. It is organised in a jet-wake structure. The wake is characterised by low relative velocities and is localised in the suction side/shroud corner. At this flow rate, it seems that presence of the vanes has only a limited effect on the impeller flow structure, except when the blades suction side are facing the diffuser vanes.

At the impeller discharge, the time-resolved sequences show that the mixing process of the unsteady and periodic flow leaving the impeller is affected by the presence of the diffuser. At the leading edge, at the suction side of diffuser vanes, the flow is rapidly mixed. Whereas, when approaching the diffuser throat, the flow still shows its periodic character observed at the impeller outlet. This indicates that the diffuser is subject to unsteady inlet conditions that can alter its performances.
I 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 [1] 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 (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 (1984), Sideris (1987) and Arndt (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 (1976), Johnson (1978), and more recently Rohne (1991) and Ubaldi (1993). However, Krain (1988) found a velocity profile that differed widely from the jet-wake type flow. In a previous study, the authors (El Hajem et al., 1998) found a jet-wake structure developing at the impeller exit. 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.

II EXPERIMENTAL SETUP

II.1 THE SHF IMPELLER

The test impeller, shown in Figure 1, is a low specific speed ($\omega_s=0.577$) 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 (Figure 1). The main dimensions of the diffuser are listed in table 2.

II.2 TEST RIG

Experiments were performed on a centrifugal pump test facility (1, figure 2) 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 (11, figure 2) 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 (10, figure 2).

Figure 1 : Test impeller and vaned diffuser

Figure 2 : Test rig Measuring equipment
Table 1 Impeller characteristics

<table>
<thead>
<tr>
<th>impeller parameter</th>
<th>description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R_1$ = 98.25 mm</td>
<td>inlet blade radius at the shroud</td>
</tr>
<tr>
<td>$R_2$ = 177.25 mm</td>
<td>impeller exit radius</td>
</tr>
<tr>
<td>$b$ = 26.7 mm</td>
<td>blade height</td>
</tr>
<tr>
<td>$Z$ = 7</td>
<td>blade number</td>
</tr>
<tr>
<td>$\beta_2$ = 22.5 deg.</td>
<td>blade angle</td>
</tr>
<tr>
<td>$Q_n$ = 0.0774 m$^3$/s</td>
<td>design flow rate</td>
</tr>
<tr>
<td>$N$ = 1188 Rpm</td>
<td>rotational speed</td>
</tr>
<tr>
<td>$\phi$ = 0.118</td>
<td>Design flow coefficient</td>
</tr>
<tr>
<td>$\psi$ = 0.481</td>
<td>Design head coefficient</td>
</tr>
<tr>
<td>$\omega$ = 0.577</td>
<td>specific speed</td>
</tr>
</tbody>
</table>

Table 2 Diffuser characteristics

<table>
<thead>
<tr>
<th>diffuser parameter</th>
<th>description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R_3$ = 182 mm</td>
<td>diffuser inlet radius</td>
</tr>
<tr>
<td>$R_4$ = 199.3 mm</td>
<td>Leading edge radius</td>
</tr>
<tr>
<td>$R_5$ = 258 mm</td>
<td>diffuser exit radius</td>
</tr>
<tr>
<td>$b_3$ = 28.1 mm</td>
<td>Diffuser width</td>
</tr>
<tr>
<td>$Z$ = 6</td>
<td>Vane number</td>
</tr>
<tr>
<td>$\alpha_3$ = 12 deg.</td>
<td>Vane angle</td>
</tr>
</tbody>
</table>

Table 3 Measuring Sections

<table>
<thead>
<tr>
<th>Angular position in the diffuser</th>
<th>$r^* = r/R_2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha$ (deg)</td>
<td></td>
</tr>
<tr>
<td>I 0 (or 60)</td>
<td>0.8182</td>
</tr>
<tr>
<td>II 10</td>
<td>0.9092</td>
</tr>
<tr>
<td>III 17</td>
<td>0.9784</td>
</tr>
<tr>
<td>IV 30</td>
<td>1.0179</td>
</tr>
<tr>
<td>V 43</td>
<td>1.0451</td>
</tr>
<tr>
<td>VI 50</td>
<td>1.0846</td>
</tr>
<tr>
<td>VII 57</td>
<td>1.1298</td>
</tr>
</tbody>
</table>

II.3 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 argon ion laser operating in multiline mode in order to operate with the blue (488 nm) and green (514.5 nm) wavelengths. Modular optics with a 310 mm focusing lens are used to derive a three beam configuration. 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 (figures 3, 4).

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 M in the axial direction, an average of 10 000 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 velocity and the turbulence intensity.

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 3 and shown on figure 5. For each plane, the flow was investigated at eight radial sections in the impeller and the diffuser (table 3), for each radii, sixteen points were explored.
III DISCUSSION OF THE EXPERIMENTAL RESULTS

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.

III.1 VELOCITY FLUCTUATIONS OF THE IMPELLER INTERNAL FLOW

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 (Figure 6). Field evolution at the nominal flow rate. The reported results correspond to the measurements obtained at passage midheight (z/b=0.5) for three sections corresponding to r/R<sub>2</sub> = 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 unsteadiness of the flow at the impeller exit.

In figure 6 corresponding to measuring radial plane I facing the diffuser leading edge, a significant evolution of the flow structure is present along the impeller passages. At r/R<sub>2</sub> = 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<sub>2</sub> = 0.909), both components reach a minimum at the channel center. In fact, the flow pattern starts to deviate from the theoretical model. This is particularly observed near the impeller exit (r/R<sub>2</sub> = 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.

III.2 RELATIVE FLOW DISTRIBUTION AT THE IMPELLER EXIT

Figure 8 represents the temporal evolution of the phase averaged relative velocity, within a single impeller passage, as sensed successively at the planes I, II, III, IV, V, VI located at different distances with respect to the diffuser leading edge. These frames show an impeller passage with the blades rotating in the clockwise direction. The shroud and the hub are respectively at the foreground and the background of the frames. The relative velocity is presented as constant velocity contours in the blade-to-blade plane.
A highly flow distortion is observed in the axial and blade-to-blade directions. A large area of the passage, extending from the pressure side to the passage centre and from the shroud to the hub, is traversed by a uniform flow. On the contrary, the remaining passage is dominated by an important velocity gradient and an accumulation of low momentum fluid in the suction side/shroud corner. The velocity reaches a minimum at the suction side, at about \( z/b = 0.4 \).

The strong velocity gradient and the accumulation of low velocity fluid near the suction side show that the flow field leaving the impeller is organized as a jet-wake structure as reported by previous investigations and as was expected according to an analysis of the action of the centrifugal forces (El Hajem 1998) on the flow field. In this paper, the authors found in a previous paper that the wake core corresponds to a flow separation.

The flow structure is almost the same at all the measuring planes, except the plane II where the wake core seems to vanish when the suction side of the impeller blades has just passed behind the trailing edge of the diffuser vanes. We also notice that the velocity gradient in the blade-to-blade direction is less pronounced as for the other planes.

### III.4 Turbulence Intensity Distribution

In order to evaluate the importance of the flow fluctuations within the impeller passages, the turbulence intensity was calculated by the same procedure given recently by Toussaint (1998). First, the periodic unsteady flow in the seven impeller passages is ensemble (phase) averaged to constitute a mean average flow in a single passage. Each impeller passage was divided into 51 measuring windows of an angular width \( \Delta \theta = 1 \) degree. For each angular bin \( \theta \), the mean velocity is computed by combining the \( n \) measurements of the equivalent locations in the different passages during the measuring time:

\[
\bar{C}_{(M)} = \frac{1}{\Delta \theta} \int_{\theta - \Delta \theta / 2}^{\theta + \Delta \theta / 2} C_{(M,t)} \, dt = \frac{\sum_{i=1}^{n} C_{(M,i)}}{n}
\]

Therefor, for each angular position \( \theta \) of the mean passage, the rms values associated with each fluctuating velocity is defined as follow:

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**Figure 7:** Blade to blade flow distribution measured at planes I, II, III, IV, V, VI, \( r/R = 0.978 \)
The turbulence level can be calculated for every angle using the following definition:

\[
\text{Tu} = \frac{\text{rms}}{C_0} \times 100 = \frac{\frac{1}{n} \sum_{i=1}^{n} [C_i - C_0]^2}{\frac{1}{n} \sum_{i=1}^{n} C_i^2} \times 100
\]

Sample plots of the turbulence intensity distribution in the interblade passage, just upstream of the impeller discharge, are shown on figure 8 and 9. On figure 8 the evolution of turbulence is given for plane I at the shroud \(z/b=0.17\), passage midheight \(z/b=0.5\) and at the hub \(z/b=0.85\). On these figure, the turbulence is given as a function of the angular position for witch the suction side of the passage corresponding to zero, while the pressure side corresponds to 51 deg.

At the hub region, the turbulence intensity is low and its distribution is uniform in the impeller passages, it hardly exceeds 15 percent. As reported by Cattanei et al. (1998), the high turbulence intensities up to 40 percent observed at the suction side/shroud corner and the passage center are due to the presence of the wake as observed if figure 9. This region of the passage is occupied by a low relative velocity flow having important velocity gradients. The hub region where the jet is prevailing the turbulence intensity is much lower. This distribution of the turbulence intensity remains almost identical in all the measuring planes except at plane II where a different turbulence distribution was observed (figure 9). In this plane, the fluctuations are higher and the turbulence level reaches 80 percent at the suction side/shroud corner. It is the same case at the passage midheight where it approaches 60 percent. The hub region \(z/b = 0.85\) is also affected and is characterized by a distinct rise of flow unsteadiness at the passage midspan.
III.5 ABSOLUTE FLOW DISTRIBUTION

On figure 10, the results are presented as constant absolute velocity contours. The frames represent, at different instants T1, T11, T31 and T41, the impeller blades R1 and R2 rotating from left to right in front of the diffuser vanes S1 and S2. 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 sections of the impeller, a large part of the channel is occupied by the parallel constant velocity contours corresponding, to the potential flow distribution with a negative gradient from the suction side to the pressure side. This inviscid behavior of the flow is only altered in a small section of the channel confined to the suction side/shroud surface. In this region where the flow has a high absolute velocity and an important velocity gradient is registered. As the fluid is achieved to the impeller discharge, the core 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 on centrifugal machines. Even though there is a slight alteration of the velocity, the flow patterns in the impeller remain almost unchanged when the distance between the measurement section and the leading edge of the diffuser is modified. The frontier between the rotor and the stator seems to be not easily crossed. At the design point, the flow within the impeller is hardly affected by the presence of the vaned diffuser as noticed by Toussaint (1998).

At the diffuser inlet, the mixing of the jet-wake structure depends on the proximity of the diffuser vanes. At the radial planes III and IV, the wake core moves towards the channel mid height. In plane IV, the flow is almost uniform when it reaches r/R=1.084, this indicates that the mixing process of the complex periodic flow coming out of the impeller has finished. In the following planes, the jet-wake is still present. A more important radial component of the velocity, as seen in figure 7, seems to delay the mixing process. When approaching the diffuser throat (planes VI and I), an
important velocity is registered on the pressure side of the vanes, the flow ruches to enter in the diffuser channels. In the suction side, a flow slow down prevents the flow structure leaving the impeller to diffuse in the radial direction.

**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 results 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.

**REFERENCES**


