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# Data reduction problems using a 3 holes directional pressure probe to investigate mean flow characteristics in the vaneless gap between impeller and diffuser radial pump.

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**Abstract.** Among several different measurement techniques that have been already performed and presented in a radial impeller pump model including PIV, a directional pressure probe has been used to obtain mean velocity field and stagnation pressure between impeller outlet and the inlet vaned diffuser sections. These measurements are supposed to get more information not only about global pump head coefficient including vaned diffuser ones but also about impeller performances itself. Pressure probe information is affected by rotor-stator interactions and impeller rotation, and this paper presents a way to explain and correct pressure probe indications in order to achieve a better evaluation of overall impeller mean performances. The use of unsteady RANS calculation results is found to be a useful way to perform better data reduction analysis for this purpose

## 1. Introduction

Hydraulic radial pump is often design including vane diffuser downstream the impeller. Depending on design specific speed, vane diffuser leading edges radius may be closed to the impeller outlet one. The radial gap between these two radii is the so-called vaneless diffuser part. Global pump performances are generally obtained measuring overall static pressure difference  $\Delta P_s$  between inlet and outlet pump sections. Total (stagnation) pressure differences  $\Delta P_t$  is then deduced assuming the following:

- No pre-swirl and constant velocity at pump inlet section
- Constant absolute velocity at pump outlet section after the vane diffuser.

Impeller performance itself is usually determined by torque measurements and by wall static pressure at impeller outlet section assuming mean radial and tangential velocity components coming from mass flow rate and mean slip factor coefficient.

In a research loop model, complementary measurements can be performed using different kind of local measurements such as optical techniques like LDV and/or PIV ones and intrusive directional pressure probe traverses. Only this last measuring technique is able to give total pressure evaluation and consequently correct impeller and pump loss evaluations.

Concerning impeller performance, depending on model outlet impeller section dimensions, and even with very small probe dimensions, one can have difficulties to get correct measurements on local flow characteristics mainly because of the following aspects:

- Probe's dimensions may cause local blockage effects, so that the measured velocity level will be higher than the real one.
- Small probe pressure hole diameter may cause Reynolds effects on pressure levels

- When vane diffusers are present, and depending on the operating point, upstream effects coming from diffuser leading edges and blade to blade gradients may be strong enough to modify probe indications compared with ideal flow inlet conditions with no velocity gradients.
- More important effects concern rotor-stator interactions on probe indications itself. These effects generally lead to higher total pressure probe levels compared with real ones even for the case where no absolute angle variation in time are present.
- Another source of errors may be attributed to leakage flow effects coming from existing radial gap between fixed and rotating parts of the pump. This effect is considered not so negligible for the present test case.

All these effects must be taken into account when pressure probe traverse are performed in such pump section close to impeller outlet. One has to keep in mind that directional probe calibration always takes place with constant upstream pressure and velocity under stationary flow conditions.

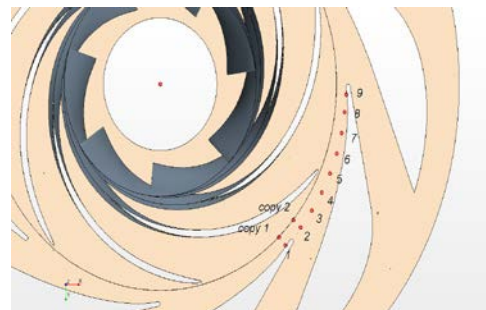
The present paper present an analysis of the different error sources in order to get better evaluation of impeller outlet performances. Data reduction has been performed for several operating points, but only results corresponding to diffuser design conditions are presented here. Diffuser design conditions corresponds to the reduced pump mass flow of  $Q/Q^*=0.77$ ;  $Q^*$  corresponds to optimum impeller mass flow rate.

## 2. Experimental results obtained from probe traverses

The directional pressure probe that was used is shown in Figure1. Measurements took place for different mass flow rates and for 8 different blade to blade positions as shown in Figure 2. For each blade to blade positions, axial traverse from hub to shroud have been perform, in order to cover a complete blade to blade section in front of the diffuser plane (section 3).



**Figure 1.** View of the directional probe



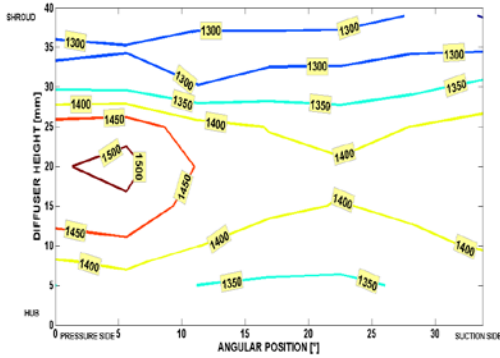
**Figure 2.** Locations of probe traverses in the vaneless gap

Only mid traverse results between hub and shroud are analysed here for simplicity. As already said, the following results are given for a particular mass flow rate corresponding to the diffuser vane design operating point.

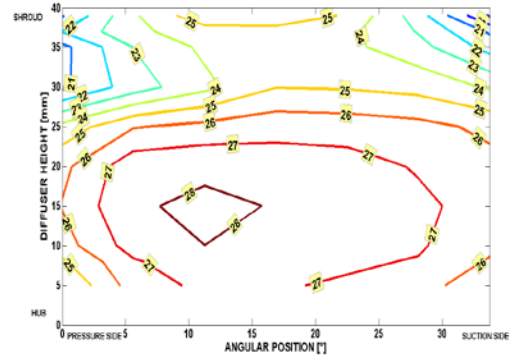
It has been already show in previous papers, references [1], [2] and [3], that rotor stator interactions are quite important, even at design point, due to the particular design of the diffuser vanes.

In order to get a complete view of probe experimental results in the whole section, iso-lines of total pressure, tangential and radial absolute velocity components and absolute angles are given in Figures 3a to 3d.

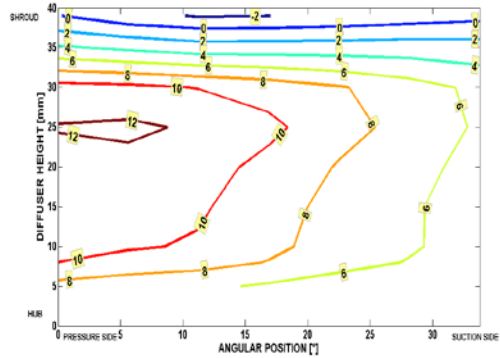
Local mid height results concerning averaged blade to blade total and static pressures extracted from previous Figures are shown respectively Figures 4 and 5, with the corresponding absolute velocity angle distribution on Figure 6. The results of unsteady calculations are also plot on the same Figures in order to be compared with experimental results.



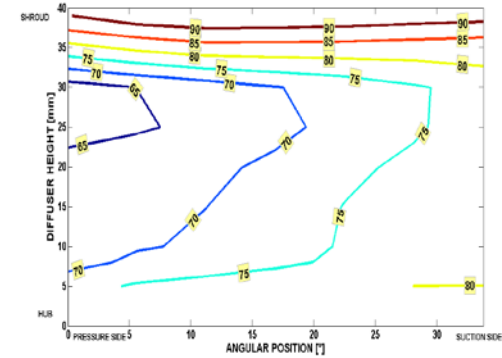
**Figure 3a.** Total pressure iso contours (pascals)



**Figure 3b.** Tangential velocity iso contours (m/s)



**Figure 3c.** Radial velocity iso contours (m/s)



**Figure 3d.** Absolute angle iso contours (degree)

### 3. Experimental results analysis

First important feature that has to be pointed out concerns the maximum measured total pressure level that can be seen on Figure 3a. This maximum level, which value is around 1500 Pascal, must be compared with the theoretical total pressure that can be obtained using Euler 1D equation or measured torque already obtained by previous works for the same mass flow rate, reference [2].

The corresponding theoretical total pressure value  $\Delta P_{t,th}$  is given by the following relation:

$$\Delta P_{t,th} = (U_2 * VU_2 - U_1 * VU_1)$$

If  $VU_1$  is assumed to be zero, then  $\Delta P_{t,th}$  reaches 1500 Pascal, taking into account measuring errors and uncertainties. This means that the pressure probe measurement give the same amount of theoretical pressure which is impossible because of the profile pressure losses occurring inside the impeller at mid span and the vaneless diffuser losses too.

When averaging the blade to blade total pressure distribution for different hub to shroud positions corresponding to the results given on Figure 4, it can be seen that the measured total pressure level is still higher than the numerical one with a pressure difference of 200 Pascal. Results analysis obtained for other mass flow rates show that the more the mass flow rate decrease, the more the pressure difference increase compared with numerical simulation results. This difference, if the calculated results are supposed to be correct, may be partly attributed to velocity fluctuations due to rotor stator interactions. As a consequence, it can be seen that probe static pressure is also affected by these effects and the static pressure difference is evaluated to be overestimated by a level of 60 Pascal as shown in

Figure 5. This has a direct consequence on the absolute velocity value given by the probe that is greater than the numerical one.

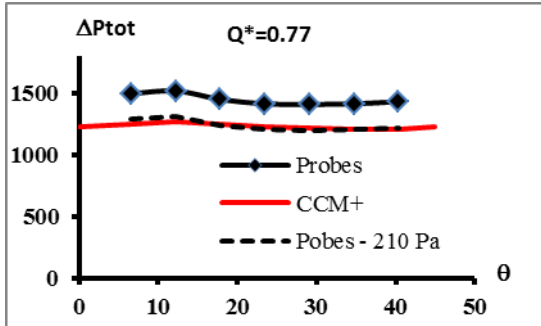


Figure 4. Mid height total pressure evolutions

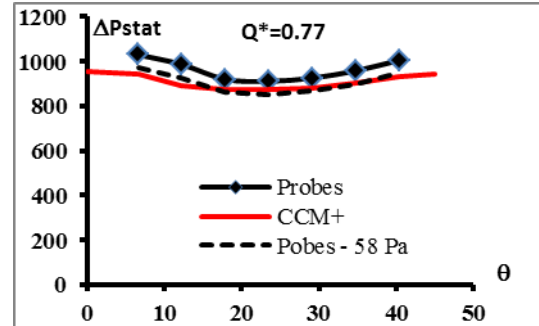


Figure 5. Mid height static pressure evolutions

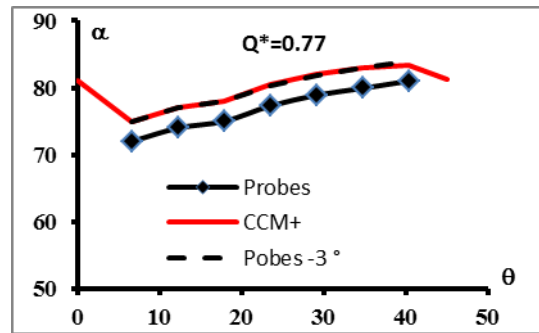


Figure 6. Mid height absolute angle evolutions

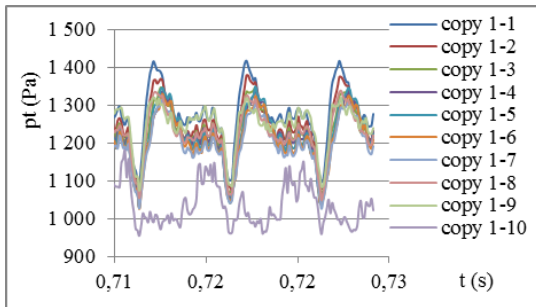


Figure 7. Instantaneous total pressure evolutions downstream of the impeller

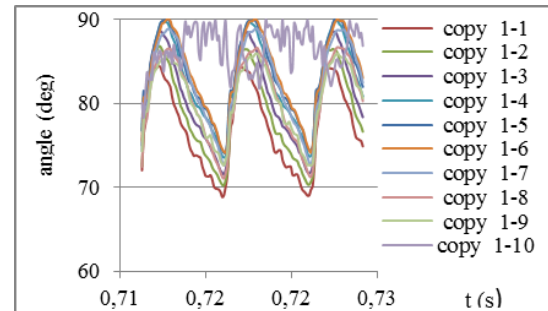


Figure 8. Instantaneous absolute angle evolutions downstream of the impeller

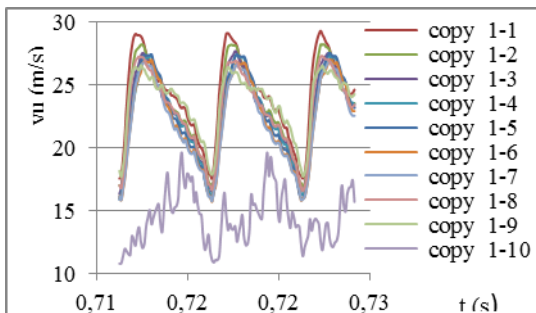


Figure 9. Instantaneous absolute tangential velocity evolutions

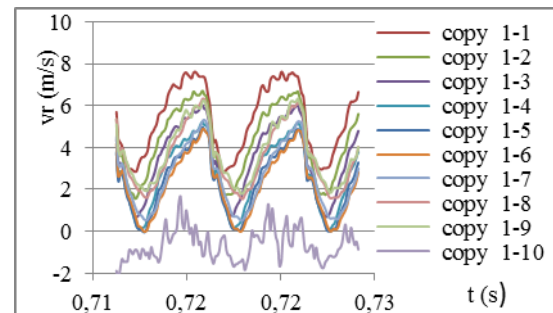


Figure 10. Instantaneous absolute radial velocity evolutions

#### **4. Probe pressure data corrections**

Unsteady simulation results are used in order to get an evaluation of the effects of flow fluctuations. As an example, Figure 7 show the numerical instantaneous total pressure level in the vane less diffuser part of the pump model for a particular point in the blade to blade plane, for several hub to shroud positions. The time dependant radial and tangential velocity components are also given in Figures 8 and 9 respectively for three consecutive impeller blade passages. So, these pressure and velocity fluctuations are integrated by the pressure probe. The total pressure corresponding quadratic evaluation leads to explain an increase of roughly 100 to 120 Pascal for the total pressure probe indication and a measured probe velocity excess of 10 to 12 per cent. This error level does not reach the 200 Pascal that have been found in Figure 4. Part of the explanation can be explained as explain below.

#### **5. Absolute angle correction**

Looking at the radial component the velocity obtained by the probe given on Figure 3d, which is a result of the use of probe calibration curves, one can get an average value that must corresponds to the real pump mass flow rate. The calculated value given by the probe is much higher, going to 28 per cent more than the measured mass flow rate which has been already obtained by a specific flow meter device.

This means that the unsteady effects cannot explain alone the velocity excess of 10 to 12 per cent that has been already determined.

One can add a complementary effect due to probe set up inside the vaneless diffuser gap, the so-called blockage effect. This one cannot exceed 3 per cent taking into account the probe diameter of 0.002 m and the leading blade tangential thickness of 0.005m compared to the diffuser blade spacing of 0.210 m. This means that about 15 per cent of error has to be checked on radial velocity component and this can be only explain by an angle error coming from the probe. This error corresponds to an angle of the order of 3 degrees, because radial component depends on the cosine of the angle. This value corresponds to the difference shown in Figure 6 when comparisons is given between unsteady mean numerical results and experimental ones

In order to find this angle difference, it is useful to look both at the numerical instantaneous angle evolution on Figure 8, the total pressure with absolute velocity components given on Figures 8, 9 and 10, respectively. Low values of absolute angle are reached for high radial velocities combined with high total pressure. This means that the probe is integrating more radial velocity fluctuation associated with low values of absolute angles.

The probe settling angle was generally below the real flow one. By making a weight averaging of the numerical tangential velocity component and total pressure, it is possible to obtain the corrected angle error due to the probe misalignment including unsteady effects on the directional pressure probe holes. This leads to an angle probe coefficient of 25 per cent. This value is quite closed to the 22 per cent already found in previous section discussion.

A difference of 3 per cent of mass flow rate is still remaining. This amount is probably due to leakage flow that is, in the present experimental set up, entering the diffuser inlet plane. As a consequence, the diffuser mass flow rate is larger than the impeller one due to this "positive" leakage flow as explain in reference '[4]'.

Finally, due to this angle error of 5 degrees, and using probe calibration curves, the final total pressure found to be overestimated by 5 per cent and the dynamic pressure by 8 per cent. This leads to a dynamic pressure correction of 40 Pascal and a total pressure correction of 25 Pascal. This last value must be added to the 120 Pascal that have been found only considering blockage and unsteady effects without probe angle misalignment correction.

In order to get a verification of these effects, the probe has been also used for a zero mean mass flow rate operating point. The measured angle obtained by the probe at mid span is equal to 70 degrees, instead of 90 degrees. So, this proves that the angle correction which is established by probe calibration under stationary flow conditions is unable to find the mean real angle.

## 6. Conclusions

- Directional probe data reduction process, used in severe unsteady conditions that exists closed to a radial impeller outlet section followed by vane diffuser, must involve an exhaustive analysis of error sources.
- Probe data generally exhibits higher total pressure levels with absolute angles values that gives too much radial velocity component and so, an overestimated mass flow rate.
- Locally, the hydraulic efficiency, calculated without total pressure corrections, may reach values greater than 1. This is one of the main reasons why probes are not used in such pumps.
- The results of unsteady numerical approaches could help to perform a better data reduction using directional probes even with usual calibration procedures.
- Part of the errors comes from the settling probe angle. It should be better to perform probe measurements allowing probe to rotate in order to get pressure equilibrium between the two holes that are used to obtain the flow angle. This technique will avoid part of the angle error due to unsteadiness.

## 7. Acknowledgments

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## 8. References

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