EXPERIMENTAL DETERMINATION OF THE REVERSE FLOW ONSET IN A CENTRIFUGAL IMPELLER

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In 1967 Dr. Canavelis completed his Science Doctorate thesis on Cavitation Erosion in Turbomachinery.

Since 1968, Dr. Canavelis has been an engineer at Bergeron, S.A. and is presently Head of the Hydraulics Department.

ABSTRACT

The importance of problems related to machines operating at partial flow rate led the Société Hydrotechnique de France (S.H.F.) to form a working group on the subject.

In order to better understand partial flow phenomena and to validate flow calculations and correlations, this group has defined an experimental program, the main results of which are presented in this paper.

A low specific speed, 7 bladed impeller was designed. Two different size similar pump models were then manufactured for experiments with water (outlet diameter 354 mm) and air (outlet diameter 517 mm).

The paper gives test results issued from flow visualization and measurements of various quantities varying with flow rate. Flow visualizations were made in the inlet and outlet regions of the impeller using threads, smoke and cavitation.

A determination of the critical flow rate at the inlet and outlet was also made using the measurements of: pump power input, mean and fluctuating wall pressures, mean and fluctuating velocities (with directional probes, laser velocimeter, and hot wires).

The influence of the variations of some parameters has been examined:

- comparison of vanless and 6 vanes diffuser,
- effect of a diffusor width modification,
- influence of variable leakage flow rate,
- comparison of results obtained with water and air.

These results were finally compared with correlations available in the technical literature and were used to validate the computer programs developed.

INTRODUCTION

In the division “Engineering Applied Fluid Mechanics” of the Societe Hydrotechnique de France, a working group was created five years ago to study the performance of uncompressible flow turbomachinery in partial flow operating range.
The working group was composed of various people from universities, research centers, pump manufacturers or pump user companies.

This subject proved to be of high interest for manufacturers as well as for users who have met in recent years to report on more and more cases of flow instabilities and mechanical troubles in pumping installations where very low flow rates are reached.

Such problems were encountered in thermal and nuclear power plants during partial load operation as well as in storage plants where important level variations were met.

Among the various projects undertaken by the working group, emphasis was placed on finding a method for the prediction of the reverse flow onset.

It was decided to follow two different paths of research: on one hand by providing flow calculations in the impeller, and on the other hand by using correlations. It was well known that the flow calculations corresponding to low flow rate were very complex because of the presence of viscous phenomena, detachment zones and secondary flows in the hydraulic channels of the impeller. This method of prediction appeared to be a very difficult task. On the other hand, the use of general prediction formulas, as presented in the literature, proved often to be inaccurate when applied to pump design different from those used to establish these correlations [1,2].

In both cases it was necessary to make experimental observations in order to validate the theoretical results. Some of the experimental results found by the working group are presented in this paper.

**EXPERIMENTAL SET UP**

**Impeller Design**

The impeller shown in Figure 1 was designed by Neypic. It is a rather conventional design whose main data are:

- outlet diameter \( D_2 = 400 \text{ mm} \)
- outlet width \( b_2 = 30.0 \text{ mm} \)
- inlet diameter \( D_1 = 220.2 \text{ mm} \)
- number of blades \( Z_r = 7 \)
- outlet blade angle \( \beta_{2h} = 22^\circ 30' \)
- mean blade thickness \( e = 7.0 \text{ mm} \)
- nominal speed of rotation \( N = 1200 \text{ rpm} \)
- nominal flow rate (water) \( Q_{n} = 0.112 \text{ m}^3/\text{s} \)
- total head (water) \( H_n = 31 \text{ m} \)

This gives the following nondimensional quantities:

\[
\varphi = \frac{Q_n}{2\pi R_2^3 b_2 \omega} = 0.118
\]

\[
\psi = \frac{gH_n}{\omega^2 R_2^3} = 0.481
\]

\[
\Omega_n = \frac{\omega (Q_n)^{1/2}}{(gH_n)^{1/6}} = 0.577
\]

![Figure 1. View of Impeller.](image)

**Impellers for tests in water and air**

Two impellers, geometrically similar to each other and corresponding to the design defined above, were cast in bronze. Table 1 gives the main dimensions of those impellers: the size of the water-impeller was chosen to be adapted easily to the existing test-rig of INSA (in Lyon), the air-impeller was chosen as large as possible taking into account speed of rotation, available power, and desired Reynolds and Mach numbers.

The main results of a dimensional check made on both of the impellers are given in Table 2.

**Water Test-rig**

Experiments were made on the INSA pump test-rig, a general view of which is shown on Figure 2. The suction pipe had a straight length of 1.40 m. The suction pipe and casing suction side were made of Perspex (Figure 3). The flow rate was measured by means of a venturi with piezometric tubes. The

<table>
<thead>
<tr>
<th>Table 1: Impellers Operating Data.</th>
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</thead>
<tbody>
<tr>
<td>Fluid</td>
</tr>
<tr>
<td>---------</td>
</tr>
<tr>
<td>Water</td>
</tr>
<tr>
<td>Air</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td></td>
</tr>
</tbody>
</table>
Table 2: Impellers Dimensions.

<table>
<thead>
<tr>
<th>Fluid</th>
<th>D2 (mm)</th>
<th>D1 (mm)</th>
<th>b2 (mm)</th>
<th>β2b</th>
<th>e</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Design</td>
<td>Measured</td>
<td>Design</td>
<td>Measured</td>
<td>Design</td>
</tr>
<tr>
<td>Water</td>
<td>354.4</td>
<td>354.56</td>
<td>195.1</td>
<td>195.1</td>
<td>26.58</td>
</tr>
<tr>
<td>Air</td>
<td>516.80</td>
<td>516.80</td>
<td>284.5</td>
<td>284.5</td>
<td>38.76</td>
</tr>
</tbody>
</table>

total head was measured in sections A and B (Figure 4) by use of two validyne pressure transducers according to the pump test codes (these transducers were calibrated with a mercury manometer). The pump was driven by a 45 kW D.C. motor with a free floating stator for torque measurement. The speed of rotation was permanently recorded from an optical gauge.

![Figure 2. Scheme of Water Test Rig.](image1)

Figure 2. Scheme of Water Test Rig.

![Figure 3. Water Impeller with Leakage Flow Control Device.](image2)

Figure 3. Water Impeller with Leakage Flow Control Device.

![Figure 4. Location of Pressure Measuring Sections in Water Test Rig.](image3)

Figure 4. Location of Pressure Measuring Sections in Water Test Rig.

Air Test-rig

The larger impeller was tested in air in an open test-rig. The impeller was driven by a D.C. motor dynamometer whose speed could be varied up to 3000 rpm.

The main dimensions of the casing are shown in Figure 5.

![Figure 5. View of Air Test Pump.](image4)
As can be seen, the impeller was fitted to a vaned-diffuser (6 blades) geometrically similar to the water diffuser. The diffuser was set up in a volute of uniform cross-section the width of which could be varied.

It must be noticed that seal A (Figure 5) was removed during the measurements reported here.

A schematic description of the test-rig is given in Figure 6. The performance of the "pump" was determined from the following measurements:

- ambient conditions (pressure, temperature, relative humidity),
- pressure difference between pump inlet and pump outlet,
- temperature measurement at pump outlet,
- pressure difference at orifice plate for flow measurement,
- speed of rotation and shaft torque.

The flow rate was measured with an orifice plate according to AFNOR or ISO standards [3, 4]. For a given rotation speed, and for each position of the flow control device, all the measurements were recorded and checked with a micro-computer.

![Figure 6. Scheme of Air Test Rig.](image)

**Performance Curves**

Overall performance results for water and air are given in Figures 7 and 8; the performance was translated into the design dimensions defined in the Impeller Design section.

It must be noticed that the power input curves corresponding to the tests in air were corrected by deducting estimated disc friction losses.

The best overall efficiency points were obtained for flow rates at about 0.090 m³/s in water and 0.075 m³/s in air.

These values were much lower than the impeller design flow rate (0.112 m³/s). This difference came essentially from the mismatch between the impeller and diffuser. So that in the present paper, the critical rate of flow is related to the impeller nominal rate of flow and not to the BEP rate of flow.

**VISUALIZATIONS**

**Experimental Devices**

In water tests, as shown on Figure 3, threads were stuck on the wall of the suction pipe immediately upstream of the impeller. Visualizations were also made on one impeller blade leading edge by means of video recording of threads or cavitation figures. Threads were also fastened between the impeller outlet and the diffuser blade inlet diameter on rods (at a diameter of 365 mm). Each rod held two threads, with each one being situated at 5 mm from the front or back wall of the diffuser.

In air, wool threads and a smoke generator were used to visualize flow in the suction pipe in front of the impeller inlet. With both techniques, the probe was introduced across the inlet tube and could be moved radially and axially. At the impeller outlet, only threads were used. A rod on which four threads were fastened (Figure 9) was placed through the diffuser passage inlet.

**Results**

In every case, the reverse flow at the impulse inlet was initiated by large fluctuations of flow direction, followed by a very stable reverse flow (Figure 7).
Results of the traverses in air are given in Figure 10 which shows the position of reverse flow onset points in the inlet pipe for various flow rates lower than the critical one. This map firmly established that reverse flow occurred at the inlet blade tip.

At the impeller outlet, the threads were rather parallel at the nominal conditions. When decreasing the flow rate, the angle of the threads with the tangential direction decreased. Suddenly, thread Number 1 (Figure 9) became more tangential than the others and soon began to oscillate, turning from time to time towards the impeller. This indicated the onset of a reverse flow near the impeller front shroud, at the impeller outlet and/or in the diffuser.

Table 3 summarizes the reverse flow detection in water and air at the impeller inlet and outlet.

<table>
<thead>
<tr>
<th></th>
<th>Q&lt;sub&gt;k&lt;/sub&gt; (inlet)/Q&lt;sub&gt;N&lt;/sub&gt;</th>
<th>Q&lt;sub&gt;k&lt;/sub&gt; (outlet)/Q&lt;sub&gt;N&lt;/sub&gt;</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air tests at 2500 rpm</td>
<td>0.67</td>
<td>from 0.78 to 0.81</td>
</tr>
<tr>
<td>Water tests at 110 rpm</td>
<td>0.66</td>
<td>from 0.68 to 0.72</td>
</tr>
</tbody>
</table>

The critical rate of flow, Q<sub>k</sub>, in the impellers was determined from an estimation of leakage flow coming from the inlet labyrinth. In air tests, a correction was made to take into account the absence of seal A (Figure 5). A labyrinth seal has been made to reduce and control the inlet leakage flow.

CRITICAL FLOW RATE DETECTION FROM PHYSICAL MEASUREMENTS

In cases where visualization was not possible, the critical flow rate could be determined by using measurements of various quantities related to pump performance.

Pump Power Input and Theoretical Head

The critical flow rate could be obtained by using the pump power input curves, according to a method described by Rey, et al. [5]. For each flow rate, the pump power input, corrected with mechanical and disk friction losses, gave the head input, H<sub>i</sub>. Then, a curve could be drawn showing the evolution of H<sub>i</sub> as a function of the impeller flow rate Q. The deviation of this curve from the theoretical head input, H<sub>th</sub>, (without reverse flow) gave an estimation of a critical flow rate (Figures 11a and 11b).
12a). One could also draw the corresponding dissipation power curves as a function of the logarithm of \( Q/Q_N \) (Figures 11b and 12b).

However, this method did not appear as a precise criterion for reverse flow onset detection in this case.

![Figure 12a. Determination of Critical Flow Rate from Theoretical and Actual Impeller Head Input (Air Tests).](image)

![Figure 12b. Determination of Critical Flow Rate from Theoretical and Actual Impeller Power Input (Air Tests).](image)

then be concluded that the relative critical flow in the impeller was higher than 0.58.

Similar analysis for the instantaneous wall pressure is in progress in the water test-rig.

![Figure 13. Wall Static Pressure Variations at Impeller Inlet (Water Tests).](image)

**Instantaneous Static Pressure at Diffuser Inlet (Air Tests)**

A piezo-resistive pressure transducer was put on the front shroud of the diffuser, at about 8 mm from impeller outlet. A frequency analysis of pressure fluctuations was made using a FFT analyser. The evolution of the pressure fluctuation amplitudes, at the impeller blade passing frequency and twice the blade passing frequency when the flow rate was varied (at constant speed of rotation), is shown in Figure 14. At partial flow, for a relative flow rate of about 0.64, the ratio of both amplitudes passed through a sharp maximum, showing a sudden variation of the signal.

**Instantaneous Velocity Measurements**

**Laser Velocimetry in Water Test**

A one component Doppler laser velocimeter was used to

![Figure 14. Pressure Fluctuation Analysis at Impeller Outlet with Bladed Diffuser (Air Tests).](image)
measure the instantaneous velocities. The Doppler signal, detected by retrodiffusion, was treated by a digital counter. Simultaneously, the impeller angular position was measured by an optical encoder delivering 1024 points per revolution. Both the velocity component and the angular position entered the central memory of the microcomputer used for the signal treatment.

The radial and peripheral components of velocity were successively recorded at the impeller outlet, 4 mm downstream from the outlet diameter, in the middle of the diffuser width.

Each graph in Figure 15 gives the results of 2000 measurements superimposed on one impeller revolution. In this representation, the fluctuation from one revolution to the next appears as a vertical scattering. The results for three flow rates, chosen as examples, show clearly the flow disorganization at a critical flow rate of about 0.070 m³/s, which appeared somewhat higher than that visualized at the impeller inlet.

Hot Wire Measurements in Air

In air, velocity measurements at the diffusor inlet (at about 10 mm from the impeller outlet) were made using a two hot wire probe.

Figure 15. Absolute Velocity Profiles at Impeller Outlet from Laser Measurements (Water Tests).

A description of that technique was done by Desmet [6]. Electric signals from both wires were sampled by a numeric memory, transferred into a microcomputer and then translated into velocities (radial and circumferential components). The measurements were made in the middle of the diffusor width, 5 mm from each wall. The evolution of velocity showed that when the flow rate was reduced, the radial component near the inlet shroud occasioned became negative. The velocity distributions (Figure 16) seemed to agree with the beginning of outlet reverse flow on the front shroud.

Figure 16. Velocity Radial Component Cd at Impeller Outlet from Hot Wire Measurements (Air Tests). A = 5 mm from Back Shroud. B = Channel Middle Line. C = 5 mm from Front Shroud.

INFLUENCE OF DIFFUSER DESIGN, LEAKAGE FLOW AND ROTATIONAL SPEED

Rotational Speed

In air, tests were made at three different speeds of rotation (Table 1) with varying Reynolds and Mach numbers. Pump performance changed very slightly. The visualized critical flow rate at the inlet was constant in all conditions, and the scattering was larger when looking at the outlet critical flow rate. The phenomena at outlet were more difficult to analyze with precision. The various results are summarized and presented in Table 4.

Table 4. Values of Critical Flow Rate Determined at Several Rotational Speeds.

<table>
<thead>
<tr>
<th>Speed of rotation (rpm)</th>
<th>Qᵢ (inlet)/Qₙ</th>
<th>Qᵢ (outlet)/Qₙ</th>
</tr>
</thead>
<tbody>
<tr>
<td>2500</td>
<td>0.67</td>
<td>0.78</td>
</tr>
<tr>
<td>2000</td>
<td>0.69</td>
<td>0.81</td>
</tr>
<tr>
<td>1700</td>
<td>0.68</td>
<td>0.78</td>
</tr>
</tbody>
</table>

Table 5: Variations of Leakage Flow.

<table>
<thead>
<tr>
<th>Valves Setting (see Figure 3)</th>
<th>Leakage Flow Direction</th>
</tr>
</thead>
<tbody>
<tr>
<td>V1 closed</td>
<td></td>
</tr>
<tr>
<td>V2 closed</td>
<td></td>
</tr>
<tr>
<td>connected with ambient pressure (P₁ &gt; P ATM)</td>
<td>reverse leakage flow at impeller inlet</td>
</tr>
<tr>
<td>connected with high pressure (P₁ &gt; P ATM)</td>
<td>reverse leakage flow at impeller outlet</td>
</tr>
<tr>
<td>connected with high pressure</td>
<td>reverse leakage flow at impeller inlet and outlet</td>
</tr>
</tbody>
</table>
Leakage Flow

A special design of labyrinths allowing some variations of the leakage flow at water impeller inlet and outlet is shown in Figure 3. Two intermediate annular chambers might be connected to external pressure conditions by means of a controlling valve. The variations obtained are presented in Table 5. The inlet critical flow rate appeared to be modified essentially in the case of an inlet leakage flow reversal, as is shown in Figure 17.

![Diagram of valve states](image)

Figure 17. Influence of Leakage Flow on Reverse Flow Onset at Impeller Inlet (Water Tests).

Diffuser Width

Preliminary measurements in water were made with a wider diffuser (43 mm width instead of 30 mm in all other experiments). Results of these visualizations with threads on the inlet pipe wall are given in Figure 18. The maximum efficiency was naturally obtained for a larger flow rate, and the transition zone corresponding to reverse flow onset was broader than with the smaller diffuser.

![Graph of water test pump performance curves](image)

Figure 18. Water Test Pump Performance Curves (Broad Diffuser).

Table 6: Values of Critical Flow Rate for Bladed and Vaneless Diffusers.

<table>
<thead>
<tr>
<th>Speed of rotation (rpm)</th>
<th>Q_K (inlet)/Q_N</th>
<th>Q_K (outlet)/Q_N</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>6 blade diffuser</td>
<td>vaneless diffuser</td>
</tr>
<tr>
<td>2500</td>
<td>0.67</td>
<td>0.65</td>
</tr>
<tr>
<td>2000</td>
<td>0.69</td>
<td>0.64</td>
</tr>
<tr>
<td>1700</td>
<td>0.68</td>
<td>0.65</td>
</tr>
</tbody>
</table>

Vaneless Diffuser

In the air test-rig, tests were also made with a vaneless diffuser instead of the vaneed diffuser used before. Table 6 gives the comparison of critical flow rates at inlet and outlet with both configurations. The relative critical flow rate appeared to be slightly lower with the vaneless diffuser.

Figure 19 shows the results of pressure fluctuations frequency analysis as already seen in Figure 14. The evolution of the amplitudes of fluctuations was quite different with the vaneless diffuser which shows the difficulty of interpreting such measurements.

![Graph of pressure fluctuation analysis](image)

Figure 19. Pressure Fluctuation Analysis at Impeller Outlet with Vaneless Diffuser (Air Tests).
Table 7: General Comparison of Critical Flow Rates.

<table>
<thead>
<tr>
<th>WATER TESTS</th>
<th>AIR TESTS</th>
<th>FRASER FORMULA</th>
</tr>
</thead>
<tbody>
<tr>
<td>narrow diffuser</td>
<td>bladed diffuser</td>
<td></td>
</tr>
<tr>
<td>standard leakage flow</td>
<td>reversed inlet leakage flow</td>
<td></td>
</tr>
<tr>
<td>1700 rpm</td>
<td>2000 rpm</td>
<td>2500 rpm</td>
</tr>
<tr>
<td>0.66</td>
<td>0.60</td>
<td>0.65</td>
</tr>
<tr>
<td>Qh in</td>
<td>QN</td>
<td>Qh out</td>
</tr>
<tr>
<td>0.72</td>
<td>0.73</td>
<td>0.77</td>
</tr>
</tbody>
</table>

DISCUSSION AND CONCLUSION

The results given in this paper, concerning the value of critical flow rate, are summarized in Table 7. They were completed with an analysis according to the Fraser correlation formula [1]. The principal conclusions which could be drawn from Table 7 are as follows:

The critical flow rate at the pump inlet appeared to be slightly higher in the air tests than in the water tests. This fact could be partly due to small inaccuracies in the flow measurements, but also, to the presence of a larger inlet leakage flow in air tests. Nevertheless, the order of magnitude was quite the same in both series of tests.

In the water tests as well as in air tests, the critical flow rate at the pump outlet was larger than the one found at the pump inlet. This fact was in contradiction with Fraser's formula.

In water tests, the reversal of inlet leakage flow lead to a much lower critical flow rate. On the contrary, a reversed outlet leakage flow rate did not seem to present a marked influence.

In air tests, the use of a vaneless diffuser lead to a critical flow rate slightly lower than with a bladed diffuser.

The influence of the speed of rotation was not well marked in air tests.

Direct determination of the critical flow rate from visualization gave rather well defined values. The use of indirect methods, such as those based on impeller head input measurements (Figures 11a and 12a), on impeller power input (Figures 11b and 12b), on pressure fluctuation analysis (Figures 14 and 19) lead to a much higher uncertainty.

Determination of the critical flow rate at the impeller outlet was more delicate than at the impeller inlet. For that reason, methods more sophisticated than visualization threads should be used. Results from hot wire measurement appeared quite satisfactory (Figure 16) with improvement needed in the laser measurements (Figure 15).

Complementary tests are now going on in order to complete these preliminary results and to check, more particularly, the following points:

- location of stalling region at impeller outlet. The stall was found near front shroud in air impeller but not well defined in water impeller.
- influence of leakage flow rate in air tests.
- influence of volute casing width.

REFERENCES


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