AN EXPERIMENTAL STUDY ON THE UNSTEADY PRESSURE DISTRIBUTION AROUND THE IMPELLER OUTLET OF A CENTRIFUGAL PUMP

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ABSTRACT
An experimental investigation is presented which analyzes the fluctuating pressure field existing in the volute of a centrifugal pump, in order to characterize the effects of blade-tongue interaction. For that purpose, pressure signals were obtained simultaneously at different locations on the volute casing and for different flow-rates by means of fast-response pressure transducers. Particular attention was paid to the pressure fluctuations at the blade passage frequency, regarding both amplitude and phase delay relative to the motion of the blades. The dependence of the obtained pressure fluctuations with respect to both flow-rate and position along the volute clearly indicates the leading role played by the tongue in the impeller-volute interaction and the strong increase of the magnitude of the dynamic forces in off-design conditions.

INTRODUCTION
When operating in off-design conditions, the impellers of centrifugal pumps with volute casings are subjected to some static radial thrust, due to the non-uniform distributions of pressure and moment flux around the impeller (Stepanoff, 1957). These pumps can also present non-steady radial forces, which are mostly associated with the frequency of rotation and the blade passage frequency (and with their harmonics). Excitation at the frequency of rotation may be due to small manufacturing imperfections and mechanical unbalance. Excitation at the blade passage frequency is the consequence of the non-uniform distribution of the flow coming out the impeller from both sides of each blade, due to the differences between pressure and suction sides and the wakes behind the blades. In centrifugal pumps with vane diffusers significant excitation may also exist at the vane passage frequency. Other frequencies can be excited too, with mechanical origin (impeller whirling) or purely hydrodynamic origin (rotating stall).

The magnitude of the resulting dynamic forces is affected by the rotor-stator interaction through the flow, and hence it is dependent on the point of operation of the pump. In the case of excitation at the blade passage frequency, the dominant factor is the interaction between the blades of the impeller and the tongue of the volute. Among others, this blade-tongue interaction was studied by LéZé et al. (1992), who conducted extensive pressure fluctuation measurements to quantify the unsteady pressure field in both impeller (with front shroud removed) and volute of a fan and relate the pressure field to the noise generation mechanisms.

Chu et al. (1995a, 1995b) and Dong et al. (1997) used particle image velocimetry complemented with noise and pressure measurements to measure the velocity distribution and compute the unsteady pressure field in the near-tongue region of the volute of a centrifugal pump. This pump was operating 35% above design flow-rate during tests, and it could be equipped with several different volutes having tongue gaps ranging from 7% to 28% of the impeller radius. They obtained an exhaustive description of the flow, which led them to conclude that primary sources in noise generation are associated with the interaction of the non-uniform outflux from the impeller (jet-wake phenomenon) with the tongue, and that...
small increments in the gap between impeller and tongue (up to gap values of 20% of the impeller radius) cause significant reduction of noise levels.

Morgenroth and Weaver (1998) investigated the transmission to the ducting system of the blade passing frequency fluctuations from a pump, for different rotational speeds and flow-rates, by identifying the acoustic wave modes from the signals of several pressure transducers along the pipeline. They found that when acoustic resonance occurred the magnitude of the pressure was a minimum for the best efficiency flow-rate. They also tested several tongues with different tip radius while keeping a constant gap between impeller and tongue (5.8% of impeller radius), and found that rounding the tongue tip resulted in a reduction of the emitted noise.

Kaupert and Staubli (1999) investigated the unsteady pressure field within a high specific speed centrifugal impeller operating in a double spiral volute, by means of 25 piezoresistive pressure transducers mounted along one single channel and a telemetry system to transport the pressure signals. They measured the pressure pulses induced by both tongues (tongue passing frequency), which propagated upstream along each channel. Their amplitude was particularly high at the trailing edge of the blades, on their pressure sides, and reached 35% of the pump head in off-design conditions.

This paper presents an experimental study on the unsteady pressure field around the outlet of the impeller of an industrial centrifugal pump with volute casing (non-dimensional specific speed \( \omega_s = 0.48 \)). Particular attention has been paid to the dependence of the blade passing frequency excitation on the point of operation of the machine. For this purpose several piezoelectric pressure transducers could be placed at 36 locations around the front side of the volute casing. The pressure signals were FFT processed to analyze the amplitude and phase delay of the fluctuations as a function of position and flow-rate. This study is a continuation of a previous work (Parrondo et al. 1996) on a similar pump with a larger impeller radius instead of 10% in the present study. The results obtained clearly show that hydrodynamic blade-tongue interaction is an important contributor to the unsteady pressure field in the volute.

**NOMENCLATURE**

- \( D_2 \): impeller diameter at outlet (=200 mm)
- \( P_0 \): total pressure at pump inlet
- \( Q, Q_N \): flow-rate, flow-rate at best efficiency point
- \( U_2 \): peripheral velocity at impeller outlet
- \( b_2 \): impeller width at outlet (=16.9 mm)
- \( f_R, f_{BP} \): frequency of rotation, blade passage frequency
- \( p, p_A \): pressure, pressure amplitude
- \( \phi \): flow coefficient = \( Q / (\pi D_2 b_2 U_2) \)
- \( \varphi \): angular position around impeller
- \( \omega_S \): specific speed = \( 2 \pi f_R Q / (\pi H) \)^{1/4} (=0.48)
- \( \psi \): head coefficient = \( gH / (U_2^2 / 2) \)

**EXPERIMENTAL EQUIPMENT**

The pump used for this investigation had single axial suction and spiral volute casing (Figure 1), and it was equipped with an impeller of 200 mm in outer diameter and 7 backward curved blades with logarithmic profile. Other impeller dimensions are: discharge width \( b_2 = 16.9 \) mm, blade angle at outlet =29º, rake angle (pressure side) =10º, rake angle (suction side) =29º, inlet diameter (tip) =52 mm. The gap between the tongue of the volute and the impeller was 10 mm (=10% of impeller radius). The pump was tested in a hydraulic set-up designed according to international standards, in which appropriate piping permitted the water to be pumped from and returned to a reservoir with a capacity of 40 m³. Flow-rate could be finely regulated by means of a set of butterfly valves located close to the reservoir discharge. The flow-rate was simultaneously measured with an ultrasonic rate-meter and with a calibrated orifice plate connected to an inclined piezometric mercury manometer. Measurement uncertainty was estimated to be less than ±3.6% for flow-rate values greater than 20 m³/h. The pump was driven by a DC-motor governed by a regulation device that allowed for continuous adjustment of the rotational speed, with a precision of ±1.0 rpm. Figure 2 shows the head and efficiency performance curves obtained for the pump, which indicate a best efficiency flow coefficient of 0.081. This corresponds to a best efficiency flow-rate \( Q_N = 0.0145 \) m³/s when the pump is run at 1620 rpm, which is the nominal speed used in the tests. At this speed, the static pressure at the pump inlet was always above atmospheric pressure.

**Figure 1:** Sketch of the test pump showing location of the pressure taps.
\[ \psi = \frac{gH}{U^2/2} \]

\[ \eta = \frac{Q}{D b U} \]

\[ \phi = \frac{Q}{nD b U} \]

**Figure 2: Performance curves of the pump (head and efficiency).**

Pressure taps with 3 mm diameter were located every 10° around the front side of the volute, at 2.5 mm from the outlet of the impeller (Figure 1). Static pressure at those positions could be obtained with a Kistler 4043A10 piezo-resistive pressure transducer and a current amplifier, which provided absolute pressure values with an uncertainty of less than ±0.5% (according to manufacturer's data). Also, four Kistler 601A miniature fast-response piezo-electric pressure transducers could be installed on the wall of the volute at any of the 36 tap locations, in order to measure the unsteady pressure. These transducers were mounted inside a special adapter, so that their sensitive membrane was 10 mm far from the internal surface of the volute. Calibration checks indicated that such cavity did not introduce any disturbing effect on the measurements within the range of frequencies of interest in this study.

Each of the transducers was connected to a charge amplifier, which provided pressure measurements with a combined uncertainty of less than ±1.5%, according to manufacturer's data. The resulting pressure signals, as well as the signal from an optical tachometer, could be digitized and stored in a personal computer equipped with a multi-channel digital-to-analog conversion card. Spectral analysis of the signals was then performed by software.

**STATIC PRESSURE DISTRIBUTION**

To begin with, a series of tests was conducted to measure the static pressure along the volute, as a function of the flow-rate, by means of the piezo-resistive pressure transducer. Figure 3 shows the peripheral distribution of the static pressure, relative to the total pressure \( P_0 \) at the pump inlet, for several values of the flow-rate; angular position \( \varphi \) is zero at the edge of the tongue and it increases with anti-clockwise orientation (see Figure 1). In agreement with the trends indicated in classic texts (e.g. Stepanoff, 1957), the static pressure around the volute is quite uniform for flow-rates in the range of the best efficiency point, but it exhibits either a maximum for low flow-rates or a minimum for high flow-rates in the region \( \varphi = 300°-330° \). These non-uniform pressure distributions are related to the acceleration (high flow-rates) or deceleration (low flow-rates) of the flow along the volute for off-design conditions. The non-uniformity of the pressure in the volute is seen from the flow in the rotating impeller as an unsteady boundary condition, and thus it affects the resulting fluctuations in the pressure field.

**UNSTEADY PRESSURE MEASUREMENTS**

For the fluctuating pressure measurements, each of the four piezoelectric pressure transducers was mounted at some given position around the volute. During each test the flow-rate was progressively increased from 0 to 1.6 \( Q_N \), with steps of 0.1 \( Q_N \), where \( Q_N \) (best efficiency flow-rate) is equal to 52.2 m³/h for the nominal rotational speed of 1620 rpm. For each flow-rate, 50 samples of 1 second long and a digitizing frequency of 1024 Hz were recorded in the computer simultaneously for the four pressure signals and the tachometer signal. Check tests proved that this procedure was sufficient to obtain consistent and repeatable results. After the tests, each recorded sample was FFT processed (a Hanning window was used) and then the frequency averaged power spectrum and cross-power spectrum between signals were finally obtained. The power spectrum provides a direct measure of the amplitude of the fluctuations at each frequency, whereas the phase of the cross-power spectrum provides a measure of the phase delay between the fluctuations of two signals, if acquired simultaneously.
0.0006 (non-dimensional value) for each of the 17 flow-rates tested.

Figure 4 shows the evolution with respect to the flow-rate of the power spectrum of the pressure at the measurement positions $\varphi = -10^\circ$ and $\varphi = 20^\circ$ (the reference position $R$). As expected, the predominant spikes correspond to the frequency of rotation of the impeller (27 Hz) and in particular to the blade passage frequency (189 Hz), together with their respective harmonics. Though both measurement positions are quite close from each other, the trends shown in the spectra are very different, because each position is at one different side of the tongue edge. This is further discussed below for the pressure fluctuations at the blade passage frequency.

Figure 5: Pressure magnitude spectrum as a function of flow-rate for positions $\varphi = -10^\circ$ and $\varphi = 20^\circ$ (position $R$ in Figure 1).

Another series of tests was conducted to check the existence or not of acoustic resonance phenomena in the piping (see Morgenroth and Weaver, 1998). For such purpose the pump was run at several speeds ranging from 1020 to 2520 rpm, while monitoring the pressure fluctuations at the reference position $R$ and at the inlet and outlet of the pump. The results obtained for position $R$, at the corresponding blade passage frequency, are shown in Figure 5 as a function of the relative flow-rate. All the curves are seen to converge in a rather narrow band, as expected from similarity considerations, except the one associated to 2040 rpm for the low range of flow-rates. This particular behavior was identified to result from an acoustic resonance in the suction line. Interestingly, after some seek the phenomenon was found to disappear when slightly opening a butterfly valve located in a branch of the suction pipe, though that branch leaded to a dead end and, hence, in no case there was any flowing water through it. These tests permitted to ensure that no special acoustic phenomena in the piping affects the pressure measurements now reported (obtained with a speed of 1620 rpm.).

RESULTS AT THE BLADE PASSAGE FREQUENCY

For each of the 36 measurement positions the amplitude and the phase of the pressure fluctuations at the blade passage frequency was obtained as a function of the flow-rate, in a fashion similar to the one of Figure 5 for the reference position $R$ ($\varphi = 20^\circ$). Figure 5 shows that the pressure amplitude in the tongue region is a minimum for the best efficiency flow-rate range, and that it increases fast for both lower and higher flow-rates. This is a foreseeable result since only for the best efficiency flow-rate there is a good matching between the flow coming out the impeller and the flow in the volute, with nearly no re-circulation through the gap between impeller and tongue. In off-design conditions however the absolute velocity at the
impeller outlet forms a great incidence angle with respect to the mean flow in the volute (behind the tongue region), which leads to big flow disturbances accompanying each blade passage.

Figure 6 shows the zero-to-peak amplitude of the pressure fluctuations as a function of the flow-rate and the angular position \( \varphi \) around the volute. The map of Figure 6 has been obtained from the evolutions of the pressure amplitude with respect to flow-rate for each of the 36 measurement points. For each flow-rate, the behavior observed is very different from that of the stationary pressure distribution (Figure 3). As stated above, in general the pressure fluctuations are comparatively small in the best efficiency region. The maximum amplitudes, which correspond to the region just behind the tongue (measurement positions from \( \varphi=10^\circ \) to \( 50^\circ \)) for very low and high flow-rates, reach a non-dimensional value of 0.05 (about 6% of the static pressure increment though the pump for high flow-rates). These fluctuation values are about 50% greater than the fluctuations measured by Parrondo et al. (1996) for the same volute and a slightly smaller impeller (190 mm), which corresponds to a tongue-impeller gap of 15.8% of impeller radius instead of the present 10%.

![Figure 6. Amplitude (zero-to-peak) of pressure fluctuation (blade passage frequency) along the volute as a function of flow-rate.](image)

For each flow-rate, Figure 6 permits to observe a modulated pattern along the volute, with node and anti-node positions, somewhat variable with the flow-rate, which is a typical effect of the superposition of correlated waves. The distance between nodes is approximately equal to the distance between consecutive blades (=360\(^\circ\)/7).

Figure 7 shows the time history of the instantaneous pressure (only the blade passage frequency component) around the impeller, during the period between the passage of two consecutive blades in front of the tongue. These results are presented for the flow-rates 20%, 60%, 100%, 130% and 160% of the best-efficiency flow-rate \( Q_N \). Bold arrows indicate the angular position of the seven blades (pressure side edge on hub shroud) along time. For \( Q/Q_N=100\% \) the pressure fluctuations are observed to behave similarly along the whole volute; they are synchronized with the passage of each blade in front of each measurement position, and so the peak values of the fluctuations occur at different instants for each position. These perturbations result from the non-uniform distribution of the flow coming out the impeller around each blade, and their effects are only local.

For off-design conditions, however, the region of the volute close to the tongue presents a simultaneous evolution of the pressure fluctuations, with big amplitudes in the whole region. They are the result of the strong interaction between the blades and the tongue, which produces acoustic pressure waves capable of propagating through the pump towards the inlet and outlet pipelines. Along the volute, this acoustic pressure is progressively reduced due to divergence and absorption. Chu et al. (1995b), who studied a pump operating at 135% of the best efficiency flow-rate, associated the noise production to the impingement on the tongue of the wake behind the passing blades and the associated trains of vortices. Comparing the graphs of Figure 7 for flow-rates above and below \( Q_N \), the fluctuations in the region close to the tongue are seen to be shifted 180\(^\circ\) from each other. This is related to the shift of the stagnation point on the tongue and the different characteristics of the leakage flow through the gap. For low flow-rates the alignment of the blades with the tongue coincides with a positive value of the pressure, whereas it coincides with a negative value for the high flow-rates (in agreement with the results of Chu et al. 1995b).

In summary, the fluctuating pressure field in the volute at the blade passage frequency may be interpreted to result from the disturbances associated to the passage of each blade in front each point of the volute and to the passage of each blade in front of the tongue. The effects of the blade-tongue interaction are particularly important for off-design conditions. The positive or negative combination of both groups of disturbances leads to the modulated pattern in the pressure amplitude suggested by Figure 6.

**CONCLUSIONS**

A systematic series of tests were conducted to measure the dynamic pressure in a number of positions around the single volute of a centrifugal pump, as a function of the flow-rate. The analysis focused on the pressure amplitude and phase delay at the blade passage frequency. The pressure fluctuations registered around the volute were found to be very dependent on the flow-rate. In general the amplitude of the fluctuations is greater in off-design conditions, in particular in the tongue region, which suggests the monitoring of the pressure in that zone as a potential means of observing the operation of the pump. Comparison of the present data with the previous results for a pump with the same volute and a smaller impeller shown that reducing the tongue gap from 15.8% to 10% of the impeller radius leads to a increase in the maximum pressure amplitudes of about 50%.

For a given flow-rate, the data obtained indicate that the pressure fluctuations at any point in the volute result from the
superposition of the perturbations induced by: i) the passage of each blade in front of that point; and ii) the passage of each blade in front of the tongue. This blade-tongue interaction appears to be dominant in the generation of the dynamic pressure field in the volute particularly for the lower range of flow-rate, and is responsible for the generation of noise.

**ACKNOWLEDGMENTS**

The authors gratefully acknowledge the financial support of the Comisión Interministerial de Ciencia y Tecnología (Spain) under Projects TAP-96-1199 and TAP-99-0738-C02-02.

**REFERENCES**


**Figure 7.** Time history of the pressure distribution along the volute (blade passage frequency component) between the passage of two blades, for several flow-rates.