The Unsteady Pressure Field in a High Specific Speed Centrifugal Pump Impeller—Part I: Influence of the Volute

An experimental investigation is presented regarding the unsteady pressure field within a high specific speed centrifugal pump impeller \((\alpha_0 = 1.7)\) which operated in a double spiral volute. For this, twenty-five piezoresistive pressure transducers were mounted within a single blade passage and sampled in the rotating impeller frame with a telemetry system. The influence of varying volume flux on the pressure transducers was evaluated in terms of pressure fluctuation magnitudes and phase differences. The magnitude information reveals that the pressure fluctuations from the impeller-volute interaction grew as the volume flux became further removed from the best efficiency point and as the trailing edge of the impeller blade was approached. These fluctuations reached 35\% of the pump head in deep part load. The upstream influence of the volute steady pressure field dominates the unsteady pressure field within the impeller at all off design load points. Acquired signal phase information permits the identification of the pressure field unsteadiness within the impeller passage as fundamentally synchronized simultaneously with the volute tongue passing frequency. Special emphasis was placed on the volume flux regime where the pump and impeller pressure discharge characteristic undergo hysteresis, as impeller inlet and outlet recirculation commence and cease. A synthesis of the rotating transducers was performed to obtain unsteady blade loading parameters. The value of the unsteady lift coefficient varies on the order of 200\% for a single blade in part load operation (at 45\% bep), an abrupt fluctuation occurring as the fore running blade suction side passes a volute tongue. The unsteady moment coefficient and center of pressure are also shown to vary significantly during the impeller-volute tongue interaction.

Introduction

Numerical prediction of the pump flow field reduces industries time and construction cost in the design or modification of pump components. Much effort has been spent predicting time averaged flow quantities, obtaining limited success near pump best point efficiency operation (e.g., Staubli et al., 1995). These quantities are useful for pump performance predictions even though agreement in part load is generally poor.

Little success has been reported in the numerical prediction of centrifugal pump unsteady flow fields and associated unsteady forces. This is due greatly to a limited existence of extensive experimental data to perform any validation (Brennen, 1994) and the numerical complexity involved. With pump construction trends moving towards larger output power concentrations (Makay, 1988) and higher heads per stage (Tamatsukuri, 1992), larger unsteady flow field fluctuations within the pump and system are encountered. Experimental validation of numerical simulations and other prediction methods involving unsteady flow fields becomes increasingly necessary.

Test Facility and Instrumentation

Under investigation was a single stage pump designed for the paper industry to transport slurry inhomogeneous substances. The impeller of outlet diameter \(D_1 = 324\) mm operated within a double spiral volute to minimize radial forces. The shrouded pump impeller is shown in Fig. 1. Geometric specifications were:

- 7 blades, shrouded impeller,
- \(D_1/D_2 = 0.83\) impeller inlet tip diameter ratio,
- \(B_1/D_1 = 0.27\) blade outlet breadth ratio,
- \(D_1/D_2 = 1.22\) volute tongues inlet diameter ratio,
- 33\° blade outlet back lean angle, 20\° blade outlet rake,
- 18\°–40\° blade inlet angle in relative system.

The experiments were performed at a rotational speed of 750 rpm \((\alpha_2 = 12.7\) m/s\), having a best efficiency point (bep) volume flux of 0.196 m\(^3\)/s \((\phi = 0.174)\) and a pump pressure rise of 0.58 bar \((\phi = 0.704)\).

Water at 18\°C for the open pump circuit was drawn from an 80 m\(^3\) reservoir into the pump through a flow straightener to provide uniform inlet flow. The impeller Reynolds number at bep was of the order 10\(^6\) based on the impeller diameter and outlet tip velocity. The measurement locations in the test facility were constructed in accordance to international acceptance norms (ISO, 1987). A booster pump well upstream increased the NPSH to avoid cavitation.

The double spiral volute casing was circumferentially fitted with 32 flush mounted pressure taps, 16 on the casing shroud side and 16 on the casing hub side, near the impeller outlet at \(r/R_2 = 1.05\). The resulting measured pressures were steady quantities in the stationary frame and will be presented here in nondimensional form using a dynamic pressure based on the impeller outlet tip velocity to form the steady pressure coefficient:

\[
C_p = \frac{P_i - P_o}{0.5 \rho u_2^2}
\]
which can be interpreted as a local \( \psi \) for the \( i \)th pressure tap since an upstream pressure difference is built. In deep part load (\( \phi < 0.06 \) or 40% bep) this coefficient was affected by the flow pre-rotation influencing \( P_1 \) directly.

On the impeller 25 piezoresistive pressure transducers were mounted within a single blade passage, Fig. 2. Their location was selected to follow the path of two “wall streamlines” on the blade passage pressure side, Pressure Side Hub (PSH) and Pressure Side Shroud (PSS), and two wall streamlines on the blade passage suction side, Suction Side Hub (SSH) and Suction Side Shroud (SSS). Each transducer had a diameter of 4.5 mm with a pressure sensitive zone of 1.3 mm diameter. The eigenfrequency of the transducers was known to be near 100 kHz in air, sufficiently high that the frequencies of interest in water, a maximum of 1 kHz, will not be detrimentally influenced. A static calibration of all transducers was performed to verify manufacturers specification. Linearity was within \( \pm 0.2\% \) over the full scale of 0 to 5 bar absolute. A dynamic calibration was deemed unnecessary because of the high transducer eigenfrequency and flush mounting of the transducers. The errors in transducer measurements were quantified as \( \pm 5\% \) for the steady value, due to the influence of sensor material creep, and \( \pm 1\% \) for the unsteady value (Gossweiler, 1993). To reduce the transfer of any mechanical stresses from blade vibration and centrifugal forces the transducers were mounted with an elastic silicon epoxy which received detailed attention before any mounting proceeded (Kaupert, 1997).

A telemetry device was mounted on the pump shaft to send the acquired pressure transducer signals to the stationary system as a high band FM signal, Fig. 3. This permits a wireless connection between the rotating and stationary frame. Since 25 pressure transducers existed each operating point was sampled twice with 16 transducers connected. Repeatability was confirmed with 7 channels measured redundantly. These measurements are unsteady quantities in the impeller frame and will be presented here in nondimensional form as the unsteady pressure coefficient,

\[
\tilde{C}_p = \frac{\tilde{P}_i}{0.5 \rho u_2^2},
\]

and the combined pressure coefficient,

\[
C_p = \frac{\tilde{P}_I + \tilde{P}_T}{0.5 \rho u_2^2},
\]

which make no reference to the upstream inlet pressure.

### Nomenclature

- \( A \) = blade area
- \( C_p \) = pressure coefficient
- \( L \) = lift in circumferential direction
- \( m \) = distance along a wall streamline
- \( Mo \) = moment about the origin
- \( n \) = pump shaft harmonic
- \( P_i \) = absolute pressure upstream of the pump
- \( P_1 \) = absolute pressure at \( i \)th pressure tap
- \( r \) = radius from impeller origin
- \( r_s \) = center of pressure with respect to impeller origin
- \( u_2 \) = impeller outlet tip velocity
- \( \theta \) = transducer meridian plane angle
- \( \phi \) = pump shaft angle
- \( \sigma \) = statistical standard deviation
- \( \psi \) = flow coefficient
- \( \omega \) = pump shaft angular frequency
- \( \omega_s \) = dimensionless specific speed at bep
- \( \delta \) = transducer meridian plane angle
- \( \psi = \frac{\Delta P}{u_2/(0.5 \rho u_2^2)} \)
- \( \omega \) = pump shaft angular frequency

### Subscripts

- \( i \) = \( i \)th pressure location
- \( 1 \) = upstream pump inlet
- \( 2 \) = impeller outlet
- \( \tilde{\cdot} \) = unsteady quantities
- \( \cdot \) = steady quantities

### Acronyms

- bep = best efficiency point
- PSH = Pressure Side Hub distance for transducers
- PSS = Pressure Side Shroud distance for transducers
- SSH = Suction Side Shroud distance for transducers
- SSS = Suction Side Shroud distance for transducers

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All signal analysis was performed for a sampling set of $2^{17}$ points per channel sampled at near 6 kHz, slightly more than 270 rotations. The signals were phase averaged using a single shaft position provided by the shaft position encoder. This provided a start point for all analysis to allow phase velocity calculations between transducers.

The pressure discharge characteristic for the pump and the impeller (Kaupert, 1996) is shown in Fig. 4 with an accompanying calculated volute head line based on estimation of the angular momentum in the volute (Loretta and Gopalakrishnan, 1986). This calculation uses the volute throat area, the assumption of free vortex flow in the volute, and the continuity equation to determine $c_{v2}$. Also shown is a zoom of the characteristic in which a hysteresis loop was found to occur.

**Impeller Outlet Flow Distortion**

The pump impeller and the double spiral volute exchange fluid to form a matching of the angular momentum exchange, Fig. 4, to determine a best efficiency point $\phi_{bep}$. Any mismatch in this angular momentum exchange causes the flow in the volute to either be accelerated for $\phi > \phi_{bep}$ or decelerated for $\phi < \phi_{bep}$. This simplistic approach can be used to interpret the circumferential pressure distribution measurement of the volute wall pressure taps at $r/R_2 = 1.05$ for four volume fluxes, Fig. 5. At operating points $\phi < \phi_{bep}$, the value of $\bar{C}_p$ can be seen to rise in the direction of impeller blade rotation with the two volute tongues acting as boundaries to separate the volute flow into two distinct halves. The volute flow was decelerated as the simplistic model predicts. At $\phi > \phi_{bep}$, the variation in $\bar{C}_p$ was smaller while at $\phi = \phi_{bep}$ a reverse tendency to part load was exhibited in the distribution. The volute flow was accelerated as the simplistic model predicts. Figure 5 is in accordance with previously reported experimental results (Adkins and Brennen, 1988, Wesche, 1987) for single spiral volutes. The dotted lines represent the average value for both the hub and shroud side measurements.

**Volute Influence on the Impeller Pressure Field**

Interpretation of the unsteady pressure field within the impeller must be made keeping the results of Fig. 5 in mind. They represent the average volute pressure a blade passage interacts with as it passes a particular point in the volute. The stationary frame pressure distribution is experienced as an unsteady pressure distribution in the impeller frame. Missing from the measurements of Fig. 5 is the unsteady pressure field in the volute caused by the impeller rotation (i.e., jet-wake flow), this however is mostly steady in the impeller frame.

Figure 6 reveals the phase averaged unsteady pressure for two transducers at 4 operating points over two rotations. The small static pressure head variation per rotation due to the horizontal alignment of the pump was subtracted. A tongue passing frequency is evident at a periodicity of twice per rotation. At bep load the pressure fluctuation in the impeller was small since the volute pressure distribution was uniform. At part load the volute steady
pressure distribution \( C_p \) rises between tongues; this manifests itself in the impeller as a rising pressure fluctuation after the blade passage rotated past a tongue (the arrows show radial alignment between the SSH trailing edge and the tongues). At overload this tendency was reversed as the volute flow had a decreasing pressure between volute tongues, seen in Fig. 5, and thus a decreasing pressure fluctuation in the impeller. The blade passing frequency is not clearly evident but does weakly exist as a result of other blades rotating past the tongue. Of interest is the size of the pressure fluctuation, growing with reduced volume flux in part load to obtain magnitudes of 35% the pump head at 40% bep flow. Other authors (Arndt, 1988) have obtained similar results with even greater fluctuation magnitudes being measured directly at the blade trailing edge.

It has previously been pointed out (Tourret et al., 1989) that pressure fluctuations within a pump impeller grow in magnitude as the volume flux is further removed from the bep and as the trailing edge \( s/L = 1 \) of a blade is approached. This was also found in this study as shown in Fig. 7 where the \( C_p \) is 2 standard deviations of a pressure signal magnitude. At \( \phi = 0.174 \) (100%) on all four “wall streamlines” the magnitude was a minimum, expected from Fig. 5 where the volute flow circumferential pressure was most uniform. As the volume flux was moved from the best point to the regime where pump impeller characteristic hysteresis was known to occur \( \phi = 0.122 \) (70%) the fluctuation magnitudes rose. This regime was measured with particularly good resolution in \( f = 0.174 \) (100%) on all four pressure transducer locations for supporting this statement, seen in Fig. 8. The harmonic magnitudes from a Fourier analysis are shown where \( n = 2 \) is the tongue passing harmonic. It can be seen that \( n = 2 \) was the dominant harmonic at the four pressure transducer positions, a factor of near five greater than the shaft harmonic \( (n = 1) \) and the blade passing harmonic \( (n = 7) \). This is typical of all pressure transducer positions in the blade channel excluding a few positions near the impeller inlet where the shaft harmonic \( (n = 1) \) can reach the same magnitude as the tongue passing harmonic \( (n = 2) \) but both are then, as seen in Fig. 7, relatively small. This suggests presentation and interpretation of the phase information at the tongue passing harmonic.

The phase information along the 4 wall streamlines are all relative to a single pump shaft position meaning individual phase differences between pressure transducers may directly be interpreted as time lags or leads. Figure 9 reveals the processed phase information from an FFT for the tongue passing harmonic along the four wall streamlines within the impeller blade passage for 13 volume fluxes. The zero phase position occurs when the suction side of the blade passage at the hub trailing edge was radially aligned with the top tongue.

For the two cases \( \phi = \phi_{\text{bep}} \) (100% and 120%) a different tendency is seen than for \( \phi < \phi_{\text{bep}} \) (80% to 10%) however for the two \( \phi = \phi_{\text{bep}} \) (100% and 120%) cases the pressure fluctuation magnitudes were relatively small, Fig. 7, and thus will not be interpreted. For \( \phi < \phi_{\text{bep}} \) (80% to 10%) on the blade passage pressure side hub and pressure side shroud (PSH & PSS), the first three pressure transducers in the blade passage have little phase change between them, i.e., slope between them very close to zero.
This is an acoustic effect in the blade passage which influences the pressure transducers simultaneously. The acoustic propagation velocity within the test facility was previously measured as 1362 ± 20 m/s in water (note: the water was not deaerated). Both the pump shaft harmonic (n = 1) and twice the tongue harmonic (n = 4) have also been analyzed and revealed similar, not shown here, phase relations meaning the wave group and phase velocity are both equal to the acoustic velocity. The coherence between transducers was also high (>0.92) indicating little disturbance in the form of noise and nonlinearities (i.e., the system is nondispersive). Near the trailing edge (n=PSH = 1, n=PSS = 1) however, a negative slope occurs in the phase because the trailing edge pressure side transducers arrive at the tongue later due to impeller blade curvature; they experience the change in the pressure field across the tongue at a later time. Along the suction side hub and suction side shroud (SSH and SSS) wall streamlines the phase difference between the pressure transducers in part load was near zero excluding the leading edge. The magnitude of the pressure fluctuation at the leading edge was relatively small and also strongly affected by inlet recirculation with associated prerotation phenomena making interpretation here too abstract. As the entire blade passage suction side hub was aligned with a tongue. Examining the interaction with bottom tongue at:

- t = 30 ms, the suction side of the blade passage approached the bottom tongue, influenced by pressure in region High.
- t = 34 ms, the blade suction side hub passed the bottom tongue, came under the influence of region Low, the entire suction side was immediately influenced by the abrupt change in pressure at the blade passage outlet and reacts with a change in circulation about the blade instantaneously. The blade passage pressure side, however, was now under the influence of the pressure in region High and Low. The first three pressure transducers (18, 19, 22) near the pressure side inlet react to Low while last two transducers (15, 23) near the trailing edge were still, due to blade curvature, in High's influence.
- t = 45 ms, as the blade passage continued to rotate the last two transducers (15, 23) moved past the tongue coming into Low's influence. This blade curvature effect is removed from the phase information of the last 2 pressure side transducers shown in Fig. 9 labelled “curvature correction” to demonstrate the geometrical phase lags.

This simplistic interpretation provides an understanding of the impeller-volute interaction physics represented in the pressure signal phase evaluations and an appreciation for the unsteady flow field during a blade passage—volute tongue interaction.

**Synthesis of the Impeller Pressure Field**

The circumferentially varying pressure field within the volute caused an unsteady pressure field within the impeller blade passage and was the dominant cause of unsteady blade loading. To quantify this unsteady blade loading coefficients were evaluated where the lifting line of action was taken as the circumferential direction of rotation, Fig. 2. This evaluation involves a summation over the pressure and suction side using Haar function interpolation for each pressure transducer acting over a given surface. Errors were introduced into the evaluation since—

- the surface resolution (i.e., only 25 pressure transducers exist within the blade passage), did not capture any local pressure gradient effects,
- extrapolation was used between transducers and to the wall regions,
- phase averaged pressures were used,
- a single blade passage, not a single blade was investigated. A shift of the blade passage suction side data by one blade passage backward was performed to remedy this for single blade loading.

The lift coefficient ($C_L$) in the circumferential direction, the moment coefficient ($C_{Mo}$), and the resulting center of pressure ($r_{cp}$) with respect to the impeller rotational axis, depicted in Fig. 2, were evaluated according to,

$$C_L(\phi, \theta) = \sum_{i=1}^{25} C_L(\phi, \theta) A_i \sin \theta \cos \delta / (\sum_{i=1}^{25} A_i)$$

$$C_{Mo}(\phi, \theta) = \sum_{i=1}^{25} C_{Mo}(\phi, \theta) A_i r_i \sin \theta \cos \delta / (R_2 \sum_{i=1}^{25} A_i)$$

$$r_{cp} / R_2 = C_{Mo}(\phi, \theta) / C_L(\phi, \theta)$$

where $A_i, \theta$, and $\delta$ are geometrical parameters. The blade outlet radius $R_2$ was selected as the characteristic length.

The unsteady value $\tilde{C}_L(\phi, \theta)$ is shown nondimensionalised in Fig. 11 using the steady $C_L(\phi)$ at a particular $\phi$ over one impeller rotation. Errors were estimated to be near ±15%. Values in the plot are shown for an impeller angle $\theta$ corresponding to the
alignment of the blade suction side at the hub with the volute tongue. At bep \( \phi = 0.174 \) (100\%) the fluctuation in \( \bar{C}_l(\phi, \theta) \) is seen to be small as the volute’s circumferential pressure distribution was least distorted, shown in Fig. 5. For lower \( \phi \) the volute’s circumferential pressure distribution becomes increasingly distorted as in Fig. 5, having a direct influence on the \( \bar{C}_l(\phi, \theta) \). Between tongues the \( \bar{C}_l(\phi, \theta) \) rose as the pressure in the volute did, reaching a maximum as the suction side of the previous blade was aligned with the volute tongue. Hereafter the blade rotated into the region of influence from the lower pressure region near the tongue, as in Fig. 10, and abruptly experienced the change in the pressure field. Deep in part load the pressure field fluctuations grew to a magnitude of 35% of the pump head. The pressure fluctuation at the impeller outlet propagated upstream through the blade passage at acoustic velocity excluding those locations where blade curvature and rotation provided a phase lag.

The determined unsteady blade loading coefficients provide an indicator of the strong unsteady flow field an impeller blade experiences and reveals the necessity to include these effects in any blade loading considerations.

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References


