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EFFECT OF MODIFICATION TO TONGUE AND IMPELLER GEOMETRY ON UNSTEADY FLOW, PRESSURE FLUCTUATIONS AND NOISE IN A CENTRIFUGAL PUMP

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ABSTRACT

Particle Image Velocimetry (PIV), pressure and noise measurements are used to study the effect of modifications to tongue and impeller geometries on the flow structure and resulting noise in a centrifugal pump. It is demonstrated that the primary sources of noise are associated with interactions of the non-uniform outflux from the impeller (jet/wake phenomenon) with the tongue. Consequently, significant reduction of noise is achieved by increasing the gap between the tongue and the impeller up to about 20% of the impeller radius. Further increase in the gap affects the performance adversely with minimal impact on the noise level. When the gap is narrow, the primary sources of noise are impingement of the wake on the tip of the tongue, and tongue oscillations when the pressure difference across it is high. At about 20% gap, the entire wake and its associated vorticity trains miss the tongue, and the only (quite weak) effect of nonuniform outflux is the impingement of the jet on the tongue. An attempt is also made to reduce the non-uniformity in outflux from the impeller by inserting short vanes between the blades. They cause reduction in the size of the original wakes, but generate an additional jet/wake phenomenon of their own. Both wakes are weak to a level that their impacts on local pressure fluctuations and noise are insignificant. The only remaining major contributor to noise is tongue oscillations. This effect is shown to be dependent on the stiffness of the tongue.

INTRODUCTION AND RATIONALE

Experiments discussed in Chu et al. (1993, 1995 a, b) demonstrated that interactions of non-uniform outflux from the impeller with the tongue can be primary contributors to noise generation within the volute of a centrifugal pump. That study included PIV measurements in sufficient spatial and temporal resolutions, that enabled computations of the phase averaged, unsteady pressure from the velocity distributions (by integrating the Reynolds Equations). The results pointed directly at specific sources of noise. Included were effects of the jet/wake phenomenon (introduced and discussed by Dean and Senoo, 1960 and Eckardt, 1975), particularly the wake, and tongue oscillations due to high pressure differences across it. The present paper continues this effort by focusing on attempts to reduce noise either by modifying the tongue geometry or by reducing the non-uniform outflux from the impeller.

LIST OF SYMBOLS

- B_2 - Impeller width
- p - pressure
- Q - overall flow rate
- Q_D - design flow rate (200 gpm)
- r - radial distance from the center of the impeller
- r_T - impeller radius
- r_v - outer radius of the volute
- T - blade passage period
- t - time
- U_T - impeller tip speed
- v_r , v_θ - absolute radial and circumferential velocities
- ξ_z - axial vorticity component
- θ - angular orientation
- D_2 - impeller diameter

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One of the conclusions of prior studies with the same pump (Dong et al., 1992a, b; Chu et al. 1995a, b) was that most unsteady phenomena occur at $r/r_T<1.2$, in agreement with measurements performed in a different pump by Hamkins and Flack (1987). Thus, an obvious solution to the noise problem is to increase the gap between the tongue and the impeller, thus keeping the tongue out of the region with severe non-uniform flows. Although it is known that the performance of a pump depends on this gap (Hira and Vasandani, 1975), little or next to nothing has been reported about the effect of this gap on the flow structure, and in turn on the mechanisms of noise generation. These issues are some of the objectives of the present paper. It includes measurements of noise, pressure, velocity and vorticity distributions associated with several tongues, ranging in gaps between 7% to 27% of impeller radius. These experiments demonstrate how and why even slight increases in the gap alter the flow structure, decrease the noise level and modify the associated spectrum. The effect of tongue stiffness is also discussed.

Another approach to noise reduction involves tackling the non-uniform outflux from the impeller. Using the original velocity distribution an attempt is made to reduce the "wake" (region of high circumferential velocity, v_θ, at the exit from the impeller) by inserting short vanes inside the impeller. Unlike typical attempts to install symmetric vanes at the center of a blade passage (Cumpsty, 1989), the location and orientation of the present vanes are selected based on prior data on the flow. As will be shown shortly, this attempt results in partial success. The overall noise level, particularly the noise associated with the wake is reduced. However, the new vanes cause additional unsteady phenomena.

SETUP AND PROCEDURES

The test model was a vertical centrifugal pump operating at 890 rpm. The impeller was 9.95 inches in diameter, and had 7 blades, all of them with an exit angle of 20°. The configuration of the perimeter (in inches) was $r_v=r_T+0.34+4(\theta/360°)$ where $\theta=0°$ was located 3° upstream of the tip of the tongue. The volute had constant thickness (0.975") which was equal to the thickness of the impeller . The 4" x 0.975" exit gradually expanded to a 4" pipe by using a fiberglass diffuser with maximum expansion angle of 7°. The design flux of this volute was 200 gpm, but the maximum efficiency occurred at a slightly lower flux (about 190 gpm). Measurements were performed 35% above design flow rate in order to highlight nonuniformities and unsteady phenomena.

Velocity measurements were performed by utilizing PIV (Figure 1), following procedures described in detail by Dong et al. (1992a, b). Briefly, this method consists of recording multiple exposure images of a flow field illuminated by a 1mm thick laser sheet and seeded with microscopic neutrally buoyant particles. The sheet can illuminate any desired section of the volute through the transparent perimeter. Illumination from the bottom in order to study normal planes is also possible. Three 20 nsec laser pulses, generated by a frequency doubled Nd-Yag laser, with a typical delay between pulses of 160 μsec, were used for recording a single image. The timing was synchronized with the orientation of the impeller using an encoder attached to the shaft of the pump. The images were recorded by a 35 mm camera and digitized to a 3000x2000 pixels array using a slide scanner. The analysis consisted of dividing the image into a large number of small windows (each consisting of 64x64 pixels, that correspond to 0.1"x0.1" in the volute), and determining the mean shift of all the particles within each window by computing the auto-correlation function of the intensity distribution. Calibration experiments (Dong et al. 1992a) have shown that the uncertainty level can be kept at about 1%, provided certain conditions associated with particle density (at least eight particles per window) and magnification (typical displacement between exposures of about 20 pixels) are satisfied. The regions sampled during the present study are outlined in Figure 3. The data for tongues 2 and 3 were recorded at the center plane of the volute every 10° of impeller orientation, namely five per blade passage. While computing the pressure within the original pump (using the PIV results), the data was recorded every 1° (Chu et al., 1995). Typically 8-12 images were necessary to obtain phase averaged distributions. This choice was based on a requirement that the variations caused by adding another distribution should be less than the error level. Vorticity ($\xi_z=\partial v/\partial x-\partial u/\partial y$) was determined by computing $\Delta v/\Delta x-\Delta u/\Delta y$.

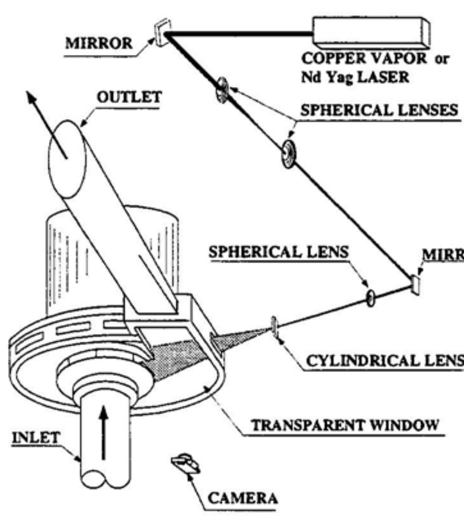
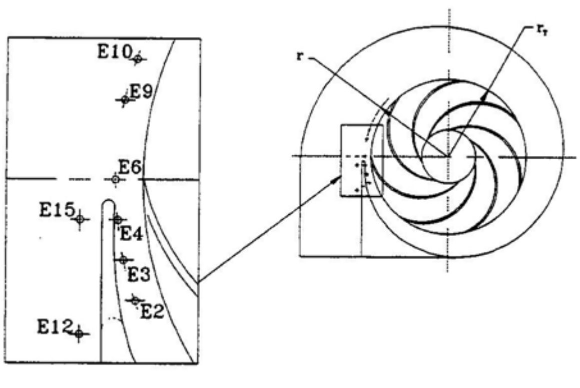


Fig. 1 A schematic description of the optical setup in the pump facility.

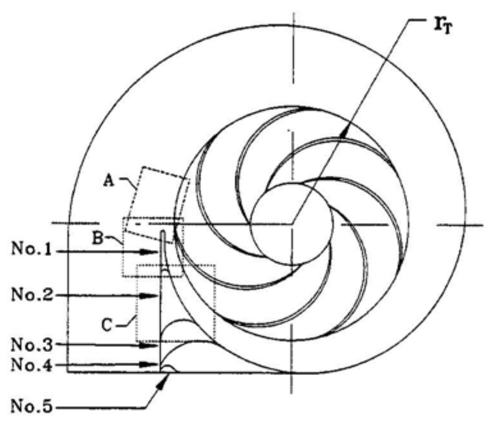
Pressure fluctuation measurements were performed on the upper (hub side) surface, mostly with flush-mounted, 0.1" diameter, piezo-electric transducers (PCB model 105B02). The noise was measured with flush mounted hydrophones (PCB model 106M12) located 3.3 ft upstream and downstream of the pump. The signal was amplified and recorded by a PC based data acquisition system. FFT was used for computing the spectra.

Mean pressure measurements were performed using 0.0197" pressure taps. The transducers were regularly calibrated, and the error level of the absolute pressure (mean + fluctuations) should be about 1%. In order to perform both measurements at the same point, part of the upper surface was replaced with an insert that could accommodate either the transducer or the taps. The sites of measurements are indicated in Figure 2. Only part of the data is included in this paper.



INDEX	E2	E3	E4	E6	E9	E10	E12	E15
r/r_T	1.066	1.066	1.066	1.066	1.066	1.066	1.267	1.187
Θ	18.85°	12.44°	6.18°	360.0°	347.6°	341.2°	21.25°	5.55°

Fig. 2 Location of pressure transducers within the volute.



TONGUE	t_c/r_T (%)
1	7
2	11
3	18
4	25
5	27

t_c: the clearance between the impeller and the tongue

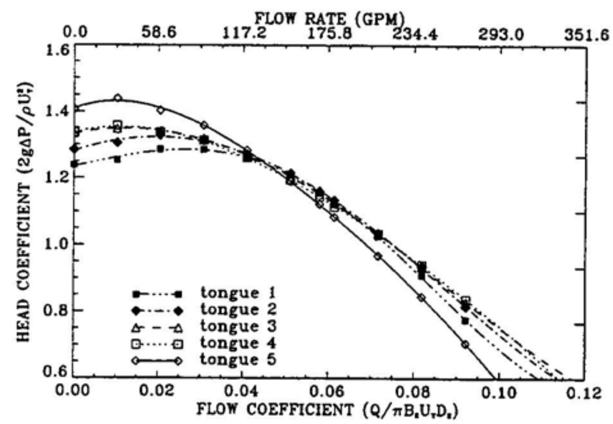
r_T: the radius of the impeller

SECTION	SIZE OF SAMPLE SECTION (INCH)
A	2.75 X 2.0
B	2.46 X 2.46
C	3.2 X 3.2

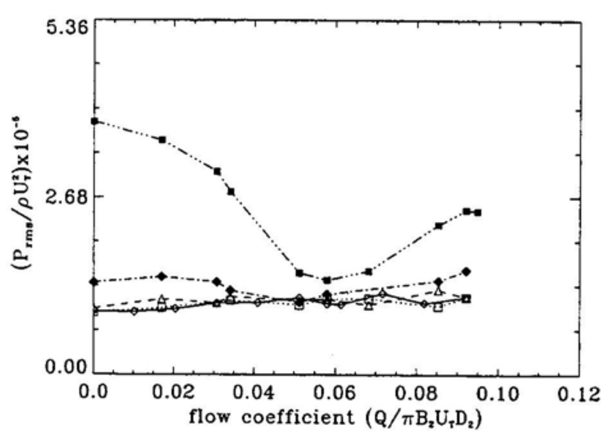
Fig. 3 Sample areas and clearances associated with different tongue geometries.

EFFECT OF TONGUE GEOMETRY

The five tongue geometries selected for the present study are sketched in Figure 3. Tongue No. 1 with 7% gap is the original geometry studied by Chu et al. (1993, 1995). Variations in pump performance, rms noise levels and spectra at 270 gpm are presented in Figure 4a-c. It is evident that increasing the gap up to 23% improves the pump performance off design conditions, and has little effect at design flux, besides a small decrease with tongue No. 4. Further increase in the gap causes a shift of the entire curve towards lower flow rates, with increase in head below design conditions, and substantial reduction above 150 gpm. The conclusions for gaps above 23% are consistent with the results of Hira and Vasandani (1975). The impact on reduction of rms noise levels (Fig. 4b) is particularly noticeable between tongues 1 and 3. Further increase in gap does not seem to have a significant effect. The spectra (Fig. 4c) show a reduction of the primary harmonics (the blade rate frequency is 105 Hz) when the gap is increased, particularly from 7% to 18%. The broad-band noise also decreases significantly with increasing gap, but with diminishing effects between tongues 3 and 4, and a slight increase between 4 and 5.

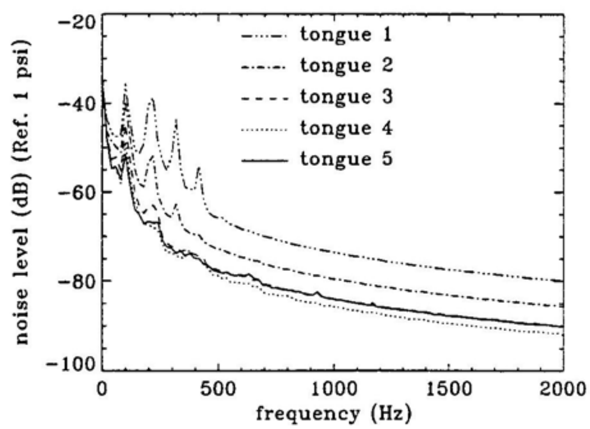


(a)



(b)

Fig. 4 (cont.)



(c)

Fig. 4 Effect of the tongue geometry on (a) the performance of the pump, (b) rms noise and (c) spectra of noise at Q/Q_D=1.35 (Q_D=200 gpm).

The spectral differences suggest that not only the amplitude of noise, but also the flow structure causing it, changes with increasing gap. The impact of changing the gap on the phase averaged velocity, vorticity, local pressure fluctuations and noise are illustrated in Figures 5-9, respectively (unless clearly stated in the caption, all the Figures in this paper contain phase averaged results). Recall that in the original study with tongue 1 (Dong et al., 1992 and Chu et al., 1995) we identified that impingement of the wake and its associated vorticity field on the tip of the tongue as well as oscillations of the tongue are primary causes of the main peaks in the noise signal. The existence of a jet (high v_r in Fig. 5) ahead of the blade and a wake (high v_θ in Fig. 6) at about 30% of blade passage behind the blade are clearly evident in the velocity distributions, irrespective of tongue geometry. The wake is also clearly marked with a negative vorticity train along its leading edge and positive train behind it (Fig. 7). The jet is also bounded by a pair of vorticity peaks with opposite signs. The negative peak is located in part within the separated region on the tip of the blade, and the positive peak is located ahead of the jet. At the request of reviewers of this paper Figure 8 contains two sample vector maps of the velocity distributions that highlight the flow non-uniformity. Note that each map actually shows the magnitude and direction of $v-v_{ref}$ (the latter is indicated on each map) in order to increase the clarity. The jet is evident in both cases, and impingement of the wake on the tongue is demonstrated in the upper example.

With tongue No. 1 the gap between the tongue and the impeller is small, and since the flow is above design conditions the leakage is also relatively small (Dong et al., 1992). Consequently, as the wake and associated vorticity trains reach the tip of the tongue, they impinge on it and move towards the exit. The impingement cause a peak in both the local pressure fluctuations (e.g. P(E15) at t/T~0.2 in Figure 9c) and far field noise (signal of tongue 1 at t/T~0.2 in Figure 10a). Due to the narrow gap the vorticity peak associated with the jet remains "stuck" in the corner as the blade moves behind the tongue, and coincides with the wake vorticity train when it arrives to the corner. This process is illustrated (in another context) in Figures 17b and 19b. Increasing the gap to 11% (tongue 2) leaves a larger space for leakage. Consequently, portion of the wake, which is closest to the impeller, leaks to the beginning of the volute, substantially reducing its impact on the tip of the tongue. This phenomenon is demonstrated by comparing the corresponding circumferential velocity distributions in Figure 6, and from the vorticity distributions

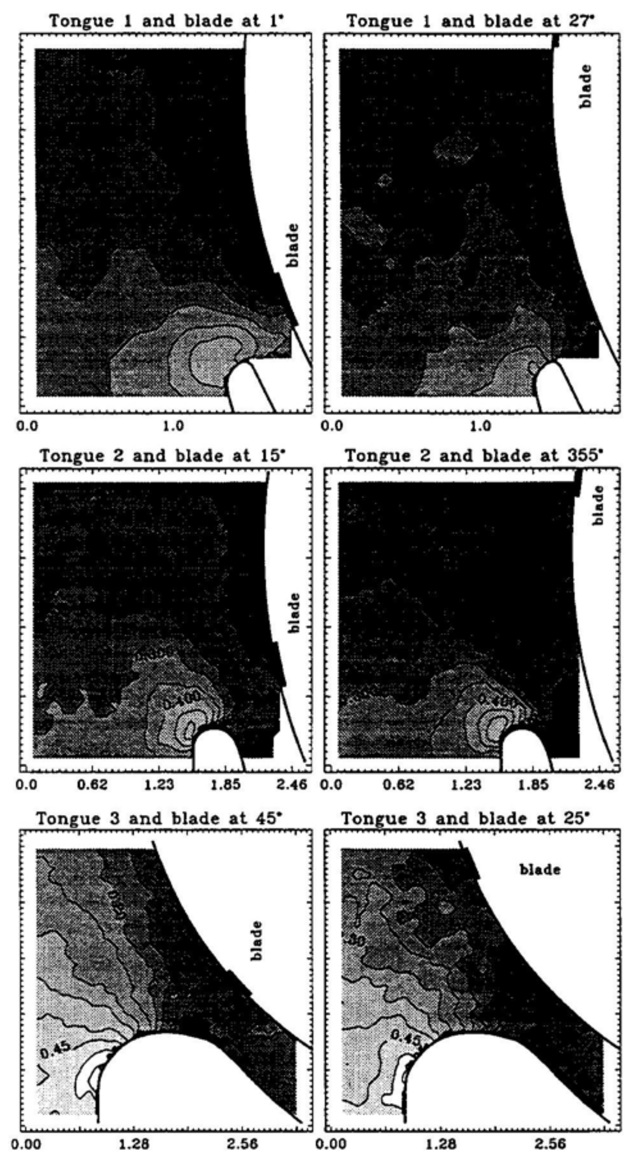


Fig. 5 Effect of modifications to tongue geometry on distributions of v_r/U_T. Increment between adjacent lines is 0.05. Scales are in inches. Flow rate is 270 gpm (Q/Q_D=1.35).

(Figure 7). With tongue 2, part of the wake vorticity trains and all of the jet vorticity peaks leak to the beginning of the volute. When the gap is increased to 18% most of the wake and associated vorticity train miss the tongue.

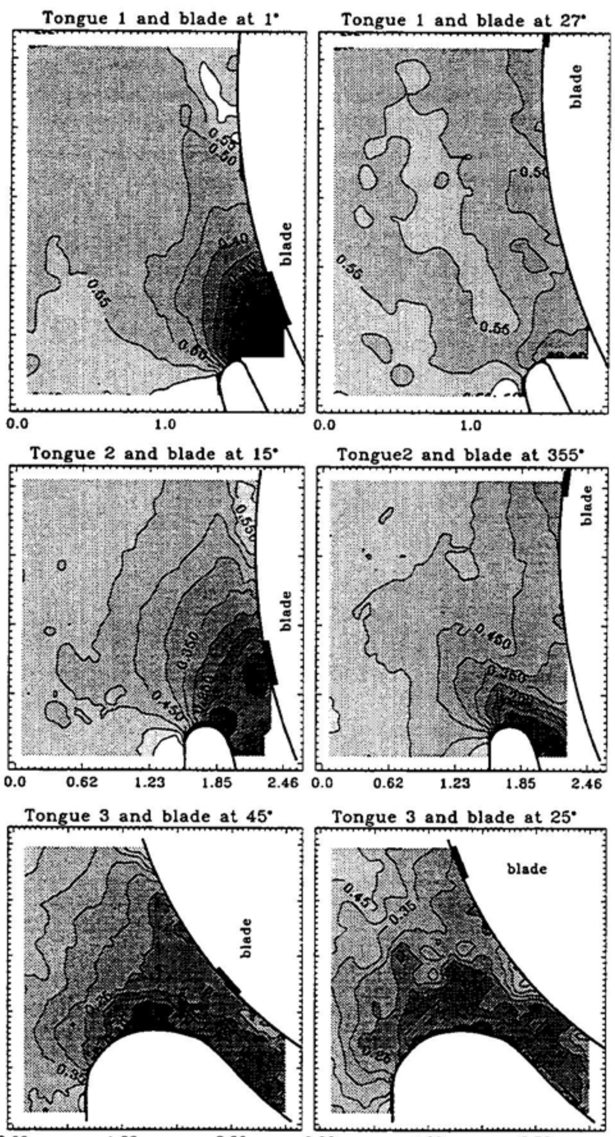


Fig. 6 Effect of modifications to tongue geometry on distributions of v_θ/U_T. Increment between adjacent lines is 0.05. Scales are in inches. Flow rate is 270 gpm (Q/Q_D=1.35).

The impact of these changes on local pressure fluctuations is demonstrated in Figures 9a and b, that compare corresponding points located upstream of the tongue, as well as in Figure 9c for points located at the exit from the volute. Near the impeller and relatively far upstream of the tongue (Figure 9a), both signal have two minima during one blade cycle. The first occurs when the blade passes by the transducer (at t/T=0.33) and the second occurs when the negative vorticity train passes by (at t/T~0.65). Although the magnitudes are different, both minima are evident for either case. As noted before, impingement of the wake on the tip of tongue 1 causes a pressure pulse which is sensed throughout the exit region. This peak is significantly smaller with tongue No. 2. It causes a small "bump" in the pressure waveform at t/T~0.3 (at E15) and a slight "kink" in the noise signal. Wake impingement does not affect the noise signals of tongues 3-5.

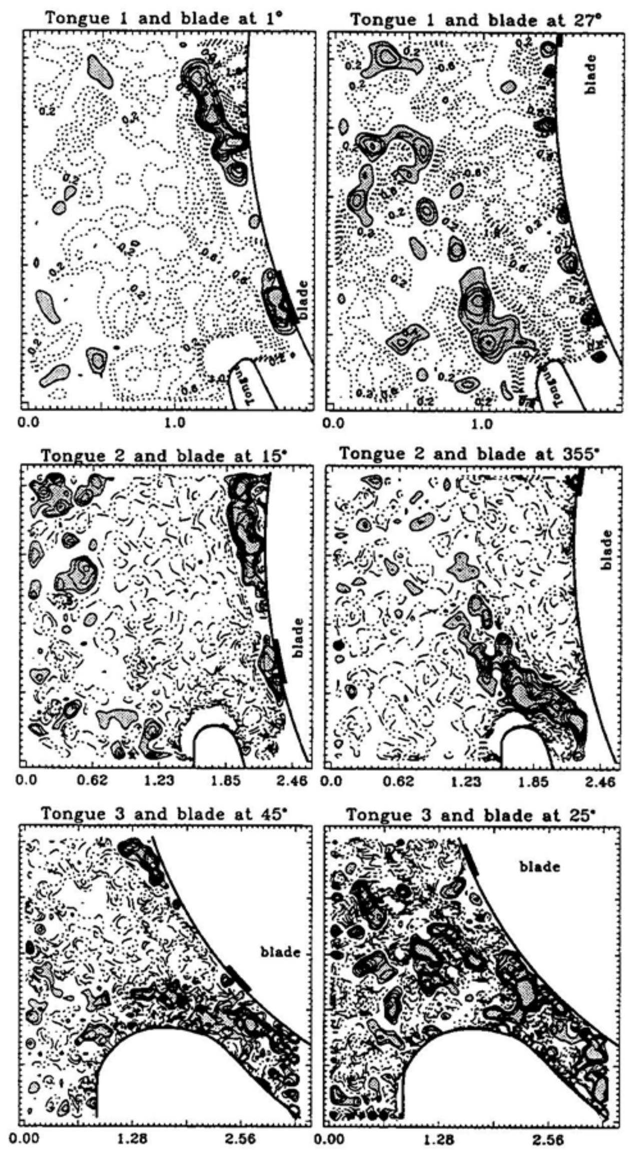


Fig. 7 Effect of modifications to tongue geometry on vorticity distributions. Increment between adjacent lines is 0.2 U_T/r_T. Solid lines represent negative vorticity, and dashed lines are positive vorticity. Scales are in inches. Flow rate is 270 gpm (Q/Q_D=1.35).

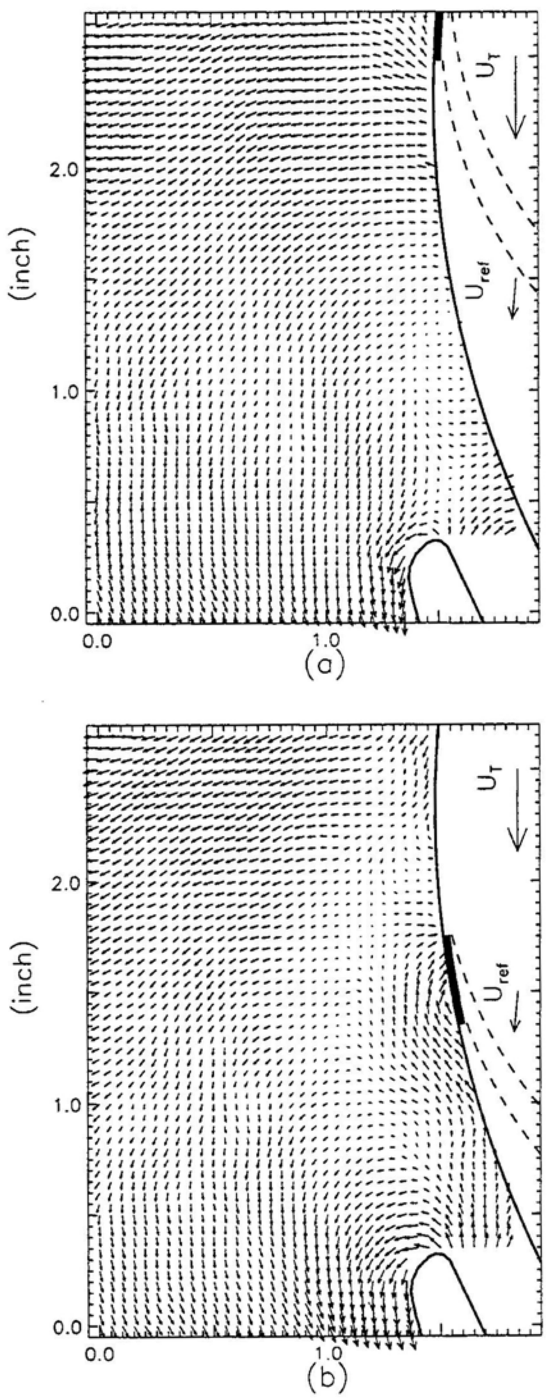


Fig. 8 Sample phase-averaged velocity distributions of tongue 1 data. The impeller is at: (a) 337° and (b) 351°. Data is shown as $v\text{-}v_{ref}$. The latter is also plotted.

Another cause for unsteady phenomena is associated with operation above design conditions and the resulting difference in mean pressure across the tongue. It causes reduction in pressure throughout the exit region (e.g. E15 at t/T~0.6) when the gap between the tongue and the impeller is minimal, namely when the blade lines up with the tip of the tongue, and an increase in pressure when the gap is large (the blade is far from the tongue). This phenomenon is responsible in part for the minima at t/T~0.6 in Figures 9a and b . The trend is reversed across the tongue, namely at the beginning of the volute (only one sample is provided for E4 in Figure 18). Combined, it causes large fluctuations in the pressure difference across the tongue (Also shown in Figure 18 for tongue 1), resulting in tongue oscillations and radiation of noise. This process is responsible for the second peak in the noise signal at t/T~0.62 (Figure 10a). The same process and resulting noise occur with tongue 2. As the tongue becomes thicker (tongue 3, for example), and the gap becomes larger, this effect disappears.

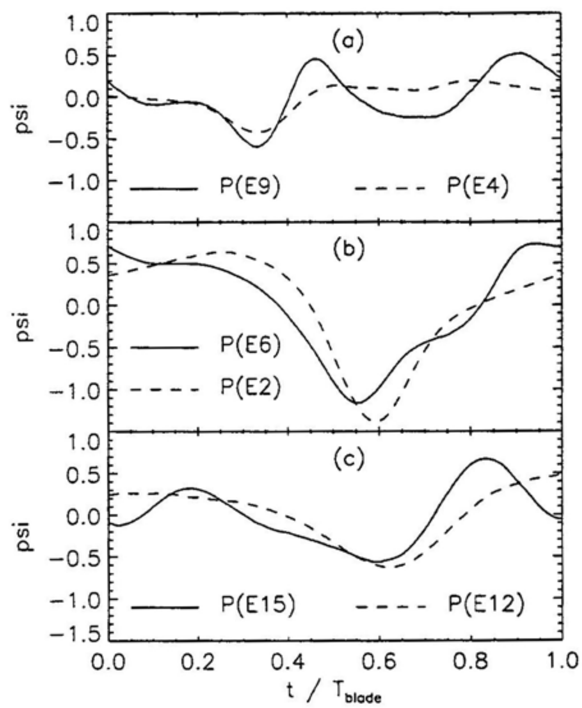
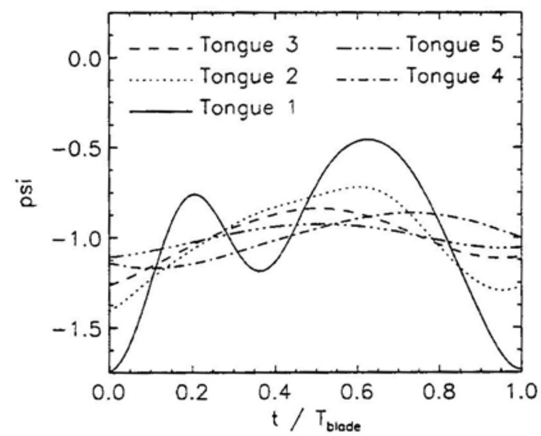
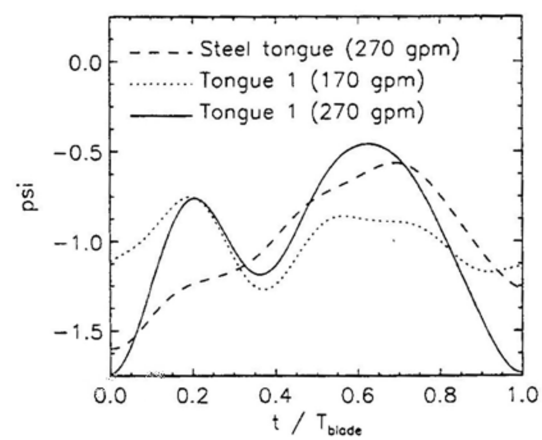


Fig. 9 Phase-averaged pressure waveforms at corresponding points for tongue 1 (solid line) and tongue 2 (dash line).

Close to design conditions (at 170 gpm in Figure 10b), when the pressure difference across the tongue is small, the noise peak associated with tongue oscillations disappears. The waveform has only one maximum, that occurs at the same time as the wake impingement peak. In order to clarify the effect of tongue oscillations further, we also replaced the lucite tongue with a steel tongue that had an identical shape (No. 1). The waveforms at E4, E6 and E10 (data not shown) do not change significantly, but there is substantial reduction in the magnitude of the peaks caused by wake impingement and tongue oscillations at the exit region (E15 - data not shown) and in the noise signal (Figure 10b). Note that the noise of a steel tongue No. 1 is still considerably higher than that of a lucite tongue No. 2. In order to explore the reasons for the considerable differences between the steel and lucite tongues, we also measured the frequency response of the lucite tongue. This task was performed by bending the tongue and releasing it abruptly in a stationary (impeller at rest), water filled facility. An extension to one of the blades, that pushed the tongue outward and released it as the impeller was manually rotated, was used for this purpose. The resulting signals at E15 and the hydrophone (see a sample spectrum at Figure 10c) were quite repeatable and had spectral peaks at 84, 136, 208 and 296 Hz. The first two peaks do not match the blade rate frequency (105 Hz), but the third is quite close to the blade's second harmonics. A comparison between the spectra of the steel and lucite tongues (Figure 10d) clearly shows that there is little change in the blade rate frequency, but there is substantial difference in the magnitude of the peaks at 210 and 315 Hz. These results suggest that resonant response of the lucite tongue may have contributed to the noise amplitude. However, as noted before, a steel tongue No. 1 is still much noisier than lucite tongues 2-5.



(a)



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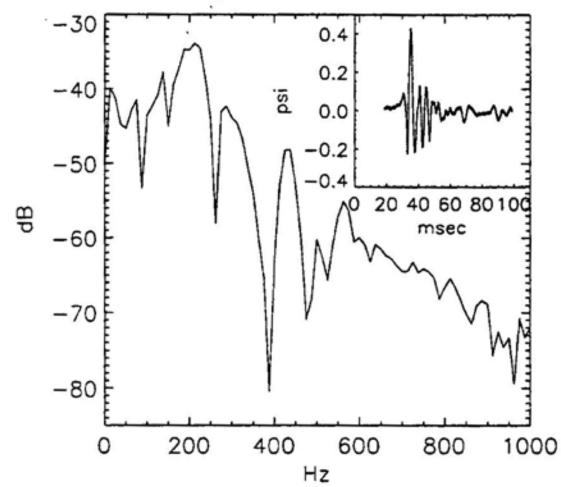

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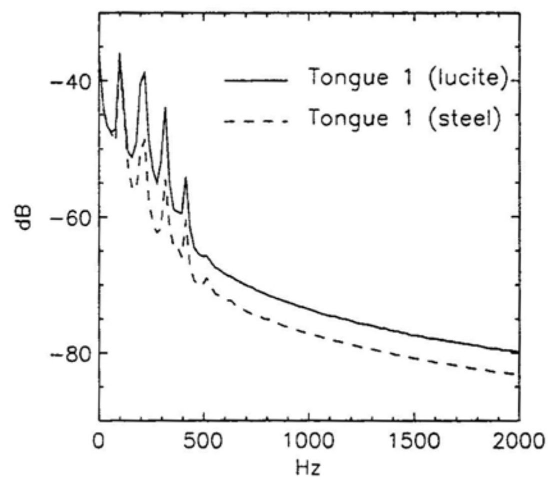

(d)

Fig. 10 (a) Effect of tongue geometry on the phase-averaged noise. (b) Noise signals of a steel tongue 1 at 270 gpm and lucite tongue 1 at 270 and 170 gpm. (c) The pressure signal resulting from bending and releasing the lucite tongue 1 and its spectrum. (d) A comparison between the noise spectra of the steel and lucite tongue.

The last unsolved issue involves the sources of noise of tongues 3, 4 and 5. This task is not simple since the noise levels are low and contain a considerably larger broad band component. Thus, the flow phenomena causing them are not as powerful and distinct. Focusing on blade-tongue interactions (tonal noise) of tongue 3 (see data in Figure 5-7) we examined the velocity distributions carefully. The wake seems to have little effect on the flow around the tongue. The jet (increased v_r ahead of the impeller), on the other hand, impinges on the tip of the tongue when the blade is between 25° and 35° and has little effect for the rest of the cycle. The noise peak occurs when the blade is at about 31° (at t/T~0.5), suggesting that these phenomena are related. The same process occurs with narrower gaps, but there the wake and tongue oscillations are dominant. With tongue 4 the amplitude of the noise signal is even slightly lower. The timing of the peak is consistent with the observations on tongue 3, namely it occurs when the impeller is located about 10° upstream of the tongue (data not shown here - it does not add anything significant). With tongue 5 we cannot identify a clear relationship between flow and noise.

EFFECT OF MODIFICATIONS TO THE IMPELLER

As discussed in the previous section and in Chu et al. (1995), when the tongue is located at $r/r_T<1.2$, the flow around it and resulting noise are dominated by nonuniformities in the outflux from the impeller. Thus, it is likely that any reduction in the jet/wake phenomenon would have a direct impact on decreasing the noise level. To achieve this goal we inserted short vanes (Figure 11) within the wake based on the data at 270 gpm, i.e. $Q/Q_D=1.35$, where $Q_D=200$ gpm (the corresponding flow coefficient of 0.07), 10° behind the origin of the negative vorticity train (Figure 7). Their exit angles, 18°, was set to reduce the magnitude of v_θ within the wake to v_θ/U_T ~0.5, namely to levels comparable to the flow outside of the wake (Figure 6).

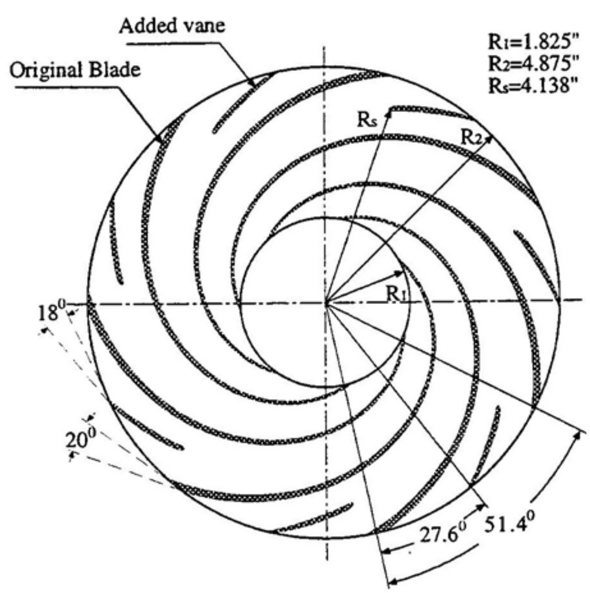

Fig. 11 A sketch of the modified impeller.

For $0.5<Q/Q_D<1.25$ the vanes improved the pump performance slightly (Figure 12), but reduced it at lower flow rates. The rms noise (Figure 13) decreased above design conditions (the conditions for which they were designed) and at low flow rates, but increased at $0.4<Q/Q_D<1$. Thus, for flows with significant outflux from the impeller near the tongue (above design conditions, according to Dong et all., 1992), the vanes contributed to noise reduction. The spectrum at 270 gpm (Figure 14) clearly shows a substantial decrease in the magnitude of the primary harmonic (blade rate frequency - 105 Hz) and its odd multiples, no change in the second peak, but a slight increase in the fourth harmonics. The latter can be easily explained by the effective doubling of the number of blades. There is also a noticeable decrease in the broadband noise level.

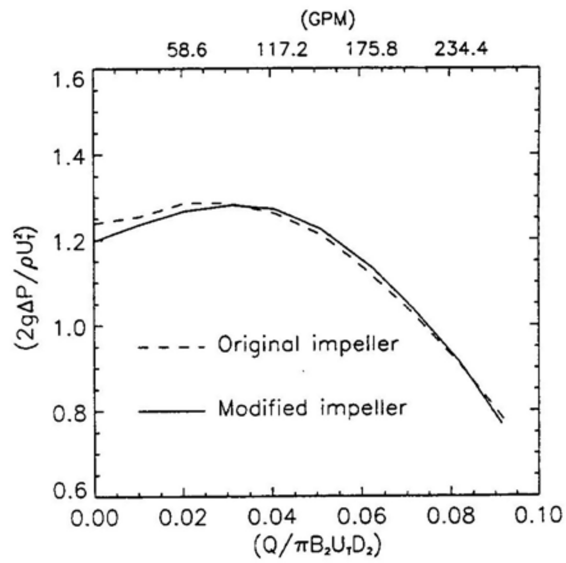

Fig. 12 Performance curves of the original and modified impellers.

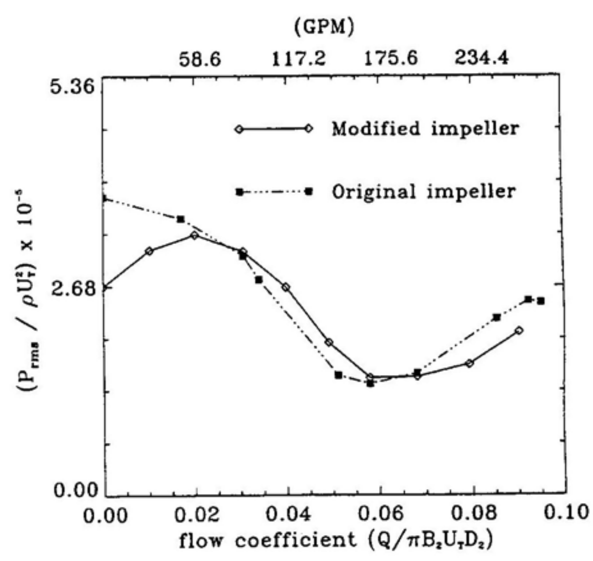

Fig. 13 Rms noise of the original and modified impellers.

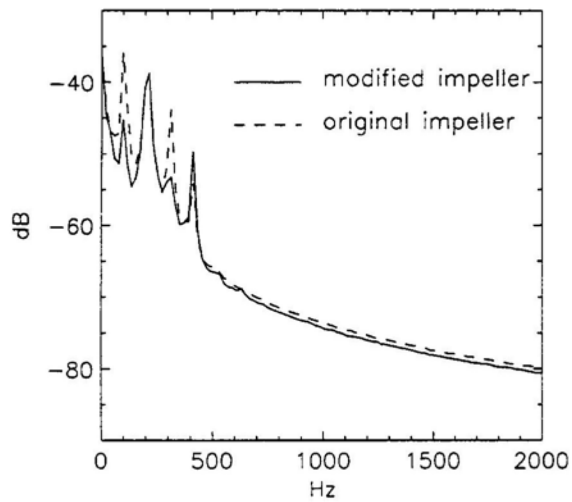

Fig. 14 Noise spectra of the original and modified impellers at Q=270 gpm.

Selected distributions of v_r,v_θ and vorticity within the modified pump are compared to the original data in Figures 15-17, respectively. They demonstrate that the vanes decrease the original blade jet/wake phenomenon, but create additional similar patterns of non-uniform outflux. For example, in Figure 16a the vane significantly reduces, but does not eliminate, the size of the wake region. However, as shown in Figure 16c, the vane generates an additional wake. Evidence for the formation of an additional vane wake is provided also by the vorticity distributions. In Figure 17a the new vorticity train extends from the impeller and in Figure 17c it appears on top of the tongue. As expected, Figure 15a shows that the vane has little effect on the blade jet (it was not designed for it). Its own jet is much weaker (Figure 15c), as long as the vane is located far from the tip of the tongue (a sample for the latter, which will be discussed later, is presented in Figure 19). Finally, note that traces of wakes generated by the previous blade and vane can be identified on the left sides of Figures 16a and c, respectively.

The vorticity peak associated with the jet, that in the original pump appears near the blade (Figure 17a), does not exist when the impeller is modified. It does form as the blade moves closer to the tongue (data not shown) as the radial velocity at the exit from the impeller increases. This phenomenon is a direct result of increasing blockage to blade passage caused by the tongue, that increases the radial velocity in the unobstructed part. In the original impeller this vorticity peak "gets stuck" between the tongue and the impeller as the blade moves behind the tongue (Figure 17b) because of the low velocity there. In the modified impeller the new vane increases the radial velocity sufficiently (Figure 15a) to push this blade vorticity away.

Comparisons of phase averaged pressures and noise waveforms, are presented in Figure 18. It is evident that near the tongue (E6) and at the exit (E15) the pressure fluctuations are considerably smaller, whereas somewhat far upstream of the tongue (E10) they are slightly higher. Detailed explanations for the signals of the unmodified impeller, resulting from direct computations of pressure from the PIV data, are presented in Chu et al., 1995. It is demonstrated there that near the impeller (E10 and E6) the pressure minima occur when the blade passes by the transducer, under the negative vorticity train (Figures 7 and 17), and when the blade lines up with the tip of the tongue. The latter occurs almost simultaneously everywhere and is associated with operation above design conditions. The maxima appear when the transducer is located between the blade and the wake, and everywhere when the blade is located far (35°) upstream of the tongue. With the modified impeller, in addition to the pressure minima associated with the original blades (points C and F at E6 and E10 , respectively) there are new pressure minima when the vane passes by the transducer (points A and I). Reductions in pressure when the vorticity trains pass by are barely noticed near the modified impeller, mostly since they are overshadowed by simultaneous blade and vane effects. For example, the blade vorticity train reaches E10 (point E) at about the same time that the vane passes by (they are designed that way).

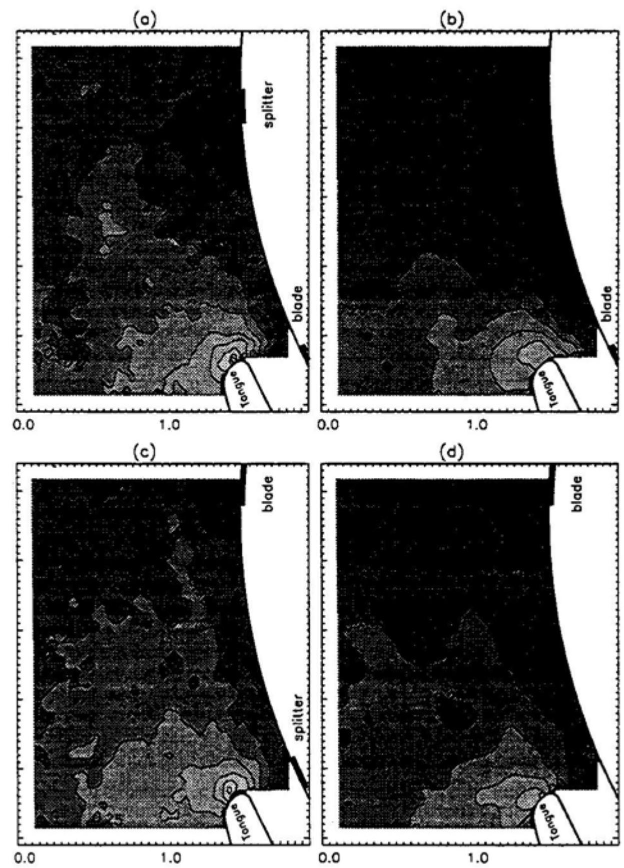

Fig. 15 v_r/U_T distributions at Q=270 gpm: (a) blade is at 8° (modified impeller); (b) 8° (original); (c) 338° (modified); and (d) 338° (original). Increment between lines is 0.05. Scales are in inches.

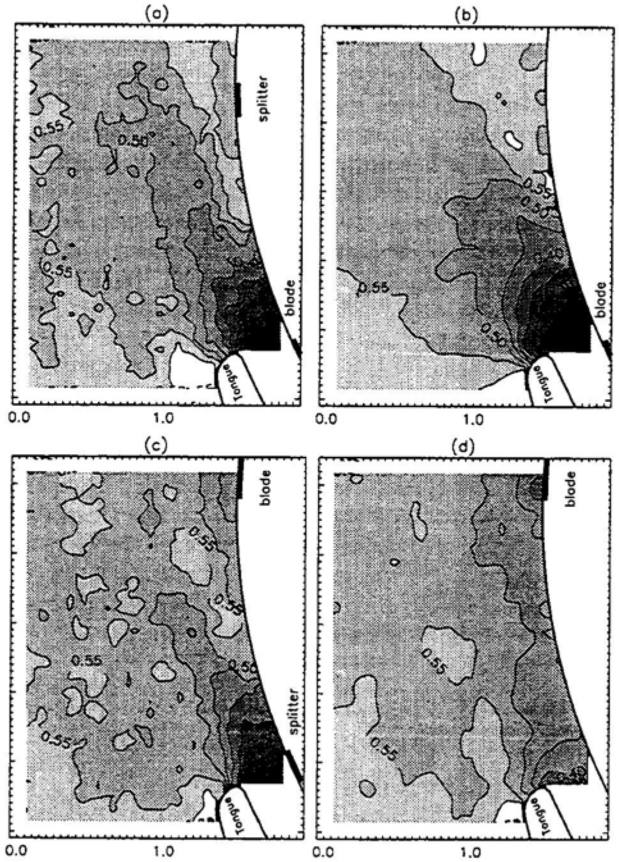

Fig. 16 v_θ/U_T distributions at Q=270 gpm: (a) blade is at 8° (modified impeller); (b) 8° (original); (c) 338° (modified); and (d) 338° (original). Increment between lines is 0.05. Scales are in inches.

Similar to the impact of the blade, the pressure decreases everywhere in the exit region when the vane lines up with the tip of the tongue (points A and E at E6 and E10, respectively). Pressure maxima occur simultaneously (point D at E6 and J at E10) when the blade is past the tip of the tongue and the vane is still far upstream of it. These peaks occur earlier than the original impeller because of the added vane and the pressure dip near it (point A). Simultaneous pressure maxima occur also when the blade is still about 14° upstream of the tongue and the vane is relatively far (10°) past it (points B and G). To recapitulate, maxima occur everywhere when the blade and the vane are far from the tip of the tongue, and the minima appear when they are very close to the tongue. Pressure minima occur also when the blade passes by the transducer, but they are barely noticed under the vorticity trains. The latter conclusion is the only result which is inconsistent with the trends observed with the original impeller.

Behind the tongue (E4) the vane impacts the pressure waveform only in its vicinity, causing a new minimum as the vane passes by (point K). Close to the blade, the signal remains quite similar to the original impeller. On the other side of the tongue, near the exit (E15), there are two minima (points L and N), that occur, like anywhere else in the exit region, when the blade and the vane coincide with the tip of the tongue. In the original pump there is a high pressure peak at E15 (point M) when the wake impinges on the tip of the tongue. Although the impingement still occurs (Figure 17c), it has little impact on the pressure at E15. It is possible that weakening of the wake (Figure 16a) in the vicinity of the tongue reduces its impact to a level that it is barely noticed. One of the contributing reasons for this weakness is the jet of the vane. As noted before this jet is fairly small until very close to the tongue, where blockage to blade passage induced by the tongue, causes an increase in radial velocity in the unobstructed space (an example of such a process with the main blade is illustrated in Figure 5). This jet "blows" the wake away from the tongue, before it has a chance to impinge on it. A clear illustration of this blowing process is provided in Figure 19. For this reason the vorticity train of the original impeller appears continuous up to the tongue (Figure 17a), whereas the train of the modified impeller is much more fragmented. Similar conclusions can be drawn from the

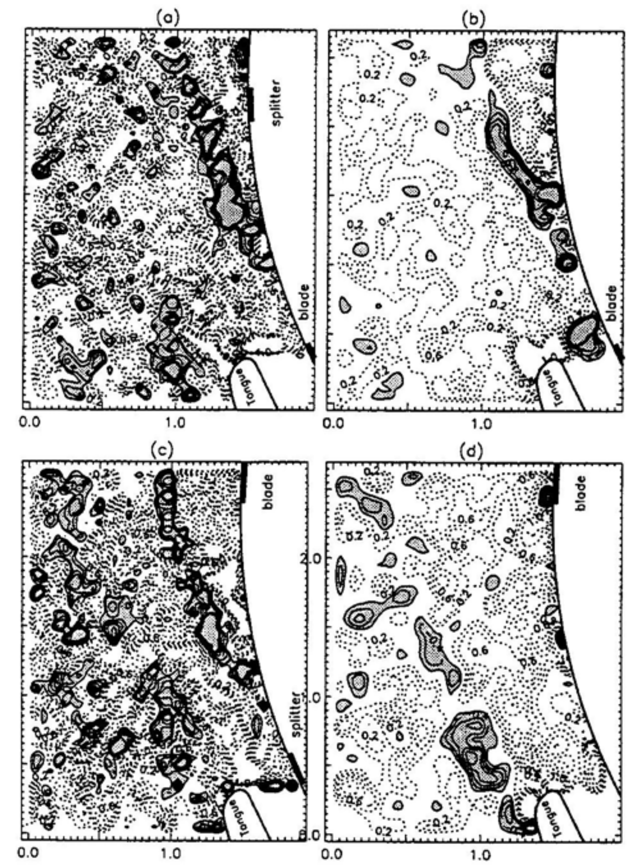

Fig. 17 Vorticity distributions at Q=270 gpm: (a) blade is at 8° (modified impeller); (b) 8° (original); (c) 338° (modified); and (d) 338° (original). Solid line represents negative vorticity, and dashed line positive. Increment between lines is 0.2 U_T/r_T. Scales are in inches.

distributions of v_θ (Figure 16a). Thus, the vane not only reduces the size of the wake, it also prevents impingement on the tip of the tongue. The blade has a similar effect on the wake of the vane (data not shown here).

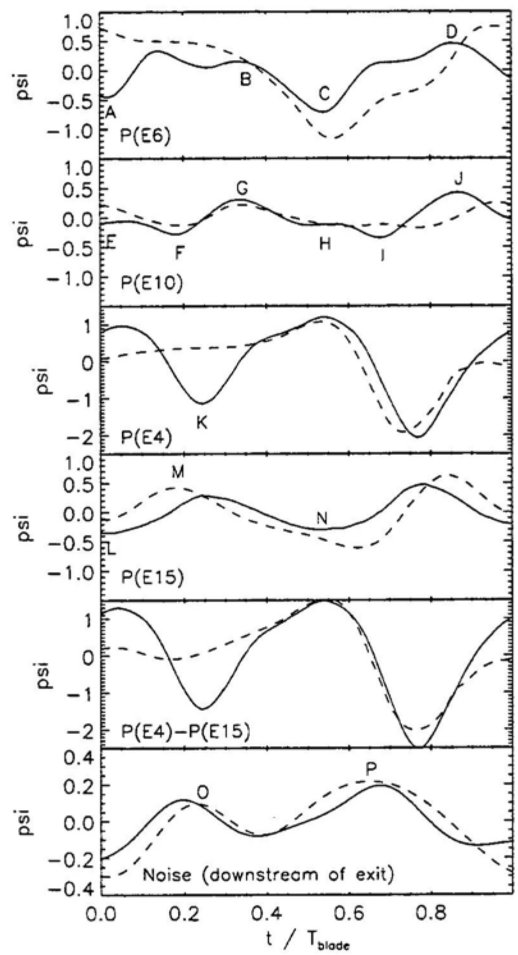

Fig. 18 Phase-averaged pressure and noise waveforms of the original (dashed line) and modified (solid line) impellers. (Note the differences in scale.)

As discussed and demonstrated in Chu et al. (1995), wake impingement causes also a peak in the noise signal (point O) of the original impeller. The delay between points M and O is approximately the time required for the pressure pulse to travel between E15 and the hydrophone. This relationship is obviously not the case with the modified impeller, first since the hydrophone peak occurs before the maximum at E15, and second, since the pressure at E15 in the modified impeller is not affected by the impingement. Thus, it is unlikely that the first peak in the noise signal has anything to do with wake impingement. Furthermore, the data in Figure 18 suggests that the hydrophone signal has no any clear resemblance or correspondence to the pressure in E15, E6 or E10. The only consistent relationship exists with the pressure at E4, or even more clearly, with the pressure difference across the tongue (P(E4)-P(E15)). Recall that with the original impeller the second noise maximum (point P) is clearly a result of tongue oscillations, induced by the pressure differences across the tongue. There, the delay between maxima in pressure difference and noise signals is about 0.7 msec, approximately the time required for sound to travel between the transducers and the hydrophone (the distance is ~3.3ft.). With the modified impeller, both noise peaks appear only about 1.4 msec after the maxima in pressure difference across the tongue. This delay is about twice the time required for sound to travel between the tongue and the hydrophone. To explain this discrepancy we note that with the original impeller, the pressure maximum behind the tongue occurs almost simultaneously everywhere when the blade lines up with the tip (there are some blade induced variations). Thus, the maximum force on the impeller should occur almost at the same time as the peak at E3. With the modified impeller, the added vane causes a pressure dip 27° ahead of the tip (similar to point K at E4). Thus, the maximum force on the tongue does not necessarily coincide with the maximum in P(E4)-P(E15) and a slight delay is quite possible. These results suggest that with the modified impeller tongue oscillations remain the only primary source of noise at the exit from the pump.

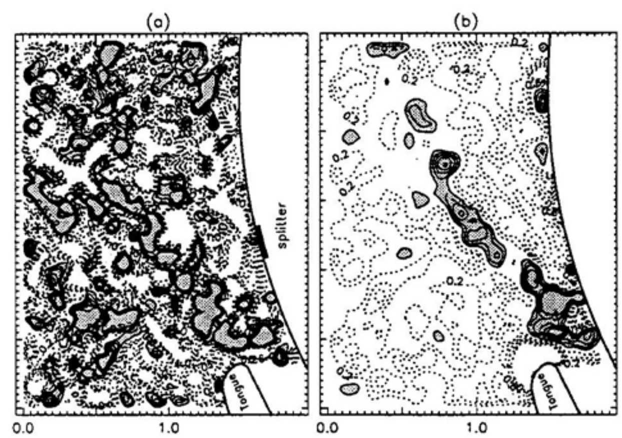

Fig 19 Vorticity distributions of: (a) modified (instantaneous) and (b) original (phase-averaged) impellers that demonstrate blowing of the wake by the jet ahead of the vane. Increment between lines is 0.02 U_T/r_T. Scales are in inches.

SUMMARY AND CONCLUSIONS

PIV, surface pressure and noise measurements have been used to study the effects of modifications to tongue and impeller geometries on the flow structure, local pressure fluctuations and noise. It is demonstrated that increasing the gap between the tongue and the impeller, up to about 20% of the impeller radius, reduces the impact of nonuniform outflux from the impeller (mostly the jet/wake phenomenon) on the flow around the tongue and noise. Further increase in the gap affects the performance adversely with minimal impact on the noise level. When the gap is narrow, the primary sources of noise are impingement of the wake on the tip of the tongue, and tongue oscillations when the pressure difference across it is high. At 20% gap, the entire wake and its associated vorticity trains miss the tongue, and the only effect of nonuniform outflux from the impeller occurs when the jet impinges on the tongue. This effect is quite small, at least for the present pump.

Based on the available data we attempted to reduce the non-uniformity in outflux from the impeller by inserting short vanes between the blades. Their location and orientation were selected to reduce the size of the wake and the magnitude of circumferential velocity within it. The experiments were performed only with the narrowest gap between the tongue and the impeller. The results showed that although the original intent was successful, namely the size of the original wake was reduced, the vanes generated a jet/wake phenomenon of their own. However, both wakes were considerably weaker to a level that their impacts on local pressure fluctuations and far field noise were insignificant. The jet in front of the vane was an added benefit, since it blew the wake of the blade away from the tongue shortly before impingement, eliminating any direct impact of the wake on noise. The only remaining contributors to noise were tongue oscillations caused by the pressure difference across it. They reach maximum level when either the blade or the vane line up with the tip of the tongue. This effect was particularly noticed with the present fairly narrow lucite tongue. As the experiments with the other tongues have shown, the impact of tongue oscillations decreases with increasing tongue thickness and stiffness.

ACKNOWLEDGMENT

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REFERENCES

[1] Chu, S., Dong, R., and Katz, J., (1993), "Unsteady Flow, Pressure Fluctuation and Noise Associated with the Blade-Tongue Interaction in a Centrifugal Pump", Symposium on Flow Noise Modeling, Measurement and Control, ASME Winter Annual Meeting, New Orleans, Louisiana, Nov. 28 - Dec. 3.

[2] Chu, S., Dong, R., and Katz, J., 1995a, "Relationship between Unsteady Flow, Pressure Fluctuations and Noise in a Centrifugal Pump. Part A: Use of PIV Data to Compute the Pressure Field", to appear in the J. of Fluids Engineering, Vol. 115, No. 1.

[3] Chu, S., Dong, R., and Katz, J., 1995b, "Relationship between Unsteady Flow, Pressure Fluctuations and Noise in a Centrifugal Pump. Part B: Effect of Blade-Tongue Interaction", to appear in ASME J. of Fluids Engineering, Vol. 115, No. 1.

[4] Cumpsty, N. A., 1989 "*Compressor Aerodynamics*", Publisher: Longman Scientific & Technical.

[5] Dean, R. C., and Senoo, Y., 1960, "Rotating Wake in Vaneless Diffusers", ASME J. of Basic Engineering, Vol. 82, No. 3, pp 563-574.

[6] Dong, R., Chu, S., and Katz, K., (1992a), "Quantitative Visualization of the Flow within the Volute of a Centrifugal Pump. Part A: Technique", ASME J. of Fluids Engineering, Vol. 114, No. 3, pp 390-395.

[7] Dong, R., Chu, S., and Katz, K., (1992b), "Quantitative Visualization of the Flow within the Volute of a Centrifugal Pump. Part B: Results", ASME J. of Fluids Engineering, Vol. 114, No. 3, pp 396-403.

[8] Eckardt, D., (1975), "Instantaneous Measurements in the Jet/Wake Discharge Flow of a Centrifugal Compressor Impeller", ASME J. of Engineering for Power, Vol. 97, No. 3, pp 337-346.

[9] Hamkins, C. P., and Flack, R. D., (1987), "Laser Velocimeter Measurements in Shrouded and Unshrouded Radial Flow Pump Impellers", ASME J. of Turbomachinery, Vol. 109, pp 70-76.

[10] Hira, D. S., and Vasandani, V. P., (1975), "Influence of Volute Tongue Length and Angle on the Pump Performance", Journal of Mechanical Engineering (Indian), Vol. 56, pp 55-59.