Hindawi Publishing Corporation
International Journal of Rotating Machinery
Volume 2009, Article ID 704845, 9 pages
doi:10.1155/2009/704845

Research Article

Experimental Investigation of the Effect of Radial Gap and Impeller Blade Exit on Flow-Induced Vibration at the Blade-Passing Frequency in a Centrifugal Pump

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Received 4 August 2009; Accepted 9 November 2009

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It has been recognized that the pressure pulsation excited by rotor-stator interaction in large pumps is strongly influenced by the radial gap between impeller and volute diffusers/tongues and the geometry of impeller blade at exit. This fluid-structure interaction phenomenon, as manifested by the pressure pulsation, is the main cause of flow-induced vibrations at the blade-passing frequency. In the present investigation, the effects of the radial gap and flow rate on pressure fluctuations, vibration, and pump performance are investigated experimentally for two different impeller designs. One impeller has a V-shaped cut at the blade's exit, while the second has a straight exit (without the V-cut). The experimental findings showed that the high vibrations at the blade-passing frequency are primarily raised by high pressure pulsation due to improper gap design. The existence of V-cut at blades exit produces lower pressure fluctuations inside the pump while maintaining nearly the same performance. The selection of proper radial gap for a given impeller-volute combination results in an appreciable reduction in vibration levels.

1. Introduction

This investigation is focused on the study of the radial gap and blade exit design in centrifugal pumps. The investigation was motivated by the need to trace the root-cause of high-level vibrations at the blade-passing frequency (BPF) and its higher harmonics, which existed in the boiler-feed pumps (BFPs) in a major power plant. The boiler feed pumps have a V-cut at impeller blades' exit. The problem caused cracking of the connecting piping welds and the gage attachments, which required frequent replacements of such components. The pump manufacturer attempted several remedial actions; however the problem persisted. A thorough study of the vibration records obtained through various field measurements at the pump casing and bearing housing as well as trending records has led to the elimination of other possible mechanical/structural sources of vibrations and pointed out to the flow-induced nature of the problem. Accordingly, the investigation was focused on the pump design, as related to the impeller-diffuser interaction and the effect of the radial gap.

The impeller/volute radial gap is an important design and performance parameter in fluid handling machines, for example, pumps, compressors, and turbines. The proper selection of such a gap is often a best compromise between efficiency and reliability. In high-energy pumps, the minimum radial gap is a critical design consideration that controls the pressure pulsation resulting from the impeller-volute interaction. Under off-design conditions, the pressure fluctuations are excited around the impeller due to such interaction, which give rise to higher pulsating radial loads on the pump rotor, thus causing higher vibration levels.

Pressure fluctuations due to impeller-volute interaction occur mainly at blade-passing frequency and its higher harmonics. A high vibration problem at blade-passing frequencies in a double volute boiler Feed Pump operating at reduced flow rates was also addressed in [1]. The energy content of these fluctuations is sensitive to the flow rate and was found to be location-dependent. Fluctuations became even more significant at partial flow rates. It is noteworthy to point out that the energies associated with the pressure fluctuations at suction and discharge pipes are much lower than those measured inside the pump. Consequently, measurements of the pressure field inside the pump are necessary to identify the source and critical locations of pressure fluctuations.

It was also noted that the impeller-volute interaction results in circumferential unevenness of pressure fluctuations around the impeller, which becomes more significant at off-design flow rates [2]. In general, the pump geometry and impeller/volute combination play an important role in the severity of the pressure pulsation inside the pump [3]. Large pumps are usually available in double (split) volute design to minimize the net radial force on the pump shaft, particularly when the pump operates away from the best efficiency point. Amplitudes of pressure fluctuations depend on both radial and axial locations inside the pump. Axially, pressure fluctuation at the blade-passing frequency is maximum at the central impeller plane while fluctuations are partially damped out at volute walls [4]. The flow-induced vibrations in large pumps have specific characteristics that can be observed in the frequency domain [5]. Harmonics of the blade-passing frequency dominate the spectrum, and there is a particular relation among their amplitudes. The rise in the amplitudes is not equal for all harmonics. The time variant and spatial nature of these pressure distributions are discussed in [6] where each interaction forcing frequency possesses a global time variant pressure distribution.

In general, the optimized gap selection depends on detailed pump design, that is, impeller/volute combination, and dynamic operating conditions. Decreasing the radial gap may produce higher heads; however it may increase the pressure pulsation inside the pump and give rise to high vibrations, due to stronger impeller-volute interaction. On the other hand, reducing the impeller-volute interaction by increasing the radial gap can reduce the pressure fluctuations. However, several reported studies showed that the total pump head drops with the gap increase. The pressure pulsations arising from the interaction of the impeller and vaned diffuser were measured in the discharge pipe at twice the impeller blade-passing frequency [7]. When the gap was increased from 3% to 12% of the impeller diameter, a reduction of 10% in the flow capacity at the rated head resulted from this modification, while the efficiency remained almost the same.

The effect of radial gap between impeller and diffuser on vibration, noise, and performance was examined, under different flow rate conditions, by trimming the impeller to achieve different gaps [8]. At the maximum radial gap between the impeller and diffuser, the overall levels of vibration and noise approached minimum values. Experiments carried out by [9] on a double volute pump suffering a high vibration level showed that increasing the radial gap reduces the pressure fluctuations at part-load conditions. Another type of high vibration at blade-passing frequency due to acoustic resonance is given by [10]. The most practical solution appeared to be changing the blade-passing frequency of the pump by changing the number of impeller blades, where the acoustic natural frequencies would not be excited within the desired operating speed range.

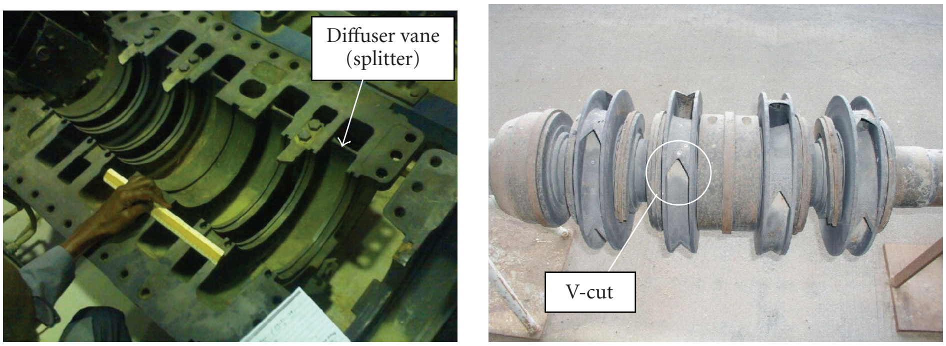
The application of numerical procedures has provided a very valuable tool to pumps design and analysis. Examples of CFD studies regarding the effect of pump variables on flow field and the unsteady pressure characteristics can be found in [11–14].

In this paper, the experimental centrifugal pump model, which was developed based on similitude analysis of the actual boiler feed pump, instrumented and tested in [1, 9], will be utilized to investigate the effect of radial gap and flow rate on pressure fluctuations inside the pump as well as monitoring the pump vibrations for two different impeller designs. In this context, one impeller has a V-shaped cut at blades exit while the second impeller has a straight exit (i.e., without the V-cut). It was of prime importance to examine the effect of the V-cut, as it existed in the impeller of the actual BFP. Moreover, the pump performance under the aforementioned design variables will be examined for comparison.

2. Experimental Setup

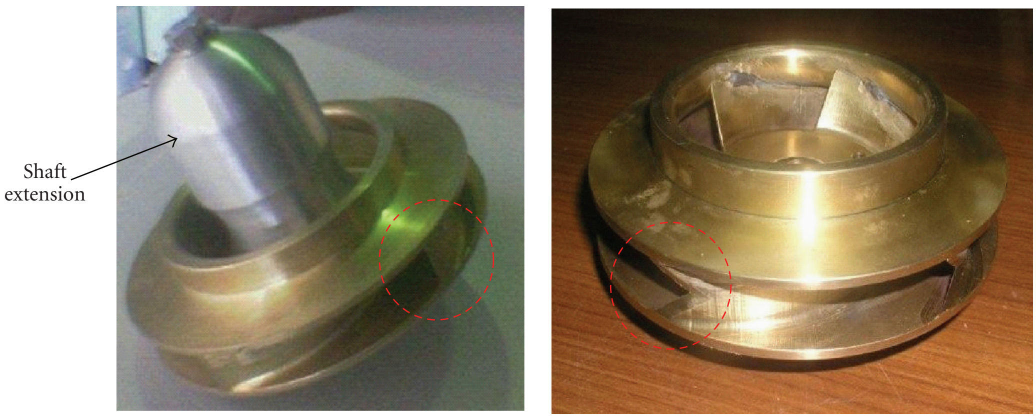
The actual boiler-feed pump is a four-stage double-casing centrifugal pump. Figure 1 shows the boiler feed pump cartridge (split volute and impellers). The rated speed is 5950 rpm at which the pump delivers 305 kg/sec of hot water at 198 bars and pump efficiency is 79%. Because it was infeasible to carry out an experimental investigation on a full-scale multi stage high-capacity pump, a single-stage scaled-down model was carefully designed based on similitude analysis by reverse engineering. A scale factor of 0.4 was selected for the impeller and volute produced using CNC machines. The scaling ratio for the model pump was selected taking into consideration similarity laws and practicality of the laboratory experimental setup. Two identical impellers were manufactured; one without V-cut (Impeller-I) while the second one has a V-cut at blades exit (Impeller-II). Each impeller was dynamically balanced on a two-plane balancing machine to minimize the effect of rotor imbalance as a source of vibration. Impeller diameter is 142 mm and the blade exit angle is 22.5°. A shaft-like extension was attached to the impellers to simulate the actual flow characteristics at the suction of the original boiler feed pump. The two impellers with the shaft extension are shown in Figure 2.

The model pump speed is 3540 rpm at 60 Hz. The flow rate is measured by an orifice meter with a discharge coefficient of 0.618. Orifice pressure taps are connected to a PDCR 4170, 700 mbar differential pressure transducer, which has $\pm 0.08\%$ FS accuracy. The flow rate is controlled by a gate valve in the discharge pipe. The investigation of the flow induced vibration behavior under variable flow rate and different impeller-gap combinations is done by measuring the pressure fluctuations inside pump volute, the pump vibration, and pump performance. Measurements of pressure fluctuations are carried out using OMEGA's DPX101-250 high-responses dynamic pressure transducers. Pressure transducers were arranged in geometrical symmetry positions around the impeller and at expected critical locations inside the pump as shown in Figure 3. The coordinates



(a) Split Volute (b) Impellers with V-cut at blade exit

FIGURE 1: Cartridge of the original boiler feed pump.



(a) Impeller-I: without V-cut (b) Impeller-II: with the V-cut

FIGURE 2: Model impellers.

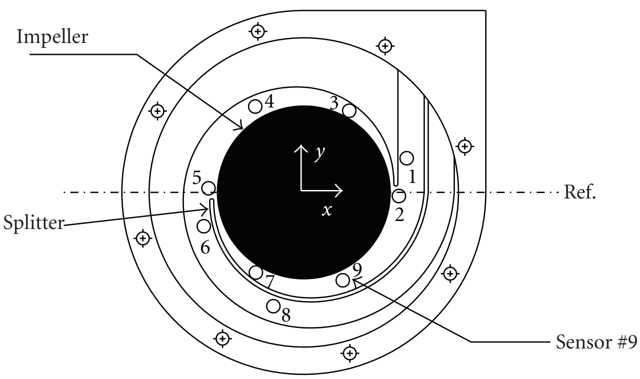


FIGURE 3: Model pump design and measuring locations.

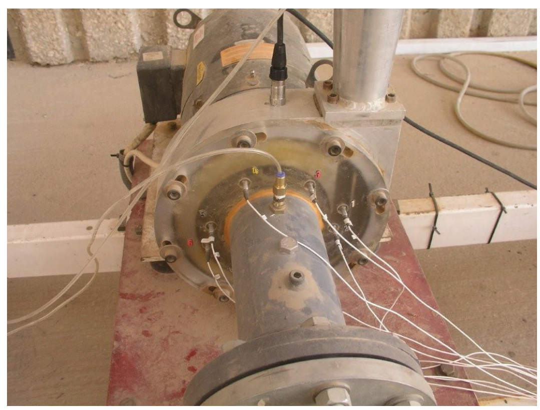


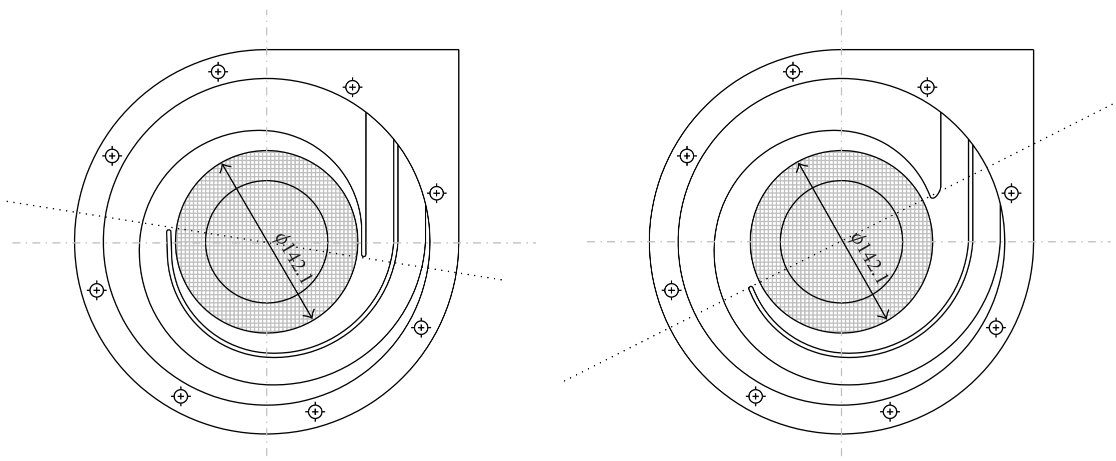
FIGURE 4: Measurements of pressure fluctuations: model pump.

of these locations relative to the volute horizontal centerline (Ref.) are given in Table 1. Nine pressure transducers were placed flush within the volute Plexiglas cover plate at the impeller discharge. One pressure transducer was placed at the discharge pipe and another at suction pipe. A photograph of the model pump testing is shown in Figure 4. In this setup, the same Plexiglas cover plate can be used for dynamic and static pressure measurements inside the pump at the same locations described in Table 1. Vibration of pump casing was measured using a B&K 4507 Delta Tron accelerometers.

The gap value of the original pump is 2.5% of the impeller diameter which is equivalent to 3.6 mm for the model pump. Smaller gaps were achieved by extending the volute splitters. The extension of volute vanes (splitters) was

Table 1: Coordinates of measuring locations.

Sensor no.	1	2	3	4	5	6	7	8	9	10	11
radius (mm)	90	78	77	81	79	88	77	97	80	Suction pipe	discharge pipe
angle (deg.)	19	358	61	120	178	199	239	255	294		



(a) Extension of volute splitters to get smaller gaps (b) Cutting back volute splitters to get larger gaps

Figure 5: Schematic for gap modifications.

done using epoxy filled mold of wax having the same vane curvature. Larger consecutive gaps were achieved by cutting back the volute tongues. The minimum gap tested in the experiment is 2 mm, while the maximum gap is 7 mm, as shown in Figure 5. This is equivalent to test gaps ranging from 1.4% to 5% of impeller diameter. Measuring locations remain fixed regardless of the tested gaps.

Table 2: Best efficiency conditions with uncertainty limits.

Parameter	Impeller-I	Impeller-II
Flow rate, Q_n [L/s]	12.07 ± 0.1771	12.4 ± 0.1769
Total head, H [m]	26.95 ± 0.2657	25.13 ± 0.2511
Efficiency, η %	55.14 ± 1.07	55.46 ± 1.10

3. Results and Discussions

The experimental matrix associated with testing the two impellers (with and without the V-cut designs) is established for different gaps of 2, 3, 3.6 (original gap), 4.85, 6, and 7 mm. Each impeller-gap combination was tested at different flow ratios (Q/Q_n) of 1, 0.75, 0.5, 0.25, zero, and maximum flow rate. Q_n represents the design flow rate corresponding to best efficiency point, for the original gap of 3.6 mm, for each impeller. Note that the maximum flow rate is not the same for the two impellers. Max flow ratio for impeller-I is 1.66 while it is 1.58 for impeller-II. Uncertainty analysis for the measured and calculated variables at the best efficiency point showed that the uncertainties are limited to 1% for the total head, 1.5% for the flow rate, and 2% for the efficiency (see Table 2). Pump specific speed is 1620.

A typical pressure fluctuation history (for two shaft revolutions time, 0.034 second) and the frequency spectrum, FFT, at location 3 are shown in Figure 6 for original gap of 3.6 mm, at design capacity $Q = Q_n$ (1 psi = 6894.75 Pa). Dominant peak occurred at a frequency of 295 Hz, which corresponds to $5x$ rpm or the first blade-passing frequency (1st BPF). The amplitude and strength of these fluctuations increased as the pump operated at off-design flow rate as shown in Figure 7. All other locations inside the pump experienced similar behavior [1].

The variation of pressure fluctuations with radial gap for the two impellers at location 4 is given in Figure 8, at different flow rates. For both impellers, the original gap of 2.5% of impeller diameter (3.6 mm) is a best compromise when the pump operates at the design flow rate. Reducing the gap increases the pressure fluctuation inside the pump. At off-design flow rates, increasing the gap decreases the pressure fluctuations. For Impeller-I, a gap of 6 mm gives the lower pressure fluctuation for all flow rates less than or equal to the design flow rate, and it is more effective at lower flow rates. Gap of 6 mm seems to be the optimum gap to be used with Impeller-I since the gap of 7 mm resulted in increasing the pressure fluctuation again. For Impeller-II, increasing the gap to 6 or 7 mm gives almost the same reduction in pressure fluctuations, which led to the conclusion that this range of gap values is compatible with Impeller-II.

A clearer picture of the effect of radial gap on pressure fluctuations is depicted by the FFT magnitudes, as represented by Figure 9. It is important to note that since the measuring locations are fixed, extending or cutting

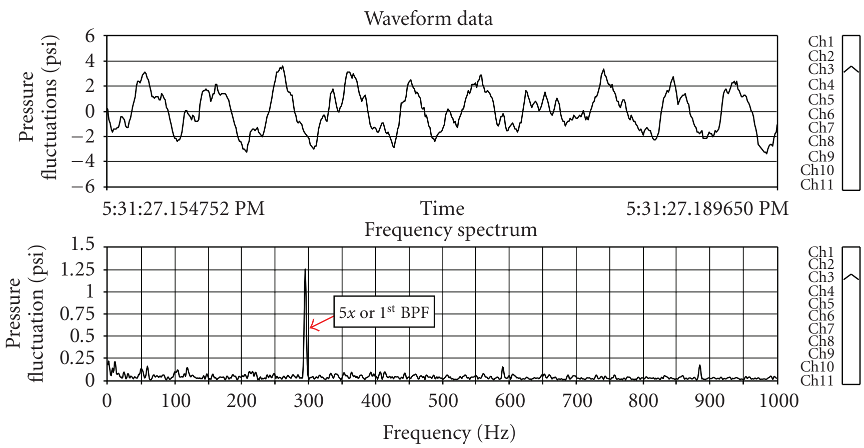
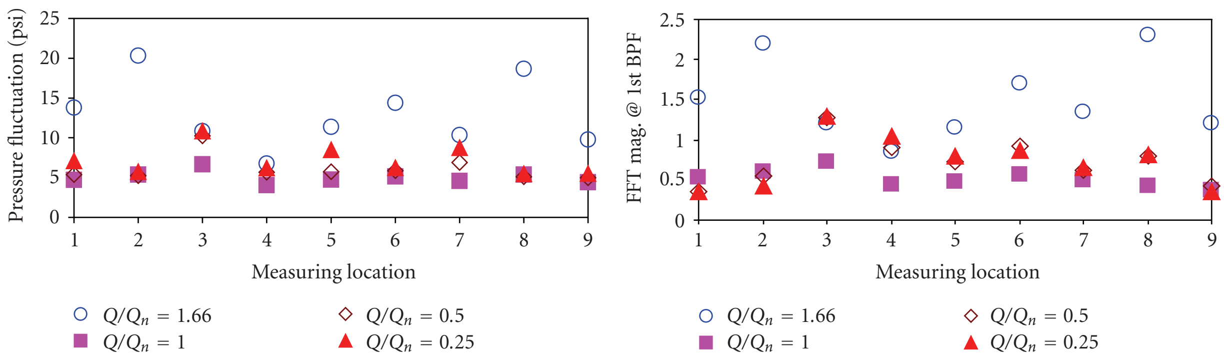


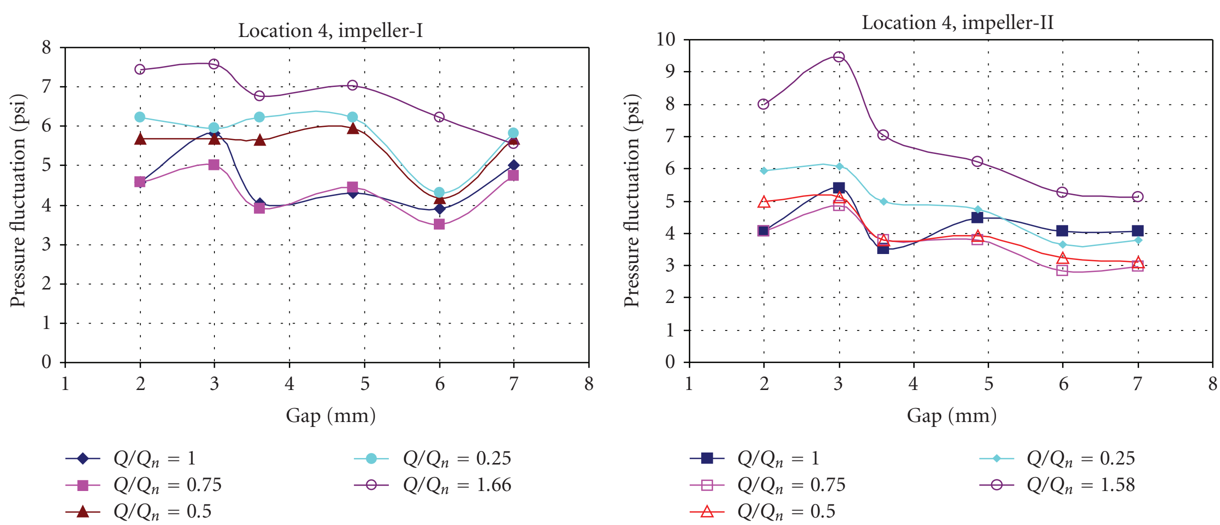
Figure 6: Waveform and frequency spectrum of pressure fluctuation at location 3, gap 3.6 mm for Impeller-I, at $Q = Q_n$.



(a) Pressure fluctuations (peak-to-peak)

(b) FFT magnitudes at 1st BPF

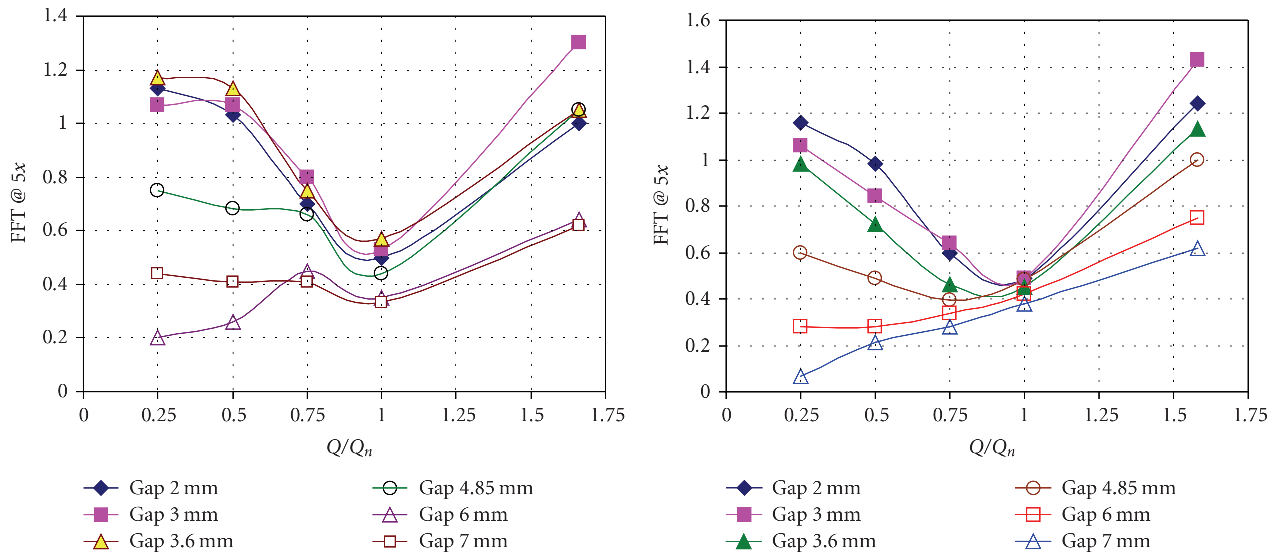
Figure 7: Effect of flow rate and measuring location on pressure fluctuations: impeller-I.



(a) Impeller-I

(b) Impeller-II

Figure 8: Variation of pressure fluctuations with radial gap at different flow rates.



(a) Impeller-I

(b) Impeller-II

Figure 9: Variation of FFT magnitudes of pressure fluctuations with radial gap at different flow rates at location 4.

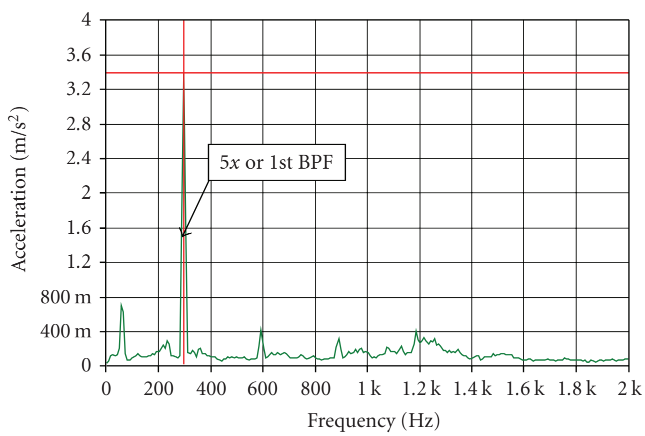
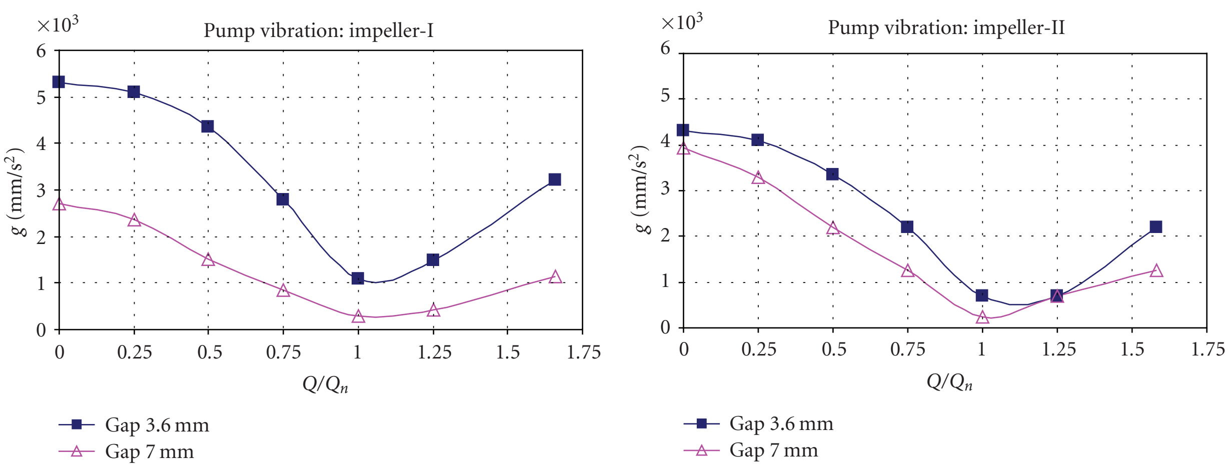


Figure 10: Typical vibration signal of model pump casing at $Q/Q_n = 0.5$.



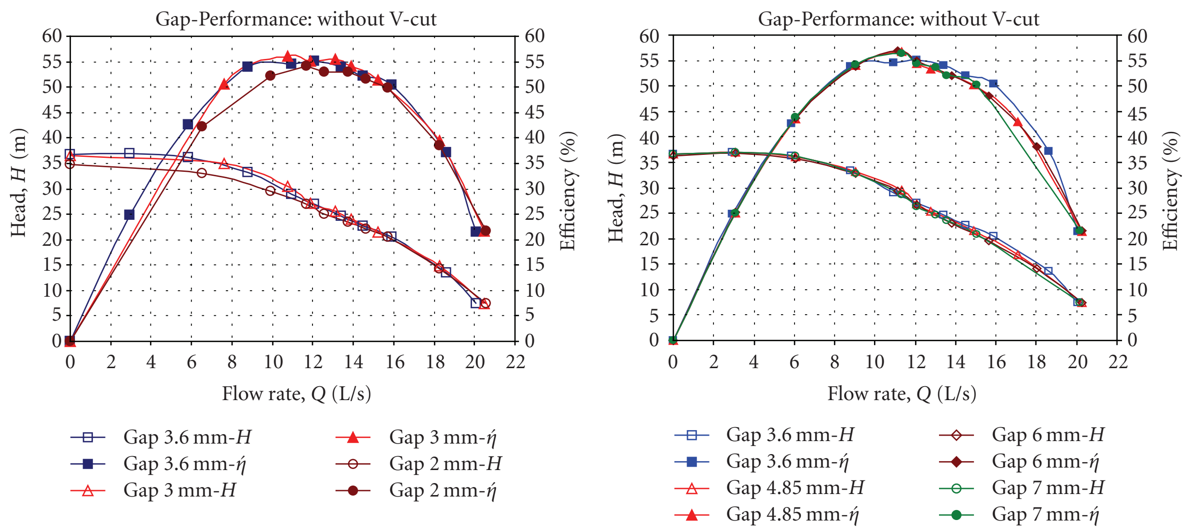
(a)

(b)

Figure 11: Effect of doubling the radial gap on vibration of pump casing.

Table 3: Effect of gap on pump performance, $Q = Q_n$.

Gap (mm)	Impeller-I		Impeller-II	
	% change in head	% change in efficiency	% change in head	% change in efficiency
2	−1.30	−1.88	0.74	0.74
3	0.55	0.11	−0.42	0.27
3.6	—	—	—	—
4.85	−0.15	−1.36	−2.92	−0.88
6	−0.93	−0.50	−4.99	−2.40
7	−2.41	−1.43	−5.51	−1.53



(a) Decreasing the gap

(b) Increasing the gap

Figure 12: Effect of gap on pump performance: impeller-I.

back the volute vanes changes relative locations of the measurement points around the impeller with respect to the pulsation source at the volute tongues. Reducing the gap increases the strength (energy content) of pressure fluctuations, and vise versa when increasing the gap. Moreover, increasing the gap decreases the FFT magnitudes with the reducing flow rate in contrast to the original gap of 3.6 mm. Figure 9 also shows that gap of 6 mm may be considered the optimum gap for Impeller-I, while a gap of 7 mm is optimum for Impeller-II. The behavior of measuring location 4 is quite general inside the pump.

The experimental findings attest to the fact that increasing the gap reduces the impeller-volute interaction and consequently the strength of pressure pulsation resulting from this interaction. Pumps, in general, are designed to operate at the best efficiency conditions. Changing the gap design shifts the operating point; usually to a lower head when the gap is increased. Therefore, the practical selection of the optimum gap should be based on the evaluation of performance loss in view of the reduction in pump vibration. It is a trade-off to select the most effective gap, which minimizes the pump vibrations while maintaining an acceptable performance according to pump application requirements.

A typical vibration signal (vertical direction) of the pump casing is shown in Figure 10 at flow ratio $Q/Q_n = 0.5$. The 1st BPF is clearly dominating the frequency spectrum. The same behavior was recorded earlier in Figure 6 for the pressure fluctuations inside the pump. The frequencies at which pressure fluctuations attained maximum peaks are the same at which the high-level vibrations were recorded on the pump casing. In either case, the maximum peak coincided with the first BPF. Such a finding has revealed more insight into this problem and consolidated the confidence that vibrations at the original BPF are primarily flow induced, as they are attuned with signal of the internal pressure pulsation.

Now, let us further examine the effect of changing the gap on pump vibrations. Increasing the gap reduces the pump vibration due to the reduction of the energy of pressure pulsation inside the pump. Figure 11 shows the effect of increasing the gap from 3.6 mm to 7 mm on the vertical vibration of pump casing, for the two impellers. For Impeller-I (without the V-cut) an average reduction of about 55% to 70% in pump vibration was obtained when the gap of 7 mm is used instead of the gap of 3.6 mm, depending on the flow rate. For Impeller-II, the existence of the V-cut actually increased the effective gap and an average reduction of about

50% in pump vibration was obtained when the gap of 7 mm is used instead of the gap of 3.6 mm.

The effect of different gaps on pump performance for impeller-I is shown in Figure 12. The figure was split into two parts for simplification. At the best efficiency conditions, all gaps gave comparable performance. Decreasing the gap did not improve the head or the efficiency except for very high flow rates. Smaller gaps are not recommended at lower flow rates while larger gaps are not recommended at higher flow rates. The original gap design of 3.6 mm (2.5% of the impeller diameter) gave good performance over a wide range of flow rates. However, larger gaps provided lower pressure fluctuations inside the pump, which result in reduction of the pump vibrations. Table 2 lists the best efficiency conditions, together with uncertainty limits, for the two impellers at 3540 rpm and the original gap of 3.6 mm (2.5%). The effect of different gaps on performance at a constant flow rate of $Q = Q_n$ for each impeller is given in Table 3. Based on the results of pressure fluctuations and pump performance at different gaps, gap of 6 mm for Impeller-I provides the minimum pressure fluctuation with acceptable performance lose of 0.92% in head and 0.5% in efficiency. This reduction in head and efficiency falls within the uncertainty limits of the measurements. However, if the pump operates at flow rates above 125% (above 15 L/s), increasing the gap would have more significant negative effect on the pump performance. For Impeller-II, the pump head drops appreciably for large gaps of 6 and 7 mm, exceeding the uncertainty limits. However, the efficiency change falls within the uncertainty limit and it is not easy to characterize the gap increase in this range.

4. Conclusions

Impeller-volute interaction is an important design factor in developing high-energy pumps like the double-volute boiler feed pumps. Different designs and combinations of impellers and gaps result in different characteristics of flow field inside the pump leading to different pump vibration behaviors. The flow-induced vibrations at the blade passing frequency stem from the pressure pulsation at the impeller exit. It was found that for the present pump, smaller gaps increased the pressure fluctuation and did not improve the performance except at very high capacities. The effect of the V-cut with different gaps on the pressure pulsation and the resulting pump vibrations at the blade-passing frequency was addressed for the first time. Different designs of impeller blade exit attained minimum pressure fluctuations inside the pump for different gaps. In general, it was concluded that increasing the gap reduces the pressure fluctuations particularly at part-load conditions. Another important conclusion, which is of particular interest to vibration analysts, is that severe vibrations at the BPF are most likely due to a design problem and can be corrected by the proper gap adjustment. Taking into account the similarity relationships, the results obtained from the experimental model pump were utilized to modify the gap in one of the actual BFP in the power plant. An average reduction of nearly 50% in pump vibrations at the BPF was obtained, as measured at both the inboard and outboard bearing housing. This was an excellent achievement, wherein the vibration levels were brought down below the severity limit. The results also serve as a testimony to the similitude accuracy of the constructed single-stage model pump, the proper selection and positioning of instrumentations, the data acquisition, and the test loop setup.

Acknowledgments

This research work was funded by the Saudi Electricity Company (SEC), Project no. CER-2289. The authors greatly appreciate the support provided by SEC and King Fahd University of Petroleum & Minerals (KFUPM) during this research. The valuable discussions and remarks made by Dr. R. Ben-Mansour and Dr. F. Al-Sulaiman are highly appreciated.

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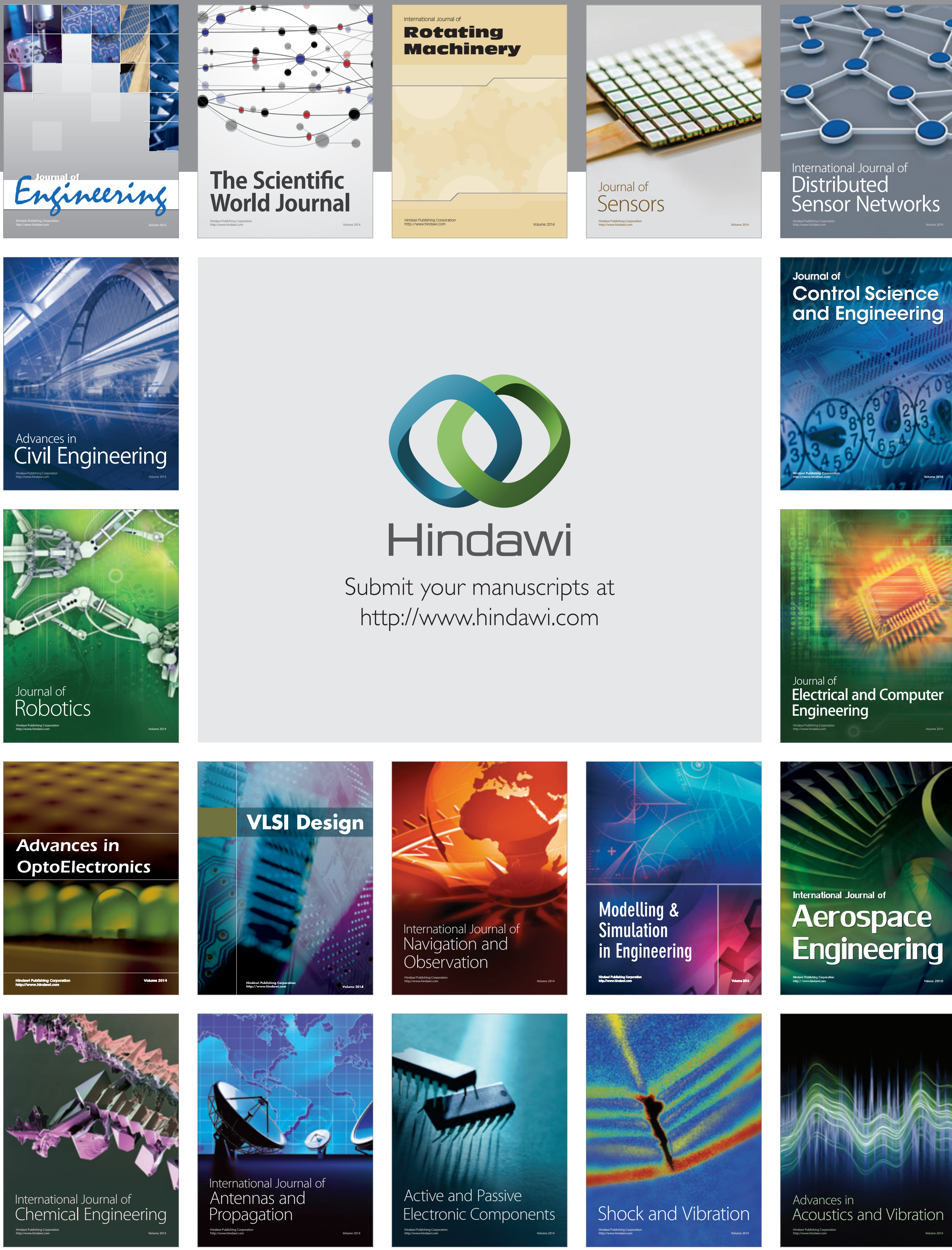
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