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Diagnosis of Centrifugal Pump Faults Using Vibration Methods

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Abstract. Pumps are the largest single consumer of power in industry. This means that faulty pumps cause a high rate of energy loss with associated performance degradation, high vibration levels and significant noise radiation. This paper investigates the correlations between pump performance parameters including head, flow rate and energy consumption and surface vibration for the purpose of both pump condition monitoring and performance assessment. Using an in-house pump system, a number of experiments have been carried out on a centrifugal pump system using five impellers: one in good condition and four others with different defects, and at different flow rates for the comparison purposes. The results have shown that each defective impeller performance curve (showing flow, head, efficiency and NPSH (Net Positive Suction Head) is different from the benchmark curve showing the performance of the impeller in good condition. The exterior vibration responses were investigated to extract several key features to represent the healthy pump condition, pump operating condition and pump energy consumption. In combination, these parameter allow an optimal decision for pump overhaul to be made [1].

1. Introduction

The current methods for monitoring (simple vibration analysis) are not adequate enough to predict incipient faults in a pump and avoid frequent breakdowns and outages that are causing the shutdown of large pumps. The condition of components such as pump shafts and impellers, roller bearings and drive parts is monitored by evaluation of specific machine vibrations, vibrations due to flow excitation, and structure borne sound in roller bearings. The examination also involves operating parameters such as the flow rate, suction pressure, output pressure, drive power, speed, bearing temperatures and leakage monitoring. Condition information is automatically transmitted to the system [2].To investigate the detection of incipient faults in the in-house horizontal centrifugal pumps, the focus was on the use of state-of the art technology/system of vibration analysis and electrical motor current signals for fault detection and diagnosis [3,4].

2. Vibration sources of a centrifugal pump

2.1 Working mechanism of a centrifugal pump

A centrifugal pump consists of the rotating elements (e.g. pump impeller and shaft) and stationary elements (e.g. electrical motor and associated cooling fan, casing box and bearings). The vibration of

the working pump is generated by both mechanical and hydrodynamic sources. The mechanical sources are invariably generated by rotation of unbalanced masses and friction in the bearings. Hydrodynamic vibration is due to fluid flow perturbations and interaction of the rotor blades particularly with the volute tongue and/or guide vanes. The generated vibration will cause the pump surface to vibrate which will then act as a loudspeaker radiating airborne noise. Thus the basic mechanisms generating both structure borne vibration and airborne noise are the same. Chudina et.al [5] have shown that pump vibration contains both broadband noise and discrete frequency peaks.

The broadband content is ascribed to flow turbulence and vortex shedding particularly in the narrow spaces between the pump rotor and adjacent stationary parts of the casing. There will also be a contribution from such mechanical sources as the rotation of the pump shaft and bearings. Turbulent noise will be at a minimum when the pump works with maximum efficiency, i.e. the pump operates at its design point. Off-design flow rates generate additional hydraulic noise, particularly for very low flow rates when internal recirculation occurs between the suction and discharge areas of the pump. When the flow rate is greater than the design flow rate, flow turbulence and boundary layer vortex shedding both increase [6,7]. The discrete components in the pump spectrum are primarily due to interaction of the rotor blades with adjacent stationary objects (e.g. volute tongue) in the flow due to the discrete nature of the pump's rotor blades. These mechanisms generate spectral peaks at the rotational frequency (RF) and/or blade passage frequency (BPF) of the pump and their higher harmonics. At or near the design point flow turbulence is at a minimum so the discrete components, particularly the lower harmonics, tend to dominate the spectrum [8]. Away from the design point the turbulent noise will increase and will, eventually, even exceed the tonal noise [5,7].

2.2 Vibration sources

All machinery with moving parts generates mechanical forces during normal operation. As the mechanical condition of the machine changes because of wear, changes in operating environment, load variations etc., so do these forces. The vibration profile that results from motion is the result of a force imbalance – there is always some imbalance in real-world applications [9]. Causes of vibrations are of major concerns because of the damage to the pump and piping that generally results from excessive vibration. Excessive vibration in a pump may be a result of improper installation or maintenance, incorrect application or hydraulic interconnections with the piping system [10]. Vibration can be defined as simply the cyclic or oscillating motion of a machine or machine component from its position of rest. Forces generated within the machine cause vibration. These forces may:

1. Change in direction with time, such as the force generated by a rotating imbalance.

2. Change in amplitude or intensity with time, such as the unbalanced magnetic forces generated in an induction motor due to an unequal air gap between the motor armature and stator.

3. Result in friction between rotating and stationary machine components in much the same way that friction from a bow causes a violin string to vibrate.

4. Result from impacts, such as the impacts generated by the rolling elements of a bearing passing over flaws in the bearing raceways.

5. Result from randomly generated forces such as flow turbulence in fluid-handling devices such as fans, blowers and pumps; or combustion turbulence in gas turbines or boilers [11].

When the internal pressure in a fluid reaches its vapour pressure, cavities form in the low-pressure regions, these collapse when they reach a place of higher pressure in the pumping system. This phenomenon takes place in a short time in a working centrifugal pump [12,13,14].

Cavitation can also cause pump vibration. It occurs when the system's available net-positive suction head (NPSHa) is less than the pump's requirement (NPSHr). Indications of a cavitating pump can include noise, fluctuating flow rates, a decrease in discharge pressure, and vibration. This induced vibration is caused by imploding vapor bubbles that introduce shock waves into the pump, shortening the life cycle of all of the pump's mechanical components. The components that commonly fail prematurely are impellers, wear rings and casings. If cavitation is present, NPSHa should be increased

above NPSHr by making changes in the system design or operation to reduce or eliminate cavitation. Cavitation does not always produce pump vibration, and the induced vibration often is random and unmeasurable [15]. One of the other causes of flow-induced vibration at blade passing frequencies in centrifugal diffuser pumps is the inappropriate radial gap between impeller and volute vanes. A small gap may be preferable for pump performance, head and efficiency. However, it may initiate strong impeller/volute interaction, resulting in high pressure pulsation inside the pump and consequent high vibrations to the pump components [16].

2.3 Vibration changes with operating conditions

With a worn impeller the loss in discharge pressure is associated with a loss of pump efficiency, therefore the power consumed by the pump will either remain the same, or increase as wear occurs. With wear, operational records will indicate a gradual change in performance over some period of time. Partially blocked pathways will usually be evidenced by a pump that delivers full discharge pressure at shut-off, with a sharp drop in discharge pressure as flow is increased. The drop in discharge pressure is often accompanied by increased vibration as the flow restriction starts to cause cavitation. If vibration is also present at shut-off the blockage may be in the impeller causing a physical imbalance [17]. Rotating stall is a flow instability occurring in most type of centrifugal pumps when the flow rate is reduced below the design value. Aside from the mechanical vibrations which can be induced by stall, the generated acoustic noise can also be an important problem [18].

Resonance conditions can cause excessive vibration levels, which in turn are potentially harmful to equipment and environment. Pumps, their support structure, and piping are subject to a variety of potential structural vibration problems (resonance conditions). Fixed-speed applications often miss these potential resonance situations because the common excitation harmonics due to running speed, vane passing frequency, plunger frequency, etc., do not coincide with the structural natural frequencies. For variable speed drive applications, the excitation frequencies become variable and the likelihood of encountering a resonance condition within the continuous operating speed range is greatly increased. Pump vibration problems typically occur with bearing housings and the support structure (base plate for horizontal applications, motor and stool for vertical applications [19].

3.   Test rig measurements

The test rig consists of a centrifugal pump, variable speed motor and the closed-loop water piping system for water circulation. A photograph of the finished test-rig is shown in figure 3.1.

The pump model is a F32/200A series centrifugal pump standardised to DIN 24255. The centrifugal pump was used in the test-rig shown in Fig. 3.2. It was a single suction, single stage, end/top discharge, closed impeller and closed-coupled centrifugal pump, which can deliver water at a rate of up to 30m3h-1 (500l/min) at a head of up to 55m. It is driven by a three-phase electric motor running at 2900rpm on 9.5A at 380-400V (nominal 4kW/5.5hp). The capacity of the tank is based on the maximum flow rate.

Figure 3.1. Photograph of the test-rig.

Figure 3.2. The test-rig pump.

Table 3.1. Specifications of the accelerometers.

Maker	Bruel & Kjaer
Type	Accelerometer (piezoelectric)
Model	YD3 8131
Frequency range	200Hz – 20KHz
Sensitive	1.51mv/ms-2
Range	<2000ms-2
Temperature range	To 2500C

For monitoring pump performances, a shaft encoder was used to measure pump speed. A flow sensor was installed in the discharge line. In addition, two pressure sensors were installed in the suction and discharge lines respectively for pump delivery head measurement.

The predicted characteristics between the NPSHa and NPSHr for this system were obtained by throttling the valve in the discharge line progressively while the pump speed is at 2900rpm and the valve in the suction line was fully open (100%) as suggested in ISO 3555. Figure 3.3 shows the relation between NPSHa, NPSHr and flow rate. NPSHr (from pump head) increases with flow rate and increases rapidly above 220 l/m. The measured NPSHa decreases with flow rate and the NPSHa-NPSHr intersection occurs at a flow rate of approximately 320 l/m and head 5.75(m).
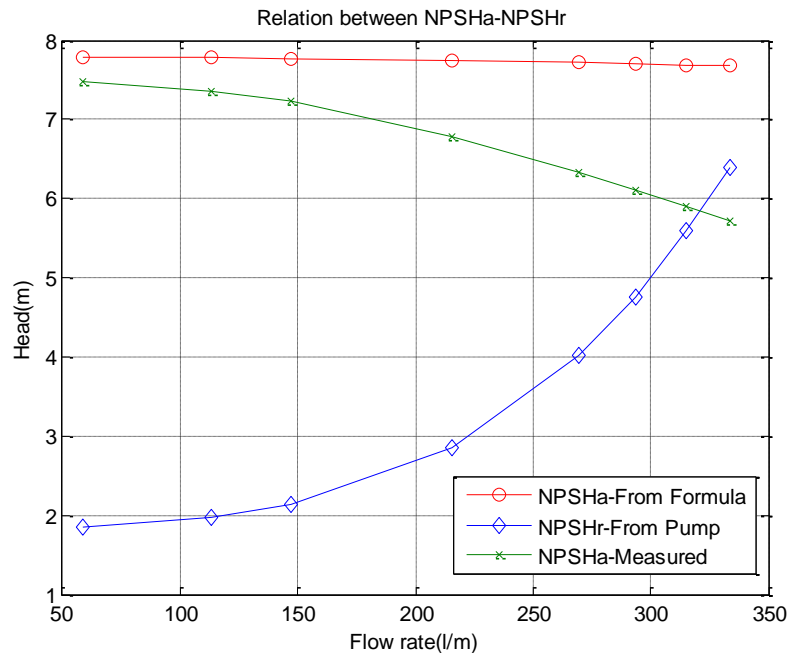
Figure 3.3. Pump performance.

Five tests were conducted to study the impellers fault detection and diagnosis, each with the same speed but with different flow rates. The sampling rate was 96,000Hz. Seven channels of various signals were collected see Table 3.2, one of them from the accelerometer which was mounted on the pump housing in a horizontal direction. All tests measurements were conducted under full pump speed.

Table 3.2. Channels and Parameters.

Channel	Parameter
Channel 1	Speed
Channel 2	Flow rate
Channel 3	Suction Pressure
Channel 4	Discharge Pressure
Channel 5	Vibration
Channel 6	Phase current
Channel 7	Differential phase voltage

4. Results and discussion

To extract useful features from the measured various signals, the data of the signals were analysed using a Matlab program.
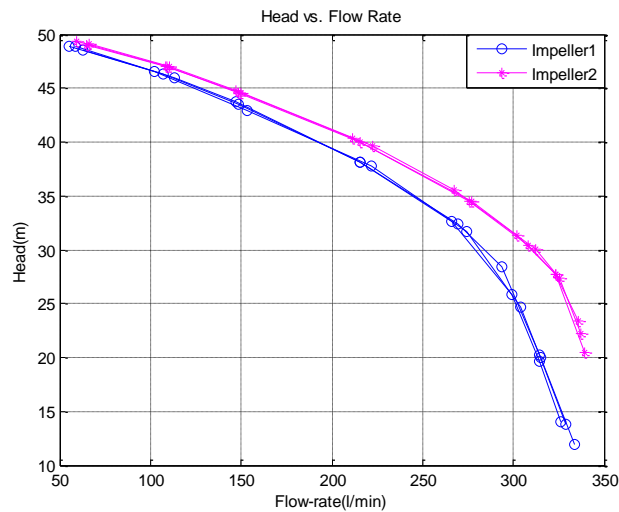
4.1 Comparison Test 1 – Test2

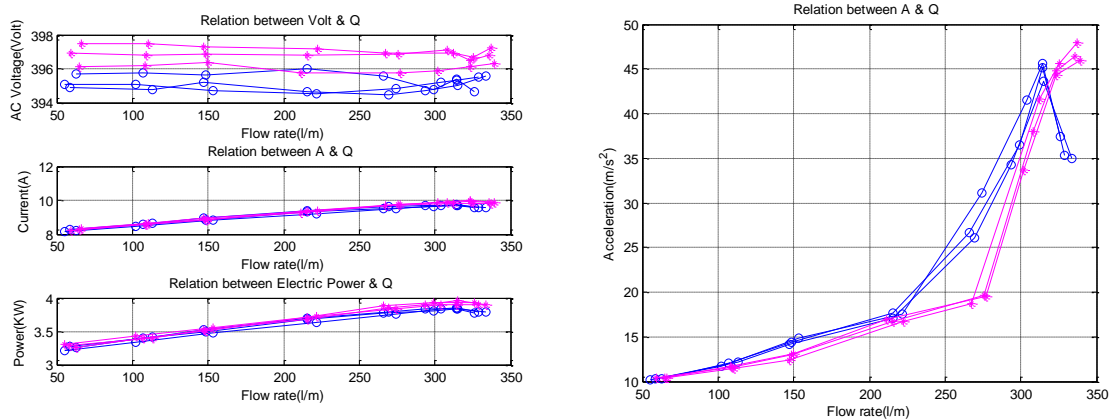**Figure 4.1**. Impeller 1 In good condition.

Figure 4.2. Impeller 2 with gap between two plates.

(a) Head vs flow rate.

(b) Voltage, current, power vs flow rate.   (c) Vibration acceleration vs flow rate.
Figure 4.3. Collected data at different flow rates (impeller 1 vs impeller 2).
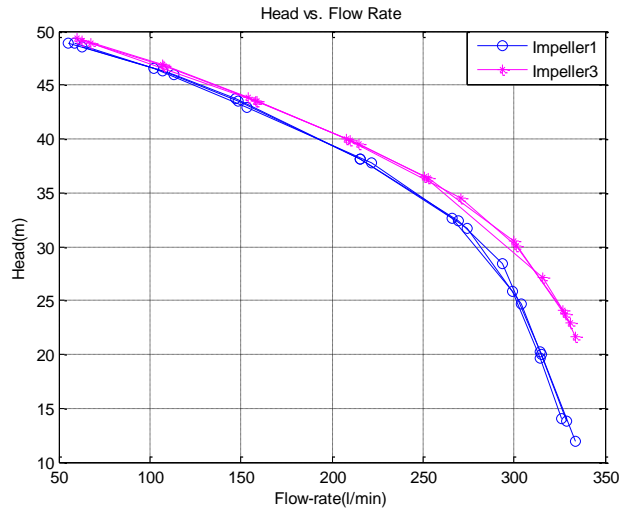
Figure 4.3 (a) compares flow rate against head the impeller in good condition (blue line) with defective impeller 2: a hammer blow on its edge reduced the gap between two plates at that point to 2.4mm (red line). The healthy impeller had a gap of 3.2mm. Impellers 1 and 2 are the same geometry. The red line shows a higher head compared to the blue line and the deviation is greater at higher flow rates. In Figure 4.3 (b) the red line shows that the faulty condition consumes more voltage than the healthy condition. The current consumption is almost the same for both conditions. Power consumption is higher for the faulty condition particularly for flow rates higher than 250 l/m. Vibration acceleration is generally greater for the faulty condition for flow rates between 100 l/m and 320 l/m, as can be seen from the blue and red lines in Figure 4.3 (c). Above a flow rate of 320 l/m the situation is reversed and the measured vibration acceleration is greater for the healthy pump. The vibration acceleration for the faulty pump is almost linearly proportional to flow between 50 to 220 l/m and then increases rapidly up to 320 l/m flow rate after which it decreases rapidly. The vibration acceleration for the healthy pump increases almost linearly between flow rates of 50 and 275 l/m then increases rapidly.
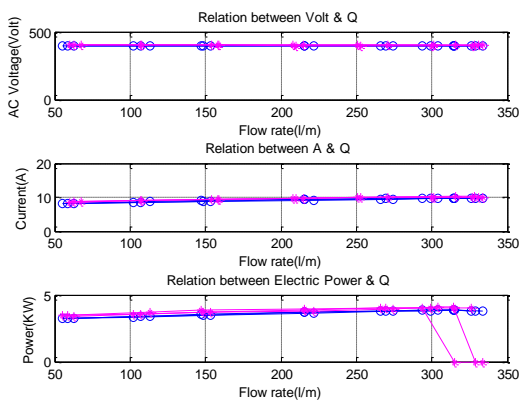
4.2 Comparison Test 1 – Test3

Figure 4.4. Impeller 3 with gap between two plates reduced to 0.7mm at a single point by a hammer blow.
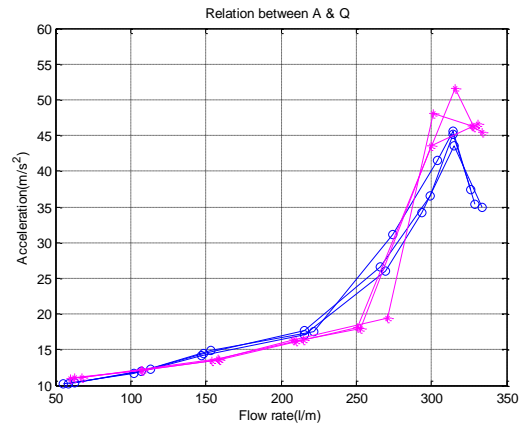
(a) Head vs flow rate.

(b) Voltage, current and power vs flow rate.

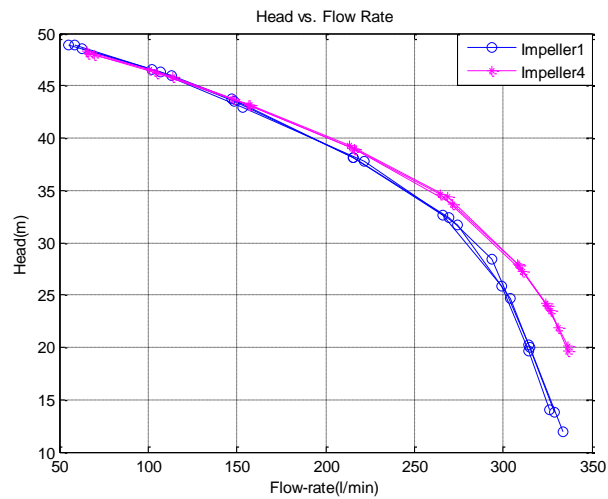(c) Vibration acceleration vs flow rate.

Figure 4.5. Collected data at different flow rates (impeller 1 vs impeller 3).

Figure 4.5 (a) compares flow rate against head the impeller in good condition (blue line) with defective impeller 3: a hammer blow on its edge reduced the gap between two plates at that point to 0.7mm (red line). Impellers 1 and 3 are the same geometry. Figure 4.5 (a) compares the healthy impeller (blue line) with the defective impeller (dented and with reduced gap between two plates (red line). Once again the faulty impeller had a higher head than the healthy impeller at all flow rates, but particularly above about 300 l/m. Figure 4.5 (b) shows the voltage remains almost the same across flow rates from 50 to 325 l/m, but current and power increase slightly with flow rate. Figure 4.5 (c) shows the vibration acceleration is almost the same in both healthy and faulty impeller for flow rates up to about 220 l/m; for flow rates 220 to 275 l/m the vibration acceleration is greater for the healthy condition, but above 275 l/m the defective impeller produced a greater level of vibration. The vibration acceleration for the defective impeller is almost linearly proportional to flow rate between 50 to 220 l/m, then increases rapidly up to 320 l/m then decreases.
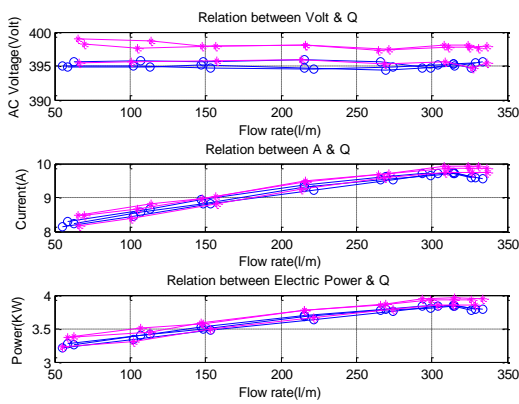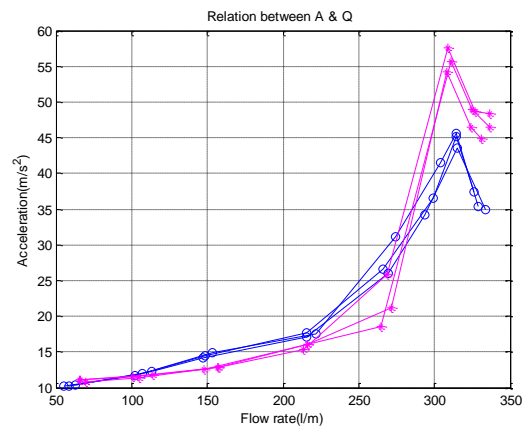
Figure 4.6. Impeller 4 with gap between two plates reduced to 0.3mm at a single point by a hammer blow.

(a) Head vs flow rate.

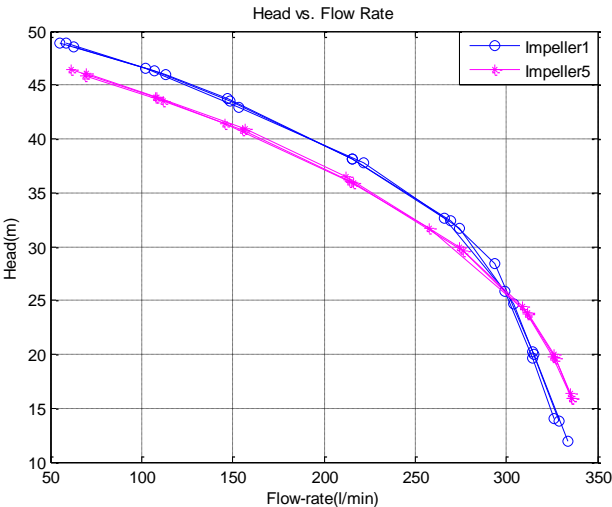(b) Voltage, current and power vs flow rate. (c) Vibration acceleration vs flow rate.

Figure 4.7. Collected data at different flow rates (impeller 1 vs impeller 4).

Figure 4.7 (a) compares flow rate against head for the healthy impeller (blue line) with defective impeller 4: a hammer blow on its edge reduced the gap between two plates at that point to 0.3mm (red line). Impellers 1 and 4 are the same geometry. As previously, the faulty impeller provides a higher head compare to the healthy impeller on flow rate above about 150 l/m, the difference is very marked at flow rates above 300 l/m. Red consumes more AC volt compared to blue in Figure 4.7 (b), where current consumption is slightly higher in Red compared to Blue. Power consumption is high in Red compared to blue. In figure 4.7 (c), acceleration is found to be more in blue compared to red for the flow rate 100-275 (m/l). Above 275 (m/l) flow rate, acceleration is more in red compared to Blue. Acceleration in blue is almost linearly proportional between 50 to 220 (m/l) flow rate then, increases rapidly up to 320 (m/l) flow rate and after that decreases rapidly. Acceleration in red is almost linearly proportional between 50 to 275 (m/l) flow rate then, increases rapidly up to 315 (m/l) flow rate and then increases rapidly.
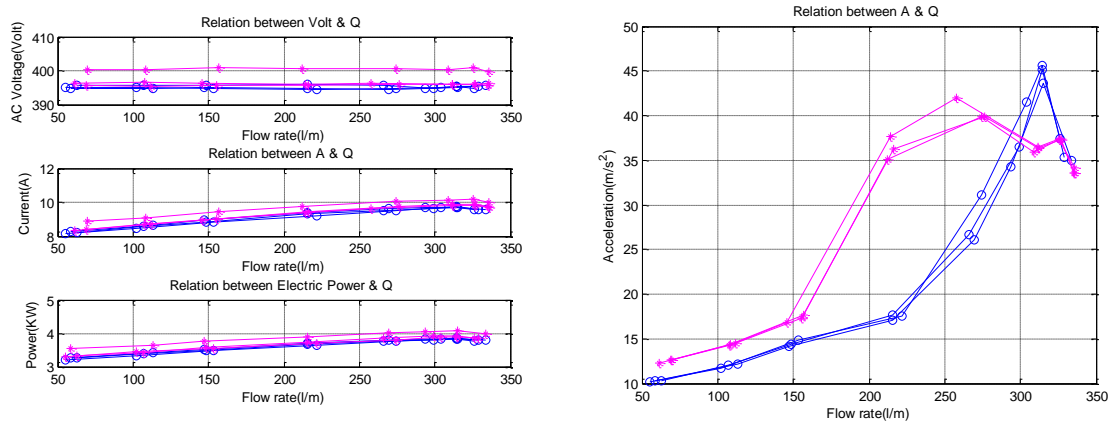
4.4 Comparison Test 1 – Test5

Figure 4.8. Impeller 5 with gap between two plates reduced by a hammer blow at three equally spaced points.

(a) Head vs flow rate.

(b) Voltage, current and power vs flow rate.   (c) Vibration acceleration vs flow rate.

Figure 4.9. Collected data at different flow rates (impeller 1 vs impeller 5).

Figure 4.9 (a) compares flow rate against head for the healthy impeller (blue line) with defective impeller 5: three equally spaced hammer blows around the circumference reduced the spacing to 2.6mm, 2.9mm and 3.0mm, respectively, as shown. Impellers 1 and 5 are the same geometry. Unlike the previous results, the healthy impeller produces a greater head up to about 300 l/m, for flows above this the defective impeller produced a higher head. Figure 4.9 (b) shows that the pump containing the faulty impeller consumes more power, requiring a higher AC voltage and current than the healthy pump. Figure 4.9 (c) shows that vibration acceleration is much greater in the pump with defective impeller for flow rates up to about 280 l/m between about 300 l/m and 325 l/m the vibration acceleration is greater for the healthy pump, but above 325 l/m the acceleration is much the same for both pumps.

5.   Conclusions

Experiments were carried out on a high-pressure centrifugal pump (single stage) to study the effect on pressure fluctuations due to different impellers faults at different flow rates. The effect on pump performance is also examined. The impeller/volute interaction is an important design parameter in developing high-energy pumps. The results for all tests show that the vibration level increases with increased of flow rate and with different readings, this is due to a different type of defect on each impeller. The experimental study has shown that data obtained from impellers with different gaps are different, even though the impellers are geometrically similar and for the same pump. This is because impellers 2, 3, 4 and 5 have different faults – both the "depth" of the dents and the number of dents. Further research work will be conducted to extract more data/features from the pump using different techniques. The pump faults may then be identified using these data/features.

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