Monitoring of Cavitation in Centrifugal Pumps using Spectral Entropy of Vibro-acoustic Measurements

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Abstract

Cavitation in centrifugal pumps causes damages to pump components and produces high levels of vibration and noise, which not only reduces pump performance but also consumes additional energy. Unfortunately, many pumps operate under a certain degree of cavitation for a number of reasons such as varying operating conditions, inadequate installation and harsh environments. To evaluate the degree of cavitation online and to take necessary actions at an early stage, this research focuses on developing a cavitation measurement technique using non-intrusive vibro-acoustic techniques. In this paper, conventional vibro-acoustics measurements are examined with different dimensionless parameters for characterizing the signals and hence for cavitation diagnosis. Conventional parameters such as peak factors and kurtosis from both the time domain and frequency domain have been evaluated to be inefficient for indicating cavitation in different stages. However, the spectral entropy has been found to be more accurate in presenting the cavitation. Especially, the spectral entropy from airborne acoustics can yield a better diagnostic result than the surface vibration.

Keywords: Cavitation, Centrifugal pump, Condition monitoring, Spectral Entropy, Vibro-acoustics.

1. Introduction

Centrifugal pumps are by far the most commonly used pump in industry for fluid delivery. All pump manufacturers provide characteristic curves showing pump performance in terms of discharge flow rate and pump head. Usually details of pump power and operating efficiency will be included. The BEP is the ideal pump operating point, it is where pump capacity and pump head combine to give maximum pump efficiency. Away from the BEP the pump will be subject to increased wear and reduced operational life. Typically, NPSH data will be provided on the pump inlet pressure at which serious cavitation begins to occur. Cavitation often begins at about a 3% drop in head from BEP, but this value will vary with the physical properties of the liquid being pumped and the surface roughness of the particular hydraulic equipment [(1)] Increase in temperature will make the system more susceptible to cavitation, as will undesirable flow conditions caused by a sharp elbows or obstructions in the suction line immediately before the pump.

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Airborne acoustic noise measurements on centrifugal pumps have been carried out by [1, 2,3]. Both investigations found that cavitation noise was clearly discernable in any analysis of the measured sound pressure. Cavitation noise exceeded that of the pump, motor, and other fluid borne noises, by as much as 20dB, and the increase in the high frequency noise content was especially noticeable. Upstream and downstream pressures were varied as were solid and gaseous contaminant concentrations. For increasing amounts of both solid and gaseous contaminants in the fluid being pumped, cavitation inception occurred sooner than with uncontaminated fluid.

A number of authors [4,5,6,7] have investigated and discussed the main feature of the vibration signal of centrifugal pumps and turbines under cavitation. They confirmed what had been previously reported, that the vibration signal from centrifugal pumps experiencing cavitation was broadband with a pronounced high frequency content and containing discrete frequency components. The upper limit to the frequency spectrum was determined more by the upper frequency limit of the transducer then the source.

This paper focused on developing dimensionless parameters for cavitation monitoring. In particular, it explores the use of an entropy spectrum method to identify the onset of cavitation. This new method is used to estimate the onset of pump cavitation online and predict possible damage evolution and energy loss in the pump system.

2. Characteristics of Pump Vibro-acoustics

A centrifugal pump has two main parts: the rotating element which consists of a shaft and an impeller and the stationary element which consists of the casing, casing box and bearing, and electrical motor which has a cooling fan. Based on pump working process, its vibro-acoustics can be understood to be generated by both hydrodynamic sources and mechanical sources. The hydrodynamic sources are usually caused by fluid flow perturbation in the pump and interaction of the rotor blades with nearby stationary objects such as the volute tongue or guide vanes. In the mean time mechanical sources are caused by vibration of unbalanced rotating masses and friction in bearing and seals.

These mechanisms of generating vibration will cause the structure of the pump to vibrate which will then radiate airborne noise into the external environment. From this point of view the basic generating mechanisms for both structure borne vibration and airborne noise are the same for a sealed pump system.

Previous investigation [8] shows that the content of vibration contains both a broadband noise and a number of discrete frequency peaks. The broadband content is due to pressure fluctuations generated by flow turbulence, viscous forces, boundary layer vortex shedding, boundary layer interaction between a higher-velocity and lower-velocity regions of the process fluid, and by vortices generated in the clearances between the rotor of the centrifugal pump and the adjacent stationary part of the casing. There will also be a contribution from mechanical sources, e.g. noise from the rotation of the pump shaft and bearings.

Turbulent noise depends strongly on the flow conditions: if the pump operates at the design point (the operating point at which a maximum proportion of energy is transferred to moving the process fluid) it will have a minimum value. Off-design (flow rate less than the design flow rate) additional hydraulic noise is created due to internal recirculation in the suction and discharge areas of the pump impeller and the overall noise of the pump increases. Off-design (flow rate more than the design flow rate) boundary layer vortex shedding increases, flow turbulence increases and additional hydraulic noise is generated [4, 9].

The discrete component characteristics present within the overall spectrum are due mainly to the interaction of the rotor blades with nearby stationary objects such as the volute tongue and periodicities in the flow due to the discrete nature of the pump's rotor blades. These two mechanisms generate discrete components at the rotational frequency (RF) and/or blade passage frequency (BPF) of the pump and their higher harmonics. Because the turbulent noise at or near the best efficiency point is at a minimum, the discrete components tend to dominate the measured spectrum and the lower harmonics ≤ 3[2] are more distinctive. With operation away from the best efficiency point the turbulent noise will increase and can even exceed the tonal noise [5,8].

Because of these characteristics, pump cavitation is expected to be diagnosed at different stages by measuring the vibration and acoustics externally.

3. Experimental Facility

For evaluating of the theoretical predications and hence for diagnostic study, a pump test rig was constructed to simulate pump cavitation at different degrees. As shown in Figure 1, the rig consists of an integrated centrifugal pump, a water tank, a suction line and discharge line. This construction forms a closed loop for water circulation.

The pump is a closed impeller, single stage general purpose 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 so that water temperature in the system can be maintained at 1ºC for more than one hour during which full measurements can be completed at different flow rates.

To simulate different operational conditions, throttling valves are installed in both the suction and discharge lines. In addition, each line has a part of transparent pipe for visualization of cavitation bubbles.

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

According to ISO 3555, the predicted characteristics between the Net Positive Suction Head Available (NPSHA) and Net Positive Suction Head Required (NPSHR) for this system are obtained by throttling the valve in the discharge line progressively while the pump speed is at 2900rpm and the valve in the suction line is fully open (100%). Figure 2 shows that pump cavitation begins at flow rate of 249 l/min and predicts cavitation will occur at about 344 l/min when the head is 6.92m, where the NPSHR is higher than the NPSHA. During the test study, it was found that both vibration and acoustics have a clear change in amplitudes and frequency contents at the flow rate 249l/min forward. It thus considered the flow range from 249l/min to 344l/min as the cavitation progression phase in which the cavitation become severer as the flow rate increases. In the mean time, flow rate higher than 344l/min will cause a full cavitation as suggested in ISO 3555.

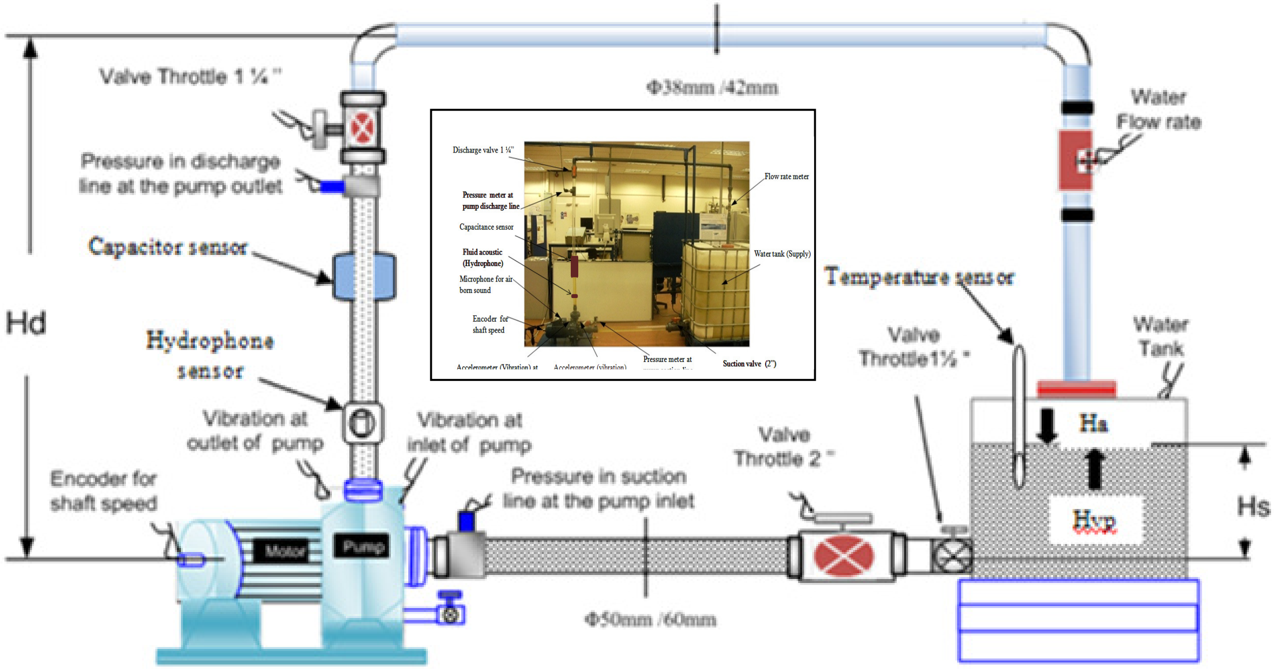

Figure 1 Schematic of pump test system

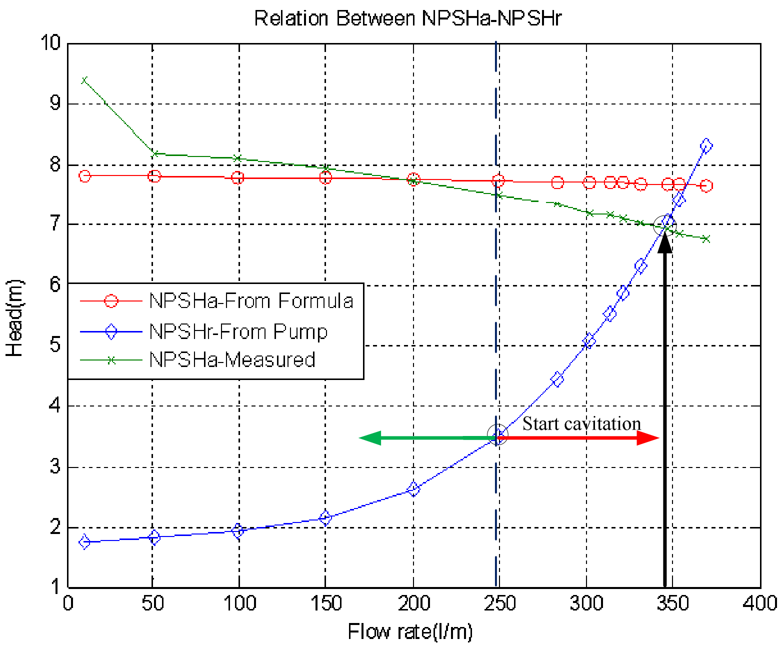

Figure 2 Performance pump curve characteristics

4. Vibro-acoustic Measurement

The vibration of the pump is measured using two identical types of accelerometers with a flat frequency band from 20Hz to 10kHz and a resonant frequency is about 40kHz. Because of the wide frequency band, it allows the high frequency pump vibration to be measured. The two sensors are placed at the pump case of inlet and outlet respectively for comparison study. During the investigation it was found that the accelerometer at suction line is less sensitive to cavitation. Only the vibration from the discharge line is discussed.

The acoustics of the pump are measured using a microphone for airborne noise and a hydrophone for fluid acoustics. Both of these transducers have a frequency band from 20Hz to 20kHz and wide dynamic range. They allow the acoustics to be measured with high accuracy. The microphone is placed 50mm from the outlet of the pump (between pump casing and back cover) and the hydrophone is placed in the discharge line at the outlet of the pump. These sensor positions are thus consistent with that of the vibration accelerometer and thus the results can be compared directly.

A high speed data acquisition system (DAS) is used to record vibro-acoustics in to a host PC hard drive. In the mean time flow rate, suction pressure, discharge pressure, and shaft speed data are also measured. The data acquisition has 8 channels with 16 bit resolution and a sampling rate of 100kHz in each channel with built-in anti-aliasing filter.

The vibro-acoustics were measured under different flow rates but at a fixed speed of 2900 rpm. The flow rate is adjusted by a throttling valve in the discharge line step by step. Each test has at least at least 15 data records which cover a flow rate range from 370l/min to 10l/min. To obtain a reliable result, this test was repeated at least 3 times with a time interval between each test is sufficiently so that the water temperature for each test is the same. The data records from each test were processed in Matlab to characterise signals and hence to find a set of consistent parameters for cavitation diagnosis i.e. for determining the progression and full phase of cavitation based on the vibro-acoustic measurements only.

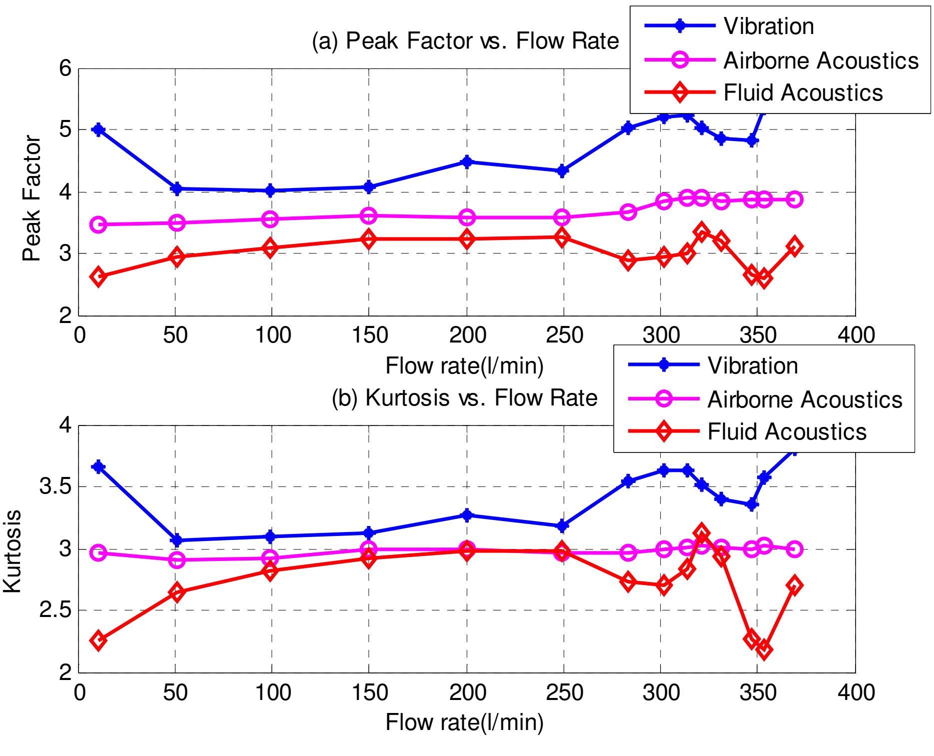

Figure 3 Conventional dimensional parameters vs. flow rate

5. Vibro-Acoustics Analysis

For comparing the performance of vibro-acoustic based monitoring and easy implementation in on–line monitoring, only dimensionless parameters are developed from vibration, airborne acoustics and fluid acoustics. Especially, the fluid acoustics is examined to examine the influences of background noises on the airborne signals and the surface vibration.

Two conventional statistical parameters: peak factor and kurtosis are calculated from raw data sets to examine their performance in charactering cavitation. As shown in Figure 3, their trends oscillate highly with flow rates and hence it is difficult to use either of them to identify the start of cavitation and to assess the severity of cavitation. Therefore, it is concluded that these conventional dimensionless parameters cannot produce a consistent result for on-line cavitation detection.

Therefore, the data sets are explored in the frequency domain. Figure 4 shows the spectra for the three types of measurement under three typical flow rates that corresponds to no cavitation, progressive cavitation and full cavitation. It can be seen in the figure that both vibration and airborne acoustics show a clear change in spectra with different flow rate acoustics. In particular, vibration spectra show that there is a steady increase in amplitude above frequency 2kHz whereas amplitude in the low frequency range is in a reverse. This is consistent with the vibration generation processes because high cavitation will produce more random noise. In the mean time, the discrete components will be reduced due to the random fluid oscillations. Similarly the airborne acoustics show a gradual increase in the high frequency range. However, fluid borne acoustics exhibits a less clear change with the flow rates. This may be due to the reflections of wave propagation. To characterize these changes in spectra for cavitation diagnosis the peak factor and kurtosis are also calculated from the spectral sequences.

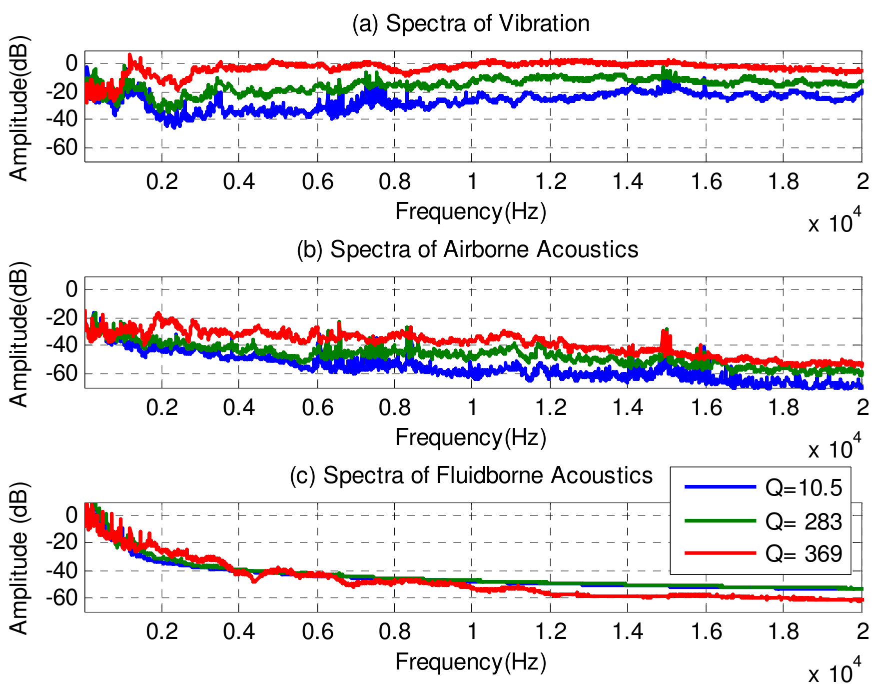

Figure 4 Spectra of vibration, airborne acoustics and fluid borne acoustics

Figure 5 shows the values of peak factor and kurtosis from the spectral sequences of three measurements. Both of the two parameters show very similar characteristics with flow rates. However, the peak factor and kurtosis from airborne acoustics and fluid borne acoustics exhibit too many oscillations. Obviously, these two are not suitable for determining the progression of cavitation.

In contrast, vibration trends of both peak factor and kurtosis exhibit a relatively smooth trend. Even though both peak factors and kurtosis are slightly higher in the flow range between 100l/min and 250l/min, the overall trends show a gradual decreasing trend and thus it is possible to use them for cavitation diagnosis.

In addition, the two trends from the vibration and the airborne acoustics show a large difference. This is not consistent with the spectra characteristics shown in (a) and (b) of Figure 4. It thus indicates that these conventional parameters may not so effective in charactering the spectra and new parameters need to be developed.

To obtain a more consistent trend, a relatively new parameter, named spectral entropy, is applied to the data sets. Originally, spectral entropy is used as a generalization of information entropy and was firstly proposed by [10] to measure information uncertainty. For a continuous random variable *x* having probability density function *f(x),* information entropy *H(x)* is defined as:

$$H(X) = -\int f(x)\log_2(f(x))dx \quad (1) \qquad (1)$$

and often used widely as a measure of disorder, unevenness of distribution, the degree of dependency, or complexity.

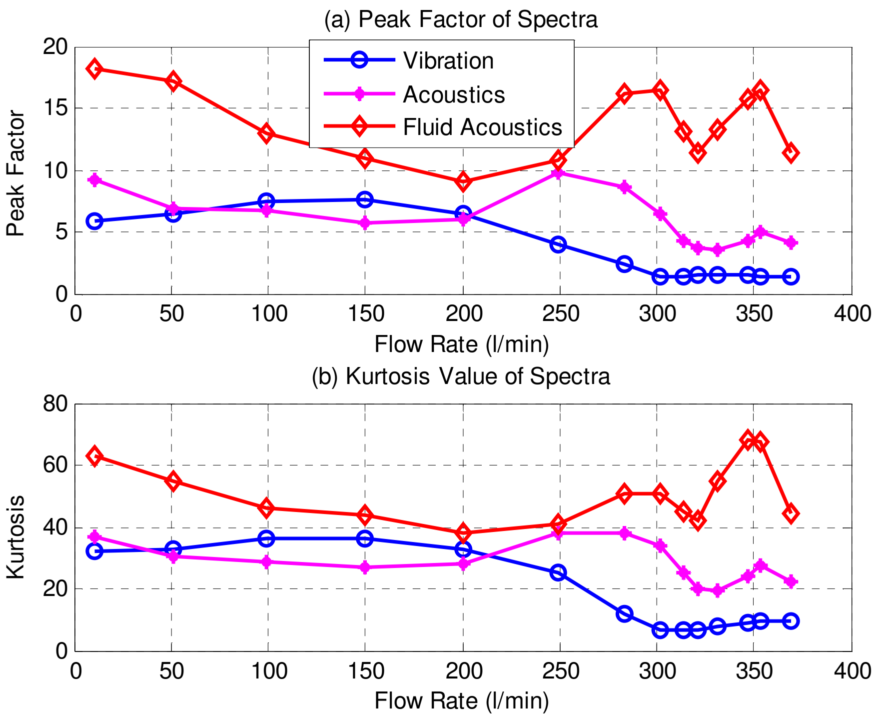

Figure 5 Peak factor and kurtosis from vibration, airborne acoustic and fluid acoustic spectra

In signal processing, a spectral entropy has been introduced [11, 12] to measure the distribution features of a spectrum *X(i), i = 1, 2, . . . ,N* of a signal *x(i), i = 1, 2, . . . ,N.* The spectrum is firstly be normalized

$$p_i = \frac{X(i)}{\sum_{j=1}^{N} X(j)} \quad (2) \qquad (2)$$

Where, $\sum_{i=1}^{N} p_i = 1$; *N* is the length of spectrum sequence, In this way, the spectrum is just like a probability distribution and its entropy, denoted by *SE* can be obtained by[11,12].

$$SE = -\sum_{i=1}^{N} p_i \log_2(p_i) \qquad (3)$$

SE may change with the length of spectrum sequence. For comparison it is usually normalized by the length to obtain a normalized spectral entropy *SEn*

$$SEn = -\frac{\sum_{i=1}^{N} p_i \log_2(p_i)}{\log_{2(N)}} \qquad (4)$$

Thus, the value range of spectral entropy defined in equation (4) is between 0 and 1. *SEn* produces larger value when amplitude distribution is flat, especially when amplitudes of each frequency component are equal, where it yields the largest value 1. Or it yields smaller value if amplitudes concentrate in few frequency components, especially when only one frequency component has non-zero amplitude, where it yields the smallest value 0. Because of this numerical property, spectral entropy is capable of evaluating the vibration signal's spectral structure.

Spectral entropy has been used in fault diagnostics to differentiate different faults, such as normal valve, spring failure, and valve plate break [13, 14, 15]. In this study, its capacity of reflecting rolling element bearing performance degradation is studied.

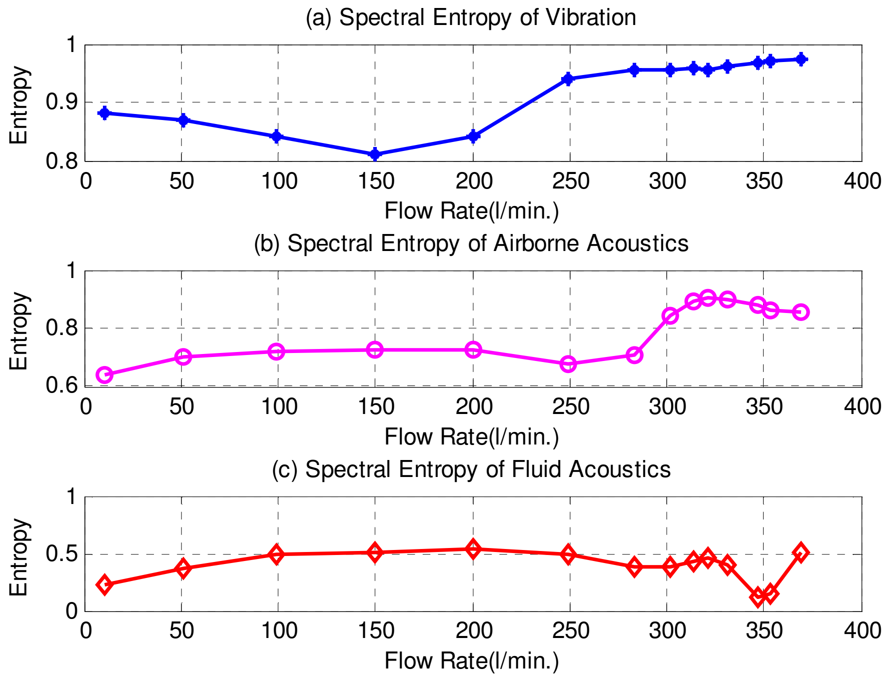

Figure 6 Spectral entropy of vibration, airborne acoustic and fluid acoustic spectra

Figure 6 shows spectral entropy trends obtained from the spectra of vibration, airborne acoustic and fluid acoustic spectra. It can be seen that the vibration and airborne acoustics exhibit a relatively similar and smooth trend. It indicates that spectral entropy gives a better characterisation of the spectra and hence they can be used for cavitation monitoring. However, the acoustics from fluid oscillates too much. So it is difficult to use it for monitoring.

Interestingly, comparing the trends between vibration and airborne acoustics and with the results in Figure 2, it is found that the acoustic trend is more consistent with that of cavitation progression. As it exhibits a nearly steady increase trend with flow rate, it allows the progression phase of cavitation between 250l/min and 330l/min to be defined more accurately. In contrast the vibration trend may indicate a too early start of the cavitation. Nevertheless, the vibration trend has the minimal entropy value at 150ml/min, which may mislead overall diagnosis results.

6. Conclusions

In exploring the characteristics of vibro-acoustic signals from a centrifugal pump with different dimensionless parameters, this study shows that the spectral entropy of the surface vibration and airborne acoustic signals can be used for monitoring cavitation in a centrifugal pump. In addition, it was found that airborne acoustics may give a better monitoring result because it has a more steady increase trend with flow rate and this may be due to that acoustic measurement can capture more global dynamics of the pump than surface vibration obtained in a local position.

Although the peak factor and kurtosis from spectral sequences also can produce similar monitoring results. It may not be so reliable because the corresponding airborne acoustics show a very different trend, which is not consistent with that the spectra of both vibration and acoustics have similar features.

The fluid acoustics in any means cannot be developed a consistent parameter for the purpose of cavitation monitoring. It may be due to effects of wave reflection and resonances in the fluid lines, which will be studied in the future.

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