ME2018_1147
A Study of Diagnosing Reciprocating Compressor Faults using EMD-entropy of the Airborne Acoustic Signals

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Abstract For more effectively and less-costly monitoring of reciprocating compressors, this paper focuses on the developing of a new monitoring method based on airborne sounds which can be easily obtained in and remote way and contains richer information of an entire machine. Possible sound sources of the compressor have been examined according to the working process of mechanical motions and fluid dynamics in order to understand the sound characteristics under different operating conditions. Especially, the sound signal from the compressor is highly non-stationary due to the periodic excitation resulted by the combined effect of gas dynamics and the mechanical forces acting on the compressor associated with the random disturbances of valve motions and flow turbulence and the variations of discharge pressures. In addition, the acoustic signals are also veritably influenced by background noises which often are of unsteady. To characterise such signals for fault detection and diagnosis, Empirical Mode Decomposition (EMD), an effective tool for non-stationary signal analysis, is used to find and enhance the inherent information that correlates more to the various acoustic events involved in compressor operations. Experimental studies, carried out based on a two-stage reciprocating compressor, have shown that Intrinsic Mode Functions (IFM) from EMD can depict more of the signals to indicate the conditions of the machine. In particular, using EMD- entropy as a diagnostic parameter allows common faults such as inter-cooler leakage (ICL) and discharge valve leakage (DVL) to be discriminated and separated from the baseline operation over a wider range of discharge pressures, demonstrating that the proposed EMD acoustic signatures can be an effective approach for monitoring reciprocating machines.

Key words Condition monitoring, Reciprocating compressor, Airborne acoustic, EMD-entropy.

1.0 Introduction

Condition monitoring of a compressor based on airborne sound analysis is difficult to perform due to its high level of noise associated with it. There are many traditional approaches including cylinder pressure monitoring, valve temperature monitoring, and vibration measuring for crosshead cylinders to diagnose any abnormal behaviour of the compressor. But all these techniques are depended on installing the respective sensors to the compressor body. Acoustic measurement can be useful in this case as it can remotely record airborne sound from the compressor and cost effective as well due to its use of minimum numbers of transducer microphones compared to the other sensors. There are many approaches to look into the airborne signal for its characteristic behaviour like time domain analysis, statistical parameter estimation, frequency domain analysis, time frequency domain analysis and advanced signal processing technique.

Elhaj *et al.* [1] used instantaneous angular speed (IAS signal) in order to find out the faults from a two-stage reciprocating compressor and validated their model with the test results obtained from the analysis. Toprak and Iftar [2] placed several microphones at different places of a virtual hemi-sphere and used sound power level as a raw data input for their analysis. Analysis shows that the obtained results from the artificial neural network and multilayer perception model is capable of finding faults from the compressor. Elhaj *et al.* [3] proposed a new method for diagnosing of valve faults in a reciprocating compressor by using time domain, frequency domain and continuous wavelet transform (CWT) of the acoustic sound signals. Lowson [4] presented a theoretical analysis of the noise from axial-flow fans and compressor. Ball *et al.* [5] analysed the acoustic signals from a diesel engine using the smoothed pseudo-Wigner-Ville distribution (SPWVD) and continuous wavelet transform (CWT) and successfully detected the faults from their analysis.

Huang *et al.* [6] introduced a new signal processing technique called Empirical Mode Decomposition (EMD), a noise assisted data analysis method, completely dependent on the local characteristics of a signal. This made the analysis of the non-stationary and no-linear signals very easy and can be successfully diagnose the faults in rotating machineries [7-9]

Information entropy can be used as a successful tool in the signal processing point of view. Huang *et al.* [10] developed an intelligent fault diagnosis method of high voltage circuit breaker using improved EMD energy entropy and multiclass support vector machine. Cui et al. [11] used information entropy and support vector machine to diagnose the valve faults from a reciprocating compressor.

From the literature review it is found that the EMD method is very effective for the analysis of non-stationary signal and entropy can be a good choice as an effective feature tool in the diagnosing of the faults in a machine. Thus, combining these two methods might be effective for diagnosing the compressor faults by analysing the airborne sound signal recorded from the compressor.

Initially this paper focuses on the finding the noise sources generated from the compressor with clear understanding of the fluid dynamic interaction with the mechanical components and keeping in mind the non-stationarity behaviour of the compressor noise signal, the developing of a robust fault feature extraction method based on EMD entropy to diagnose the different health conditions of the double stage single acting reciprocating compressor.

2.0 Acoustic sources associated with the compressor

To utilise acoustic signals for monitoring and diagnosis, it is necessary to understand the possible sound sources of a reciprocating compressor and their primary characteristics associating with common faults.

2.1 Basics of Sources

According to the working process of a reciprocating compressor, the acoustics can be broadly caused by aerodynamic and mechanical forces. The piston moving back and forth in a cylinder forms zones of compression and rarefaction, which formalises acoustic periodic pulsation waves that propagate through the system at the speed of sound of the gas [12]. In addition, the mass flow through the discharge or suction is further modulated to cause the pulsations and produces turbulent noise. These acoustic waves propagate through the system and are reflected due to the impedance inequality. This incident and reflected waves correspond to not only one cylinder but the other cylinder as well, which often cause acoustic resonances. The pulsation is a function of speed of sound of the gas and the frequency of the pulsation [13].

$$c = f \times \lambda \tag{1}$$

where c= speed of the sound; f = frequency and λ is the wavelength.

From more study, it can be found that there is a linear relationship between the velocity of the face of the piston and the resultant pressure. As the motion of the piston is nearly sinusoidal, the pressure wave also follows the same trend. In general, the pulsating flow responsible for the results in the increment of sound level of the compressor.

The inertia forces produced by the motion of the pistons and related part give rise to the level of vibration and noise in the RC. Imbalance of con-rod and crank mechanism is also another cause to produce vibration in compressor. Static and dynamic unbalances produced by the unbalance masses accumulated in the rotating parts of the machine is also considered as a major exciting factor that causes the vibration hence noises from the compressor body. Another major source of noise from the cylinder is the knocking sound produced by the pistons and crank con-rod system during crossover. The gap between the piston and cylinder liner permits the piston to move with a certain velocity in the transverse direction, impacting against the wall of the cylinder. These contribute to an intense vibration of the cylinder walls at their resonant frequency.

One other important acoustic source in a reciprocating compressor is its suction and discharge valves. The valve noise has oscillatory energy transmitted to the

outside through pipes and the housing shell. The compressor valve produces flow oscillations, even if it does not flutter. The suction and discharge valve system of the air compressor consist of a stainless-steel ring plate of high tensile strength riveted to a valve assembly block. During the compression and expansion cycle air is forced through a set of ports located in the valve assembly block and directed against the valve element. The valves open, dump mass into the discharge system or take it from the suction system, close, and after one cycle, open again. As a result a valve knocking sound is produced.
Although these mechanical and aerodynamic forces are dominant, the driving motors and transmission system also contribute to the overall acoustics of a compressor.

2.2 The effect of faults on acoustics

One of common faults in compressors is the air leakage, which usually happen on various valves and pipeline joints. In this study, intercooler leakage (ICL) and discharge valve leakage (DVL), which are more typical due to the high vibration and thermal shocks, are examined for the evaluation of acoustic based condition monitoring.

Influence of ICL:
If there is a leakage in intercooler, the pressure drops in the system. The intercooler leakage causes a significant pressure drop in the first stage cylinder and reduces the force to keep the discharge valve closed. As a result, the discharge valve opens earlier for the first stage and causes delay in the opening of discharge and suction valves for the second stage. Insufficient pressure causes the second stage suction valve improper lifting and results in valve fluttering. The insufficient gas flow pressure causes the valve descending back to its closed position very late compared to its rise time and Flutter occurs. This phenomenon creates an audible noise which can be heard as a clattering sound generated from the compressor. Moreover, the leakage flow streams also create broad band sound due to turbulences.

Influence of DVL:
In case of discharge valve leakage, the mass flow takes place across the cylinder through the leakage because of the constant area of opening in the form of leaking area. Hence this leaking mass flow rate depends on the pressure difference across the valve. Until the valve is closed this area remains constant. During this time the pulsating pressure increases as the compressed pressure passes through the discharge valve and increases the air pressure in the pipe resulting an increasing pressure difference between reservoir and pipe. The fluid flows in the opposite direction to its regular state. The back flow in this case can contribute to the turbulence of the gas flow resulting in pulsation. Valve also generates pressure pulsation because of its vibration and the resulting pressure will mainly be in the natural frequency range of the valve. As a result, resonance of the valve occurs. Due to the pulsation the impact velocity of the valve plate against the seat increases. Short

pulsation can cause high valve impact. These resonating sound and valve impacts have a rich influence for the radiation of noise through the compressor shell.
The acoustic signal from the compressor is highly non-stationary due to the periodic excitation resulted by the combined effect of the gas dynamics and the mechanical forces acting on the compressor associated with the random disturbances of valve motions, flow turbulences and the variations of discharge pressures. This phenomenon gives rise in the randomness or complexity in the airborne signal recorded from the compressor.

3.0 Experimental set up

The experiment is carried out on a two-stage single acting reciprocating compressor identified as the Broom Wade TS9 and is represented in Figure 1 below. This machine has proven suitable for condition monitoring purposes over the years. The V-shaped reciprocating compressor chamber is made up of two cylinders positioned at an angle of 90° to each other giving it the V-shape. These cylinders are tailored to deliver compressed air between 5.5 bar (0.55Mpa) to 8.3 bar (0.8Mpa) to a horizontally positioned receiver tank with a maximum capacity of 13.8 bar (1.38Mpa). Also included in the compressor chamber is an intercooler coil connected from the first stage cylinder (after discharge) to the second stage cylinder (before suction). The compressor chamber is powered by a 2.5KW three-phase induction motor, which transfers electrical current to the compressor pulley to mechanical move the crankshaft.
Two different faults have been simulated: inter-cooler leakage fault and discharge valve leakage fault. One microphone sensor has been used to record the airborne signals placed at 60 cm distant from the main compressor body.

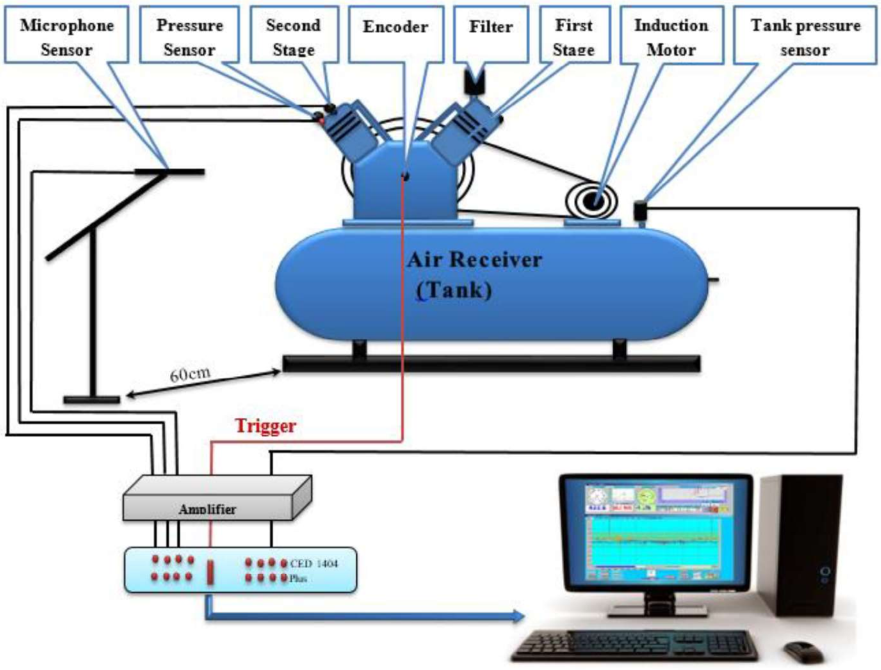

Figure 12 Schematic diagram of the compressor test rig.

The noise from the compressor propagates as longitudinal waves through the elastic air medium. The sound waves cause changes in the air pressure and can be recorded with pressure sensitive instrument like microphone. The sound pressure that we heard is dependent on the distance from the source and the acoustic environment in which sound propagates [12]. If the sound comes contact with a surface some portion of the sound is reflected and some is absorbed. Hence the measured sound pressure is partly consisting of sound that generates from source and partly of the sound that is reflected by surrounding surfaces. As sound propagates outwards from the source its' energy get dissipated and becomes weaker the further it travels from the source. The shape of the sound wave with no obstacles in its way is approximately spherical. This condition is often termed as 'free field' condition where the energy emitted at a given time will diffuse in all directions and, one second later, will be distributed over the surface of a sphere of 340 m radius [13].

A simple point source radiates uniformly in all directions. In general, however, the radiation of sound from a typical source is directional, being greater in some directions than in others. The directional properties of a sound source may be quantified by the introduction of a directivity factor describing the angular dependence of the sound intensity. To avoid the near field effect the sound measurements are made at a sufficient distance from the source which is usually greater than 1m [13].

Whenever sound waves encounter an obstacle, such as when a noise source is placed within boundaries, part of the acoustic energy is reflected, part is absorbed and part is transmitted. The relative amounts of acoustic energy reflected, absorbed and transmitted greatly depend on the nature of the obstacle. Different surfaces have different ways of reflecting, absorbing and transmitting an incident sound wave. The higher the proportion of the incident sound reflected, the higher the contribution of the reflected sound to the total sound in the closed space. As the surfaces become less reflective, and more absorbing of noise, the reflected noise becomes less and the situation tends to a "free field" condition where the only significant sound is the direct sound. In practice, there is always some absorption at each reflection and therefore most work spaces may be considered as semi-reverberant. The phenomenon of reverberation has little effect in the area very close to the source, where the direct sound dominates [13].

Due to the physics behind the propagation of sound it is very important to place the microphone sensor in a proper position. Keeping in the mind, the reverberation, directional propagation property of the sound, the microphone sensor has to be placed at some position where the direct sound dominates and nearly free field property exists. As well as from a study of the proper placement of microphone sensors for identification of RC noise it has been found, that there are three locations where noise response found to be good: first is over the cylinder head, second is at NRV (Non-Return Valve) side, third one is at opposite of NRV side and fourth one is at the opposite side of the flywheel [14].

In this present investigation one microphone sensor was placed at a distance of 60cm from the compressor body at the side opposite to the flywheel minimising the effect of directional propagation and reverberation property of the sound.

4.0 Methodology

In this study, EMD based entropy method has been used to find out the difference between the health conditions of the compressor under a range of operating discharge pressures like 60 psi, 70 psi, 80 psi, 90 psi, 100 psi, 110 psi and 120 psi.
An adaptive signal processing method Empirical Mode Decomposition has been used in this investigation. EMD decomposes a time domain signal into several intrinsic mode functions preserving the frequency components from the original signal in each mode of oscillatory function. As the name suggests the process is empirical which implies that the decomposition is based on the local characteristics of the signal. Thus, it can be effective for non-stationary signals like the airborne signal from the compressor body. The method is found to be superior and different from the traditional techniques like Short Time Fourier Transform (STFT) and Wavelet transform (WT) used for non-stationary signal analysis. The STFT process assumes some portion of the signal as stationary and performs FFT for the windowed segments of the signal and in Wavelet transform there are different types of wavelets available that are used to match the signal behaviour. The WT method is based on the basis functions. Therefore, prior knowledge about the acquired signal is necessary to apply this method, whereas EMD does not require any basis function and is purely adaptive in nature. This kind of data driven tech-

nique makes it easier to analyse any kind of signal whether it is non-stationary or non-linear.

EMD splits a raw signal into several subsets of IFMs. More details can be investigated with these IFMs, which allows key features to be extracted for discriminating different scenarios for fault diagnosis.

Step 1: If C_1, C_2, ..,C_i, .., C_N be the IMF components obtained by the EMD method for different datasets, then the energy and energy ratio of these IMF components are calculated as follows:

$$E_i = \sum_{n=1}^{N_1} C_i(n)^2 \tag{2}$$

$$P_{Ei} = \frac{E_i}{\sum_{i=1}^{N} E_i} \tag{3}$$

where E_i is the energy of the i^{th} IMF component; N_1 is the number of sample points present in the IMF; P_{Ei} is the energy ratio for i^{th} IMF component; N is the total number of IMFs generated by the EMD process.

Step 2: Form the energy ratio vector P_E by putting all the energy ratios for individual IMFs.

$$P_E = [P_{E1}, P_{E2}, P_{E3}, ..., P_{EN}] \tag{4}$$

Step 3: Find the entropy of the energy ratio vector $E_{entropy}$.

$$E_{entropy} = -\sum_{i=1}^{N} P_{Ei} \log_2 P_{Ei} \tag{5}$$

Step 4: Consider four different datasets for each machine condition and determine the average entropy value for respective tank pressure.

5.0 Results and Discussion

5.1 Change of cylinder pressure

The effect of the simulated faults on in-cylinder pressure at 120 psi is shown in Figure 2. The intercooler and discharge valve leakage faults cause a change in the pressure waveform in both stages.

When the intercooler leakage occurs the pressure inside the 1st stage cylinder is lower than the baseline one. This causes a reduction of force to keep the 1st stage discharge valve closed. Thus, the discharge valve opens earlier in 1st stage cylinder with intercooler leakage fault.

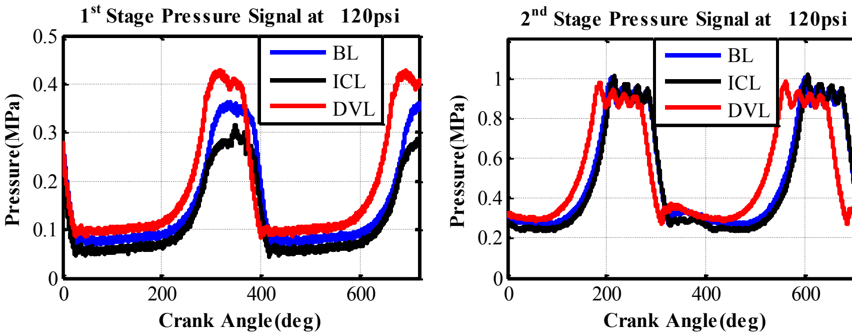

Figure 13 1st stage and 2nd stage pressure comparison for different compressor health conditions at 120 psi.

As for the 2nd stage, the intercooler leakage causes a small change in the cylinder pressure; the pressure inside the cylinder is lower than in healthy operation. This will result in reduction in discharge efficiency. The drop in the cylinder pressure causes a delay in the opening of both suction and discharge valves.

On the other hand, discharge valve leakage leads to a slight but significant increase in the pressure of the first stage cylinder compared to the leaky valve (intercooler leakage) and healthy operations. This increase in the cylinder pressure causes the first stage suction and discharge valves to open early due to the higher pressure in the cylinder. The second stage discharge valve leakage causes the both valves open and close earlier than for the healthy operation. The discharge valve leakage fault causes the second stage cylinder pressure to build-up earlier than the healthy operation; thus, the process of discharge occurs earlier.

5.2 Acoustic waveform analysis

The airborne sound signal from the compressor with different health conditions at different pressures has been acquired by the microphone installed at 60 cm distant from the compressor body. The comparison of the time domain acoustical waveform for different health conditions at the pressure of 120 psi has been shown in Figure 3.

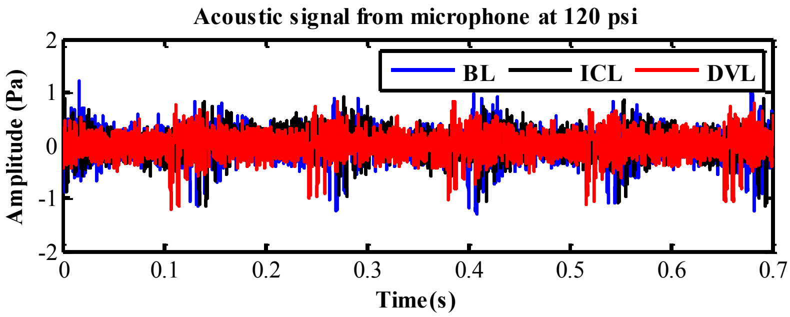

Figure 14 Time domain acoustic waveform comparison at 120 psi.

From the analysis of the above figure it can be shown that the waveforms of the baseline (BL), intercooler leakage (ICL) and discharge valve leakage (DVL) has some differences with each other. The time differences between the excitation of the signals shows a clear difference between the machine conditions. These excitations are caused by the opening and closing of the valves. From the time domain waveform representation, it can be seen that opening of the valves with different machine conditions follows the same trend obtained from the pressure analysis in the previous subsection. Result shows that the opening of the valve in case of discharge valve leakage is the earliest whereas, for intercooler leakage the delay time for opening the valve is maximum. Though it is hard to reach a conclusion about the machine conditions because the opening and closing of the valves also depend on other factors, the time domain waveform analysis is not sufficient to differentiate between the compressor conditions. Further advanced signal processing method is required to diagnose the faults in a compressor.

5.3 EMD of the acoustic signal

As the compressor sound signal is highly non-stationary due to the periodic excitation resulted by the combined effect of gas dynamics and the mechanical forces acting on the compressor associated with the random disturbances of valve motions and flow turbulences and the variations of discharge pressures, from the literature review it has been found that the data driven technique EMD is very effective for non-stationary and non-linear application hence the proposed method based on EMD has been applied to find out the faults from the compressor. Figure 4 shows the first five intrinsic mode functions (IMFs) obtained from the EMD process for different machine conditions at the pressure of 120 psi.

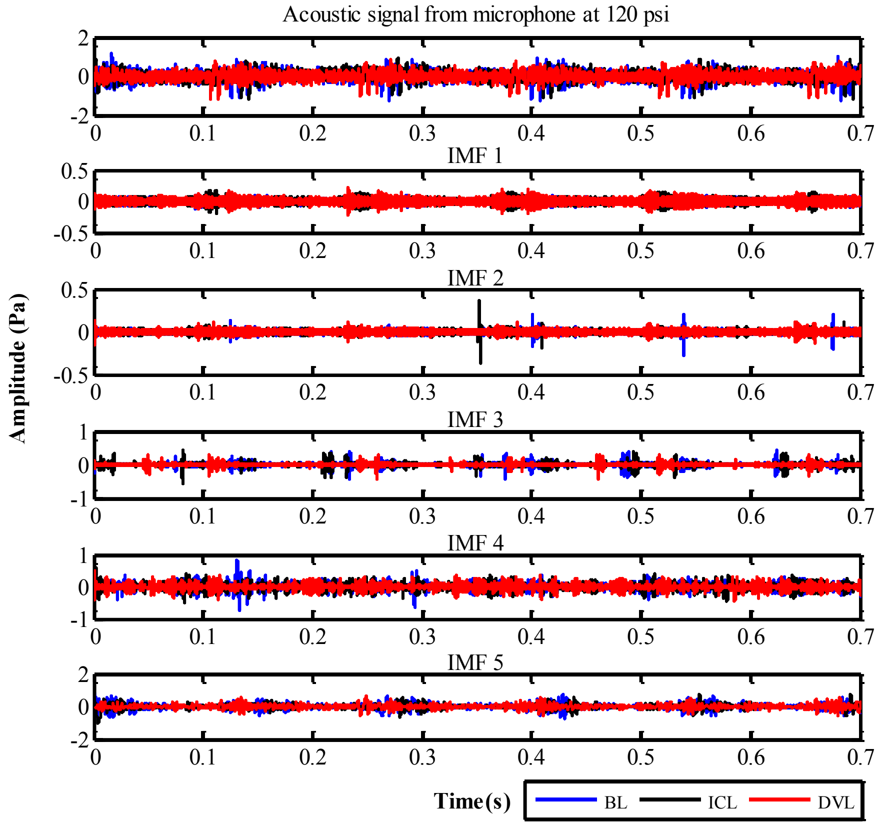

Figure 15 Decomposed IMF signals for three different compressor conditions at the pressure of 120 psi

From the Figure 4 it has been seen that the decomposed IMFs from EMD show the randomness of the signals whereas, normal time domain signals fail to show the property. Randomness is a phenomenon which gives an indication of how randomly a signal changes over the time. Generally, the sound signal from the compressor has a periodic behaviour due to the steady rotational motion of the mechanical systems. This periodic phenomenon can be seen in Figure 3. Due to the instable and turbulent fluid force the sound signals contain some sort of randomness which cannot be identified from the time domain analysis, whereas, EMD analysis makes it possible to identify that property in the IMFs generated by the process (Figure 4). In both occasion, the baseline and the faulty conditions with ICL and DVL show some sort of randomness in signal information obtained from the decomposed IMFs by EMD process. Especially if the 5th IMF for different machine conditions are plotted separately with less time duration the randomness of the signals are clearly visible in Figure 5.

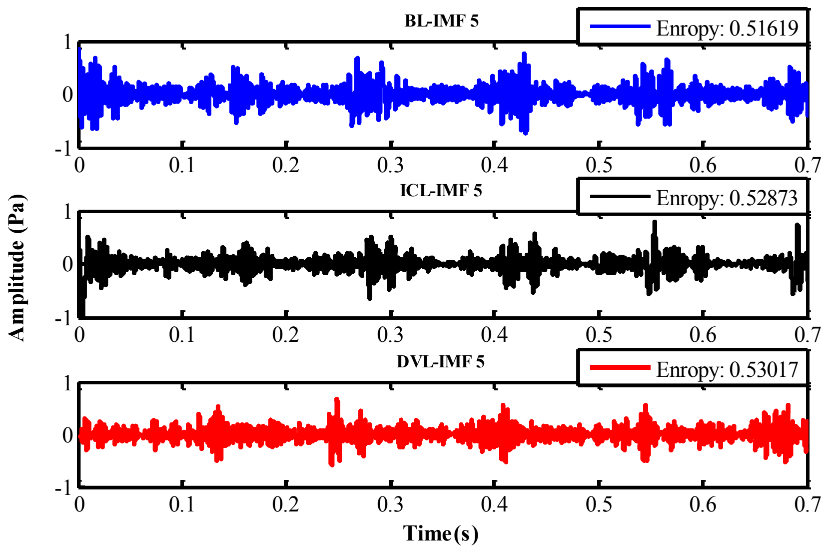

Figure 16 Decomposed third IMF at 120 psi show its randomness property for different compressor conditions.

5.4 EMD-Entropy calculation

Entropy is a parameter which measures the randomness of the signal. More randomness means less information, less organized which results in an increase of entropy values in this case. Provided that a faulty compressor can cause more randomness in acoustic signals due to turbulence flows, the entropy of the decomposed IMFs has been chosen as a feature parameter in this analysis. Though the concept of entropy and RMS looks like same, but there is an obvious difference between the two. The statistical parameter RMS of an acoustic signal provides an average loudness. This describes the strength of a signal. The RMS value is only related to the signal amplitude. The root-mean-square pressure is most often used to characterize a sound wave because it is directly related to the energy carried by the sound wave, which is called the intensity. The intensity of a sound wave is the average amount of energy transmitted per unit time through a unit area in a specified direction. The RMS pressure is most often used to characterize sound waves that have the simple shape. Not all sound waves have such simple shapes, which contain only one frequency. Especially in compressor the non-stationary behaviour of the sound signal and its change in amplitudes over time gives an overall energy content of the signals while measure the RMS value. The results obtained from the RMS and entropy values are compared and shown in Figure 6.

Four sets of data for each condition have been considered in the analysis. Finally, the average RMS and entropy values from the four datasets for baseline, intercooler leakage and discharge valve leakage faults have been calculated for a broad range of pressures from 60 psi to 120 psi and are shown in Figure 6. From the results it has been shown that there is a clear separation of the compressor machine conditions while analysing entropy values whereas, RMS value does not give the clear indication of the machine conditions as it's not reliable.

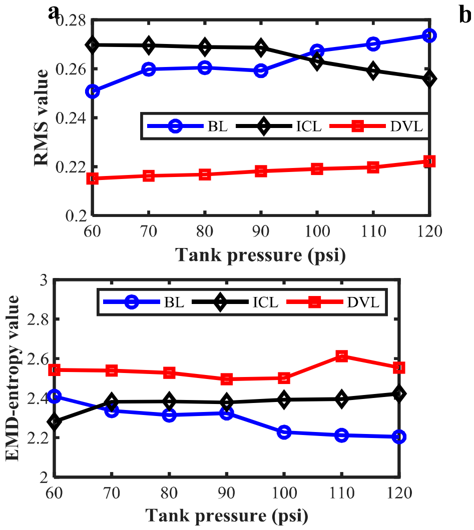

Figure 17 (a) RMS values with changing tank pressure, (b) Entropy values correspond to the tank pressure for different conditions.

From the Figure 6(b), the physics behind the noise generation for different machine conditions can be explained by the change of energy entropy with the increasing tank pressure.

At low pressure, the performance of the compressor is not stable and this is reflected in the pressure range of 60-70 psi. The energy entropy in the particular low pressure zone for different machine conditions is inconsistent. At the intermediate pressure zone (70-90 psi), the working of the compressor system is normal and the energy entropies for the respective machine conditions almost follow a straight line path. There are no abrupt changes of the entropy values in this pressure region. At high pressure (>90 psi), the severity increases and it is noticeable that the entropy values for ICL and DVL are much higher compared to the baseline and an increasing trend in entropies for both the fault conditions is recorded with the corresponding pressure whereas, when the compressor works in a normal condition the sound generated from it is more organized and as the pressure increases it works more smoothly. That indicates the decreasing of entropy values with the pressure in case of baseline condition.

The sound mechanism in the presence of ICL and DVL can also be explained from the results obtained from the analysis.

Presence of ICL fault: The presence of this kind of fault causes a significant pressure drop in the first stage cylinder and can be shown in Figure 2. As a result, a delay is caused in the opening of the suction valve and can be shown in the time domain waveform of the acoustic signal (Figure 3). Any presence of fault increases the randomness of the signal and the entropy values go up. The same thing is noticeable in Figure 6(b), where the ICL fault has the higher entropy values than the normal baseline condition.

Presence of DVL fault: The presence of this kind of fault increases the pulsating pressure as the compressed pressure passes through the discharge valve. This increases the air pressure in the pipe resulting an increasing pressure difference between reservoir and pipe. The back flow in this case can contribute to the turbulence of the gas flow resulting in pulsation and produces noise. This turbulence results in the increase of entropy values for the airborne signal recorded with this seeded fault and from Figure 6(b) it can be found that this kind of fault has the highest entropy values due to the pulsation occurred.
From the analysis, it is confirmed that the entropy values change with the increase of pressure conditions. The increasing value of the entropy with the increase of tank pressure indicates the severity of the simulated faults with the pressure change. The entropy values for three conditions of the compressor show a clear separation among them. From the analysis result, the proposed method based on EMD-entropy is found to be effective for diagnosing faults in the reciprocating compressor.

6.0 Conclusion

This paper focuses on the study of airborne sound signal characteristics for the diagnosis of faults in a reciprocating compressor. The experimental results have shown that the acoustic signal contains rich information reflecting the differences of simulated fault cases but it is highly non-stationary due to a combination of different sources including various noises. An effective signal processing method, EMD, is used to detail the insight of the signal where the randomness is clearly identified, and therefore EMD-entropy is used to quantify the amount of randomness present in the signals for three different cases. The diagnostic results show that the entropy values provide a better separation between the faulty cases, compared with the conventional RMS analysis, demonstrating acoustic based condition monitoring can be more efficient, along with its simplicity of signal acquisition.

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