Centrifugal Pump Impeller Crack Detection Using Vibration Analysis

Waleed Abdulkarem, Rajakannu Amuthakkannan, and Khalid F. Al-Raheem

Abstract**—**The detection of the centrifugal pump impeller blades cracks using vibration analysis technique is investigated using both time and frequency domain methods. In time domain a time index parameter is applied as fault indicator and the power spectrum analysis is used as a frequency domain analysis. Initially, the vibration of centrifugal pump is measured at healthy condition and compared with nine different artificial cracks sizes which are introduced in the impeller blades. The results show the effectiveness of using both time index parameter and frequency spectrum for fault diagnosis. The amplitude of the impeller passing frequency and the vibration time index are increased as the blade crack size is developed, which can be used as indicator for fault severity.

Keywords**—** centrifugal pump, impeller crack, vibration index, power spectrum

I. Introduction

CONDITION monitoring is considered as one of the most powerful type of maintenance. It is continuously monitoring the Machine, the maintenance will be schedule based on results of continuous measurements of the machine condition. Since each mechanical component produce warning of the it's impeding failure, a special electronic transducer used to detect and diagnosis that failure. Actually it is very effective and applicable method and it is not doubt when Raymond, S. B., [9] mentioned "it applies to 80% of maintenance, according to the International Foundation for Research in Maintenance (IFIRM). This technique is applied for the costly, large and significant machinery which is directly affects the production process. Since condition monitoring would be very powerful maintenance technique for the continuously running machines. Therefore it would be very efficient to apply it this project problem in which the centrifugal pump impeller will be maintain continuously.

In fact many of earlier researchers have done many investigations and examinations in this field. Golbabaei, *et al.*, [3] have studied failure detection and optimization of a centrifugal-pump volute casing. They have presented fatigue life expected for both optimized and real failed volute casing. They have seen the theoretical fatigue life expectations are in acceptable arrangement with volute casings practical life under study. Harihara and Parlos [4] investigated the used of sensor-less impeller cracks detection in motor driven centrifugal pumps using current signal. They were able to detect the impeller faults where the vibration level increases with time as the number of the faults increases. University of Strathclyde [11] has designed computer –based system for fault analysis and diagnosis simulation (FADS). They have been noticed high level of vibration at impeller blade passing frequency and its harmonics.

Abdel-Rahman and El-Shaikh [10] have investigated the effects of pump unbalance and misalignment on the overall pump vibration level. Overall vibration levels indicate severity of vibration and compared with ISO 10816-1. Also, the vibration spectra, which indicate the relation of vibration amplitude with frequency, are measured to determine the excitation frequencies and the source of high vibration. The unbalance condition for the motor fan showed a high vibration spectrum in the axial and radial directions of maximum amplitude at the motor speed.

Zouari, *et al.*,[6] described a diagnosis system for centrifugal pump based on the artificial neural network as pattern recognition and fault classifier for different fault conditions (misalignment, cavitation, partial flow, and air injection). The collected vibration data are pre-processed in both time and frequency domains to determine feature vector which is used as input to ANN. The results show a high rate of correct classification (98-100%). Non –dimensional time domain features (i.e kurtosis, Skewness, etc.) have been used as input to ANN for centrifugal pump fault diagnosis (cavitation, impeller damage and unbalance), [5]

Rainer Nordmann and Martin Aenis [7] developed built-in software for identification, rotor crack and wear detection, and diagnosis in a centrifugal pump based on the measurement of the systems outputs, i.e., displacements.

Carsten, et *al* [2] developed an algorithm based approach for fault detection and isolation in a centrifugal pump using of structural analysis. The results show that it is possible to distinguish between the four of the five faults under consideration (clogging inside the pump, increased friction due to either rub impact or bearing faults, increased leakage flow, performance degradation due to cavitation and dry running).

Hernandez-Solis and Carlsson [1] formed a diagnostic approach for centrifugal pump faults (cavitation and impeller degradation) based on motor current and power signatures. The signals were analyzed in the frequency domain by means of their Power Spectral Density (PSD). The results show that cavitation enhance oscillations in the shaft with frequencies

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that are multiples of the rotation speed times the number of blades of oscillations in the shaft. Birajdar *et al*. [8] developed a diagnosis method of centrifugal pumps faults using sound pressure level and vibrations frequency spectrums.

In this paper an investigation has been carried out to correlate the impeller fault condition with both time and frequency domain features of centrifugal pump vibration signal.

II. VIBRATION ANALYSIS

A. Machine Vibration Index (VI)

It is a time domain parameter namely Vibration Index (VI) which correlates the vibration amplitudes (Vib) in three directions, vertical (Y), horizontal (H) and axial (Z), with the present and the severity of the fault and defined by Harihara and Parlos (2008):

$$VI = \frac{1}{3}\sum_{XYZ}\sqrt{\frac{1}{N}\sum_{i=1}^{N} Vib_{X,i}^{2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

2.2 Power Spectrum (PS)

It is the power of the spectrum measured in decibel (dB) and given by:

$$PS(dB) = \frac{1}{2\pi}\int_{-\infty}^{\infty} |F(w)^{2}|\, dw \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

Where, F(w) is the Fourier Transform (FT) of vibration signal.

However it is more accurate and powerful than the spectrum analysis as it reduce the unwanted noise or distortion and focus more on the feature signal.

By using power spectrum analysis the machinery fault could be diagnose even with its severity. The presence and severity of the impeller crack impeller cause a clear increment in the amplitude of the spectrum at the impeller passing. If the crack size increases more, the harmonics of impeller frequency will be introduced. With more crack size a sidebands of the impeller frequency at the impeller rotational speed and its harmonics will appear in the frequency spectrum.

III. EXPERIMENTAL SETUP

The condition monitoring process has been applied on a centrifugal pump with the following specifications:-

- Power: 0.5HP, 0.37 kW
- Rotational Speed: 2900 rpm
- No. of Blades: 36 each side

The pump is fitted to a water flow system. As shown in Fig. 1, once the pump operates, it sucks the water from the tank to the upper plastic tank though piping system. The water will reach to a specific level inside the boxes depending on the open of the water valves, and then discharge again to the tank.

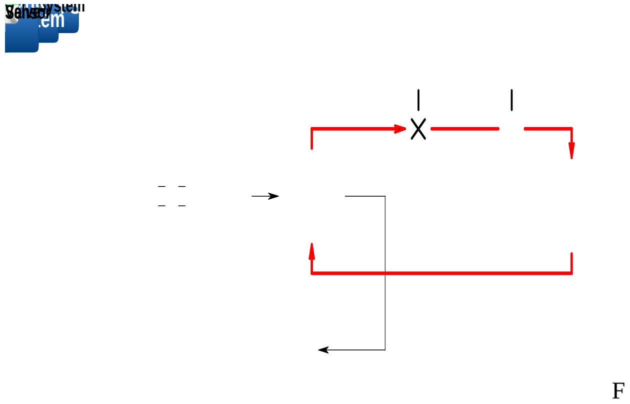


Fig. 1 System working principle

The vibration is measured with an National Instruments N1SCX1 module with DAQ in three directions as shown in Fig. 2

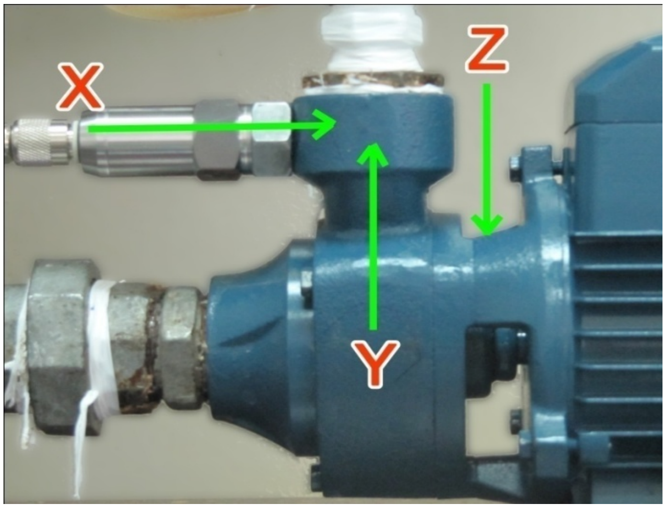
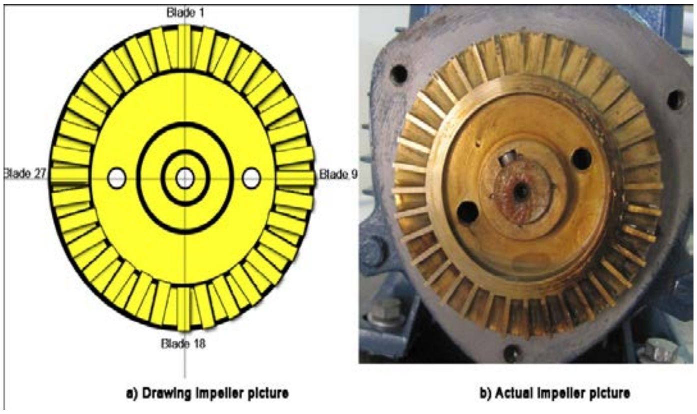


Fig. 2 Accelerometer mounting directions

Before any measurements the main valve of the pump is set as half opened for all condition. Initially the vibration of the centrifugal pump impeller is measured in healthy condition by using mentioned devices. At healthy condition the shape of the impeller will be as shown in Figure 3 where it have 36 blades in each side.



a) Drawing Impeller picture

b) Actual Impeller picture

Fig.3 Impeller at healthy condition

After that a hacksaw is used to introduce the first artificial fault (Crack1) in the impeller blade 1 which has width of 2mm as shown in Fig. 4. Then the second artificial fault in blade 1

(Crack 2) is introduced with same width of the first crack. Finally, the third artificial fault is introduced in which the whole impeller blade is removed. The vibration signals using are recorded for the above fault condition cases.

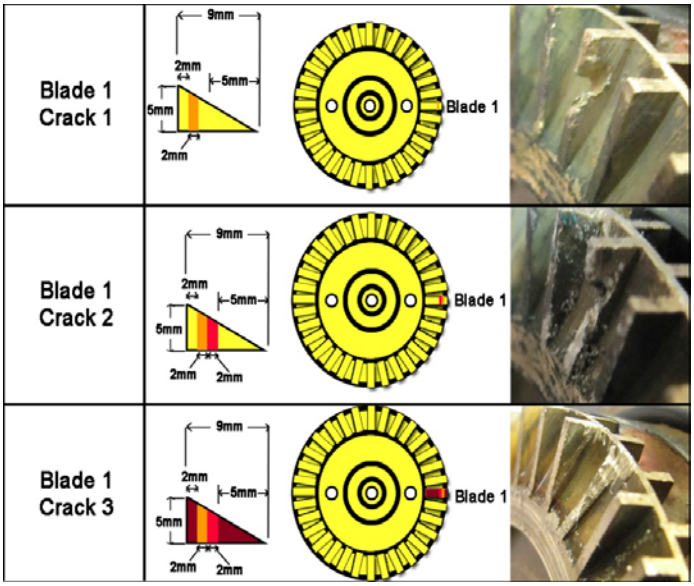


Fig.4 Blade 1 impeller cracks

Moreover, cracks in blade no. 9 which is located at 90 degree of blade 1, and blade 18 which is located at 180 degree of blade 1 are introduced. The different Impeller cracks conditions are shown in Table I.

TABLE I
IMPELLER BLADES CRACKS CONDITION

Impeller Blade	Crack No.	Description
Blade 1	Crack 1	The crack has width of 2mm and height of 4mm
	Crack 2	The crack has width of 2mm and height of 3mm along with crack 1
	Crack 3	The whole blade is removed
Blade 9	Crack 4	The crack has width of 2mm and height of 4mm along with crack 3
	Crack 5	The crack has width of 2mm and height of 3mm along with crack 4
	Crack 6	The whole blade is removed along with crack 3
Blade 18	Crack 7	The crack has width of 2mm and height of 4mm along with crack 6
	Crack 8	The crack has width of 2mm and height of 3mm along with crack 7
	Crack 9	The whole blade is removed along with crack 6

The final shape of the impeller blades with the nine artificial cracks will be as shown in Fig. 5

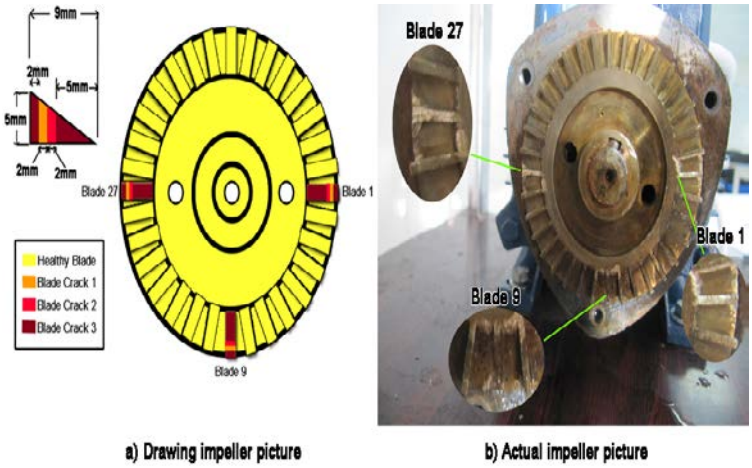


Fig.5 Impeller blades with 9 cracks

IV. RESULT AND DISCUSSION

A. Machine Vibration Index

As shown in Fig. 6, the vibration index increases as the impeller blades cracks increases. At healthy condition, the index vibration is 0.793 while with crack 9 it reaches to 1.341.

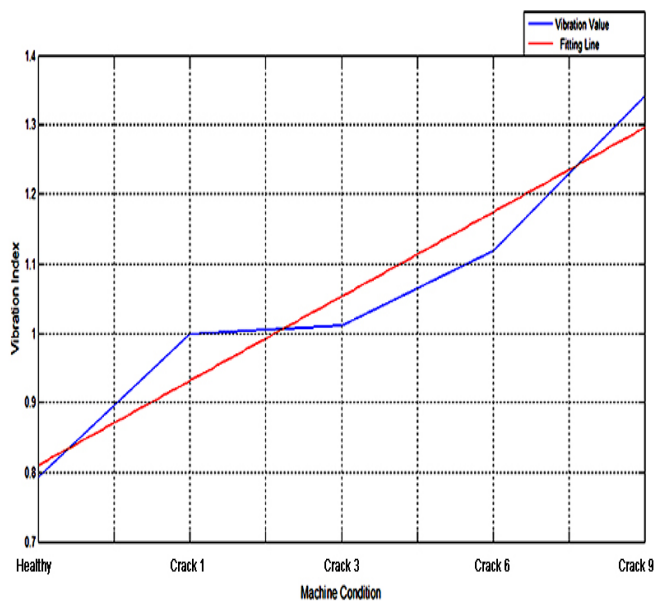


Fig.6 Machine vibration index response

B. Power spectrum

As shown Figure 7 (Crack 7), there are three peaks at frequencies of 1655Hz, 3310Hz and 4965 Hz, with different amplitudes. The 2nd and 3rd frequency peaks are equals to the multiples of the first peak (harmonics). Table II shows the peaks with their corresponding frequencies.

TABLE II
POWER SPECTRUM PEAKS FREQUENCIES WITH THEIR RELATIVE AMPLITUDES

Machine Condition	Amplitude/Peaks Frequencies		
	1655 Hz	3310 Hz	4965 Hz
Healthy	1.352 dB	0.548 dB	6.31 dB
Crack 1	4.326 dB	8.895 dB	11.25 dB
Crack 2	5.364 dB	1.904 dB	8.425 dB
Crack 7	7.327 dB	5.501 dB	2.871 dB
Crack 9	8.62 dB	10.49 dB	0.202 dB

This variation in peaks frequencies could be clearer when it is drawn in logarithmic tending scale as shown in Fig. 8.

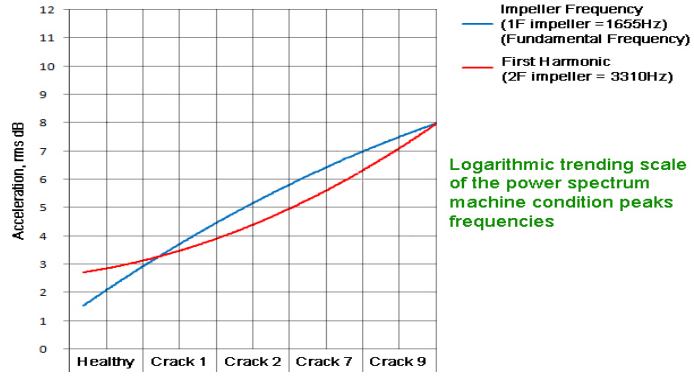


Fig.8 Logarithmic trending scale of the crack condition peaks frequencies.

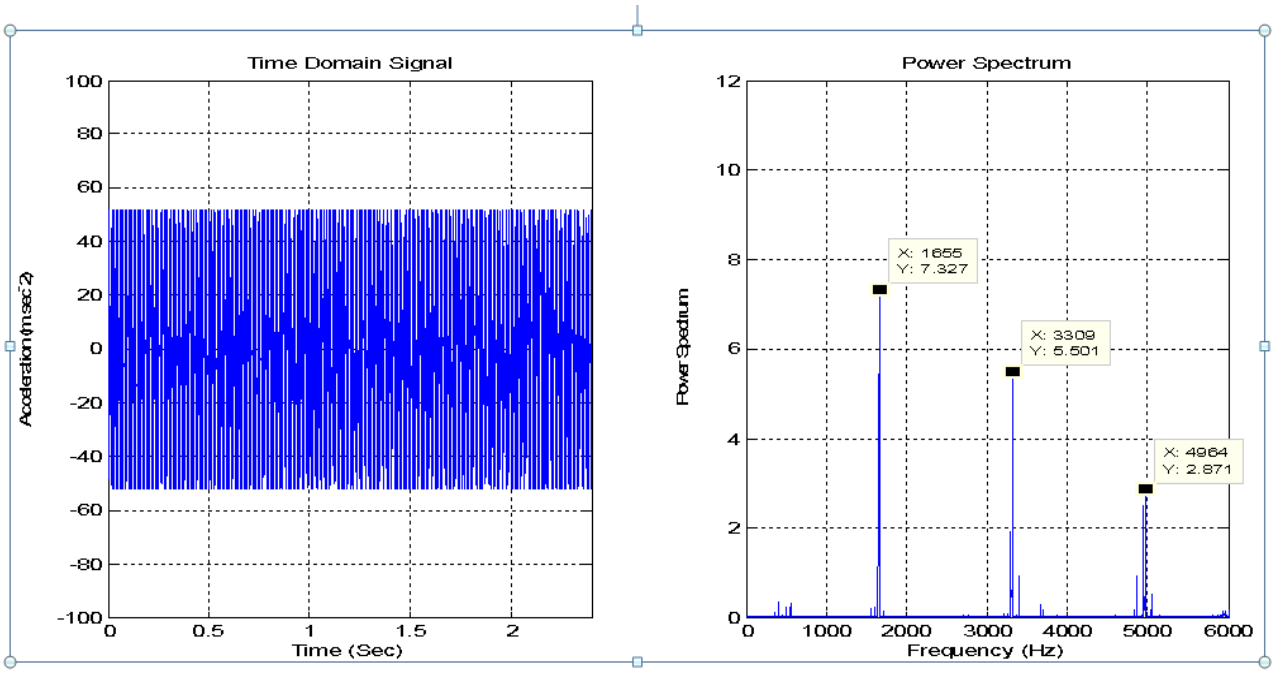


Fig.7 The time domain signal and the corresponding power spectrum for cracked impeller.

By neglecting the measuring errors and machine speed fluctuating, the peaks in all of the crack conditions are appeared at the same frequencies. It obviously, the first peak amplitude is rises as the blade crack size increases. The vibration amplitude at healthy condition is 1.352 dB and it is proportionally increases with crack size until it reaches to maximum value of 8.626 dB at crack 9.

It is clear that the peak frequency is related to the impeller frequency ($F_{impeller}$) which equal:-

$$F_{impeller} = \frac{No.of\ impeller\ blades \times RPM}{60} \qquad \ldots\ldots\ldots\ldots\ldots\ldots(3)$$

$$F_{impeller} = \frac{36 \times 2900}{60} = 1740\ Hz$$

By ignoring machine speed variations and measuring errors:

$$F_{impeller} = 1st\ F_{peak}$$
$$\therefore F_{impeller} = 1655Hz$$

It is possible to diagnose the cause of the second peak and third peak by the following:

$$1F_{impeller} = F_{impeller} = 1655\ Hz$$
$$2F_{impeller} = 1655 \times 2 = 3310\ Hz$$
$$3F_{impeller} = 1655 \times 3 = 4965\ Hz$$

Sothat, $2nd\ Peak = 2F_{impeller}$ and $3rd\ Peak = 3F_{impeller}$, that means the 2nd and 3rd peak are the harmonics of the $1F_{impeller}$ as shown in Figure 9.

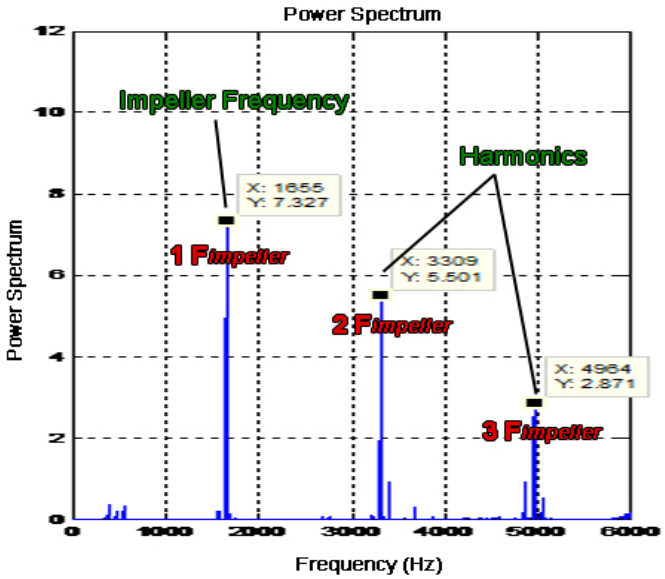


Fig.9 Power spectrum of impeller frequency with its harmonics at crack 7

Moreover, Fig. 9 shows sidebands at the first peak (1F impeller) at frequency ± 45Hz which is approximately equal the machine running frequency (48.3Hz) as shown in Figure 10.

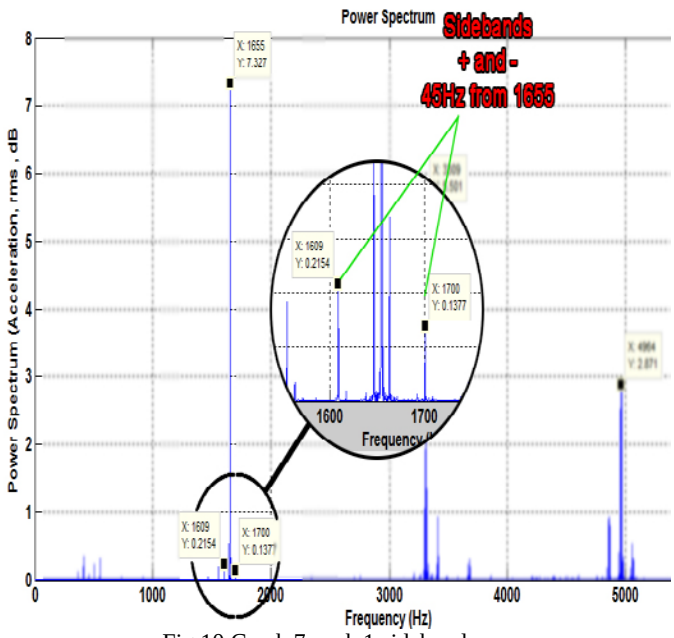


Fig.10 Crack 7 peak 1 sidebands

The calculation to find the location of the sidebands theoretically is given by:

$$Sideband\ 1 = 1655 + 45 = 1700Hz$$

$$Sideband\ 2 = 1655 - 45 = 1610Hz$$

However, it has seen that the harmonics (2nd and 3rd harmonics) of the impeller frequency include very small sidebands amplitude which could be eliminated.

V. Conclusion

This paper investigated the centrifugal pump impeller cracks diagnosis using vibration analysis in both time and frequency domains. The vibration is measured at healthy condition and faulty conditions. The vibration index is applied as a time domain fault indicator and power spectrum at specified frequencies as a frequency fault indicator. It is concluded that the vibration index increases as impeller crack size increases. By using power spectrum analysis, as the impeller crack size increases, the fundamental frequency (impeller frequency) increases. In addition to that, there are two harmonics of impeller frequency and sidebands at impeller rotation frequency i.e. ±45Hz.

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