Using Vibration Monitoring for Rolling Elements Bearing Inspection

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ABSTRACT - In Khartoum Refinery Company (KRC) Pump 1104B suffers of high vibration level. Vibration analysis techniques is used to find out the root causes of the pump vibration. The collected data from the pump indicates high vibration in the pump drive end (DE). After collected data, was uploaded to software in which set up database of measurement points. Spectrum analysis; was used. The analysis result indicates that the angular contact bearing in the pump drive end has been failed. During this work theoretical analysis through mathematical model had been done with the assumption that good bearing has good oil film damping coefficient to restrict the rolling elements movement while the shaft is rotating. The comparison between the mathematical analysis and the measuring data was almost similar. The visual inspection results after bearing box dismantling is angular contact bearing defect that confirmed the theoretical &software analysis of finding vibration source. These results can lead to apply this mathematical analysis as prediction tool to evaluate the bearing condition before total damage also can be applied with the equipment that has no analyzer software because the vibration phenomenon is same.
Keywords*: Bearings, Vibration, rolling elements, damping coefficient.*

المستخلص – في شركة مصفاة الخرطوم (KRC) المضخة 1104 B تعاني من مستوى اهتزازعال. أستخدمت تقنيات تحليل الاهتزازات لمعرفة الأسباب الجذرية لذلك. البيانات التي تم جمعها من المضخة تشير الى اهتزاز عالي في نهاية محرك مضخة (DE). تم تحميل البيانات التي تم جمعها على برنامج أنشئ كقاعدة بيانات لنقاط القياس وحلل الطيف. تشير نتيجة التحليل الى فشل في محمل التلامس الزاوي في نهاية محرك المضخة. من التحليل النظري خلال النموذج الرياضي مع افتراض أن أفضل محمل به قفص تزيين له أفضل معامل لتخميد الإهتزاز بعناصره المتحركة بتقييد حركة العناصر المتحركة في حالة دوران العمود. المقارنة بين التحقيقات النظرية وتحليل الطيف تشير الى تشابه قريب فى نتائج التحليل. أثبت الكشف البصري بعد التفكيك وجود عيب واضح فى محمل الاتصال الزاوي الشيئ الذى أكد نتائج التحليل النظري والبرنامج لإيجاد الأسباب الجذرية للإهتزاز. هذه النتائج تقود الى أنه يمكن تطبيق هذاالتحليل الرياضي كأداة توقع لتقييم حالة المعدات قبل حدوث الفشل كما يمكن تطبيقه لمعرفة السبب الجذري للاهتزاز مع المعدات التي لا تتوفر لها برامج تحليل لأن ظاهرة الاهتزاز هي نفسها.

Introduction:

Vibration is unpredictable phenomenon in early stages which can lead to total damage. All mechanical equipment generates a vibration profile corresponding to their motions which has impacts into their operation condition. The fundamental frequency or one times RPM can be caused by imbalance. When a rotor has one heavy point, the weight of the heavy point, distance from the shaft center line, an angular rate of velocity or speed all combine to create a forcing function or vector at same magnitude and direction. If the rotor has more than one heavy point, the weight will be the vector sum of all the heavy points, an unbalance cause surface force. Amplitude can be maximized by looseness and resonance. It can be minimized by mass, rigidity, and damping. Amplitude can also be maximized or minimized by the most instruments measure amplitude. Harmonic content and signal distortion are other good ways to determine problem severity. All generated frequencies are equal to events, times and speed. If a rotor has imbalance, the heavy spot rotates at the same speed as the rotor [1].

Vibration analysis is the common predictive maintenance technique that used with maintenance management programs. Predictive maintenance and vibration monitoring characteristics of rotating machinery has become synonymous to detect budding problems and to head off catastrophic failure [1].

Objectives

The objectives of this research are listed below:

1. To study and analyzing vibration signals which that can be picked up and use the result to reduce the plant down time that takes place according to unpredictable maintenance .
2. Using the latest technology of vibration measurement to improve the maintenance method of rotating equipment.

Bearing common defects [2]

A group of problems that is common in many bearing includes rolling elements wear, raceway wear, bearings Loose on the Shaft, bearings Loose in the housing and cage loose. All of them have impact in bearing vibration status.

Rolling element bearings defect stages:

Defects can occur in any of the bearing parts and will cause high-frequency vibrations. In fact, the severity of the wear keeps changing the vibration pattern. In most cases, it is possible to identify the component of the bearing that is defective due to the specific vibration frequencies that are excited. Raceways and rolling element defects are easily detected.

Bearing defects generate both the bearing defect frequency and the ringing random vibrations that are the resonant frequencies of the bearing components.

Bearing defect frequencies are not integrally to the harmonic of running speed [3] It is very interesting to note that in fast Fourier transform (FFT) the mathematical calculation that derive a velocity spectrum from a velocity waveform to find both the inner and outer race defect frequencies.

The FFT spectrum for bearing defects can be split into four zones (A, B, C and D), where we will note the changes as bearing wear progresses. These zones are described as [2]:

Zone A: machine rpm and harmonics zone

Zone B: bearing defect frequencies zone (0.8–5) kHz.

Zone C: bearing component natural frequencies zone (5–20 kHz)

Zone D: high-frequency-detection (HFD) zone (beyond 20 kHz) [2].

Stage 1 of bearing defect [2]

The rolling elements or raceways of bearing do not have visible defects during the first stage only appear dull gray and forget the new bearing shine Figure 1 shows that clearly.

The bearing wear first indications will appear in ultrasonic frequency ranges from 20 to 60 kHz. The frequencies could be evaluated through high-frequency detection techniques.

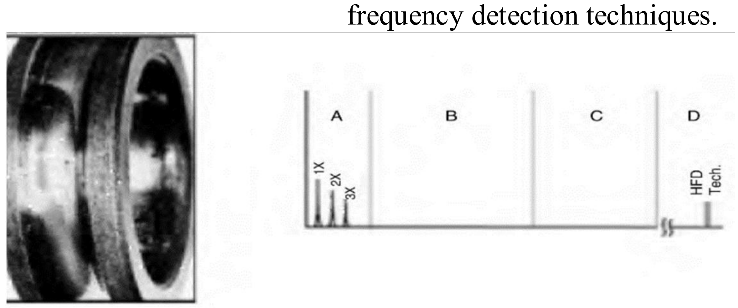

Figure 1: Small defect in the bearings raceways

Stage 2 of bearing defect [2]

In this stage the fatigue begins to develop minute pits in raceways see Figure 2. Rolling elements passing through raceways when touching these pits it starts to generate the bearing component natural frequencies that predominantly occur in the frequency range (0.5 to 2 kHz). At the end of stage two the sideband frequencies (bearing defect frequency ± rpm) appear above and below the natural frequency peak as per the vibration severity. The amplitude in high-frequency detection (HFD) techniques may double compare to the stage one readings.

Stage 3 of bearing defect [2]

The FFT shows visible discrete bearing frequencies and harmonics with a number of sidebands in the third stage (Figure 3). Wear become visible may expand through to the edge of the bearing raceway. The pits of the earlier stage are developing from minute into bigger pits and their numbers also increase when well-formed sidebands accompany any bearing defect.

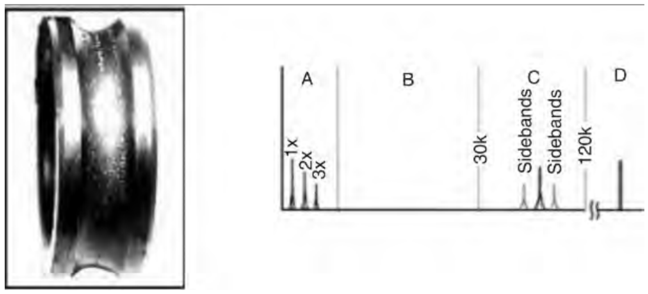

Figure 2: more obvious wear in the form of pits

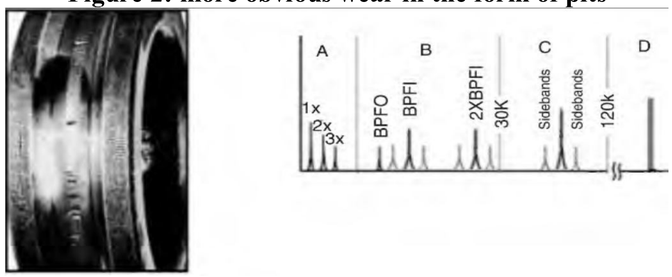

Figure 3: Wear clearly visible over the breadth of the bearing

Stage 4 of bearing defect [2]:

In the last damage stage the pits merge together for creating rough tracks of the bearing raceways or/and rolling elements see Figure 4. The amplitude of the 1 rpm component will raise this cause growth of many running speed harmonics. The visualization is higher bearings clearances allow higher rotor displacement. The bearing defect frequencies that have been separated from the bearing component natural frequencies in this stage begin to merge into a random, broadband high-frequency. The amplitude will drop and the width of the noise will increase. Finally, the amplitude will rise again with increasing of the noise floor span.

Amplitudes of the high-frequency noise floor and some of the HFD may decrease due to pits flattening to spalls; spike energy will usually grow to extreme amplitudes. By this time, the bearing excessively vibrating, become hot and making a lot of noise. If it is continue to run, it will break the cage and making loose in the rolling elements. Then the elements may run into each other, twisting, turning and welded to one another, until the machine trip on overload. This will lead to serious damage in the shaft area under the bearing [2].

The mathematical analysis:

With the assumption the defective bearing moving elements are moving with unpredictable motion magnitude and direction that proportional to the moving element inertia &speed.

The relation between both coordinate systems is given by the following relationship [3].

Mathematical analysis is to study the effecting forces in the bearing and their impact in vibration readings. This can be done through substituting the bearing physical parameter into bearing general mathematical equation at specific direction to find out the amplitude value that proportional to $\frac{\zeta}{\sin \omega t}$ and the rolling element inertia (I) when ζ is the displacement in X direction.

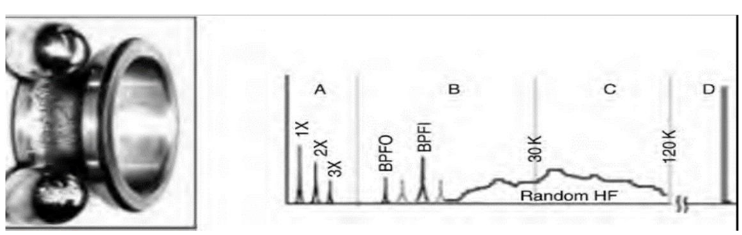

Figure 4: severely damaged bearing in final stage of wear

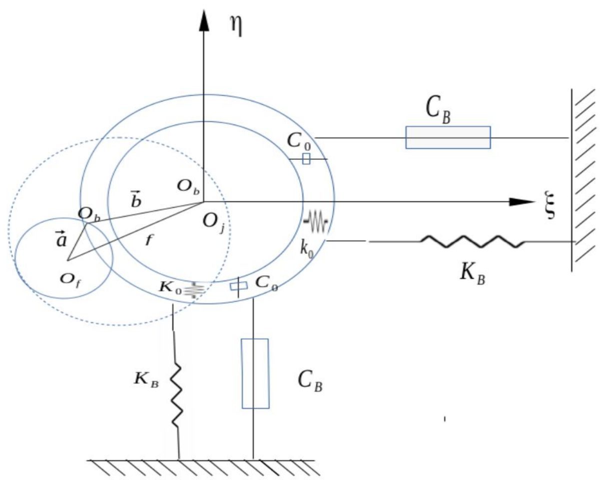

Figure 5: The bearing general mathematical model:

$$\begin{bmatrix} \cos\omega t & -\sin\omega t & 0 & 0 \\ \sin\omega t & \cos\omega t & 0 & 0 \\ 0 & I & 0 & 0 \\ 0 & \cos\omega t & -\sin\omega t & 0 \\ 0 & \sin\omega t & \cos\omega t & 0 \\ 0 & 0 & 0 & I \end{bmatrix} \begin{bmatrix} x \\ y \\ z \\ \theta^x \\ \theta^y \\ \theta^z \end{bmatrix} = \begin{bmatrix} \boldsymbol{\zeta} \\ \boldsymbol{\eta} \\ \boldsymbol{\xi} \\ \boldsymbol{\theta} \\ \boldsymbol{\theta} \\ \boldsymbol{\theta} \end{bmatrix}$$

The case study parameters:

The pump specification is as the following:
Q= 155 m^3/hr, H= 110 m, NPSH=4.1m, Power= 75 kW, N= 2950 rpm.

Bearing parameters:

Parameters	Value
Bearing mass	2.65 kg
Bearing diameter	83 mm
Rolling element diameter	25mm
Rolling element mass	73g

The assumption is the oil film and rolling grooves have lost their function after the last stage of failure that means they lost their damping capability to the rolling ball.

The ball inertia is I=2/5mr^2, The shaft frequency is N= 2950 RPM or CPM, $\boldsymbol{\omega = \frac{2\pi N}{60} = \frac{2\times\pi\times2950}{60} = 308.8}$ **rad/sec.** The substituted matrix is:

$$\begin{bmatrix} \cos 309t & -\sin 309t & 0 & 0 \\ \sin 309t & \cos 309t & 0 & 0 \\ 0 & 4562 & 0 & 0 \\ 0 & \cos 309t & -\sin 309t & 0 \\ 0 & \sin 309t & \cos 309t & 0 \\ 0 & 0 & 0 & 4562 \end{bmatrix} \begin{bmatrix} \boldsymbol{x} \\ \boldsymbol{y} \\ \boldsymbol{z} \\ \boldsymbol{\theta^x} \\ \boldsymbol{\theta^y} \\ \boldsymbol{\theta^z} \end{bmatrix} = \begin{bmatrix} \boldsymbol{\zeta} \\ \boldsymbol{\eta} \\ \boldsymbol{\xi} \\ \boldsymbol{\theta} \\ \boldsymbol{\theta} \\ \boldsymbol{\theta} \end{bmatrix} \quad (1)$$

The assumption is the main forces that has impact in the rolling element vibration in the bearing are the raceway wear because rough surface is force the rolling elements to create motion perpendicular to raceway direction to start the shaft vibration when smooth surface lead to smooth rotating shaft and cage looseness value means the miss control in rolling elements motion that means it's also vibration source according to the assumption bearing has two main parameters affected on its status and vibration reading free rolling element vibration and Oil film damping effect.

The free rolling element vibration calculations

Rolling element moment of inertia with the rotational in horizontal axis (X-axis) has substituted in bearing equilibrium general equation (1) by neglecting the vertical & axial axes to plot the results in (RMS) VS the motor frequency

(CPM) using Matlab program to find the graph in Figure 6 that shows free rolling element vibration spectrum.

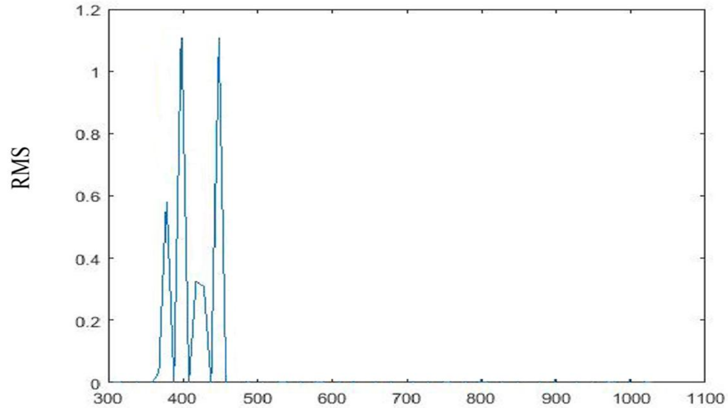

Figure 6: free rolling element vibration spectrum in X – axis.

Oil film (cage) damping effect

The cage function is to control rolling elements movement between inner and outer raceways, for that reason the proper oil film is rolling element vibration damper. Rolling element moment of inertia with the rotational in horizontal axis (X-axis) has substituted the equation (3) that illustrates the damping coefficient proportional to external load, shaft eccentricity and shaft mass by neglecting the vertical &axial axis's to plot the results in (RMS) VS the motor frequency (Hz) using matlab program to find the graph in Figure 7 oil film damping effect on rolling element [5] spectrum

$$\frac{\ddot{x}}{c\omega}+\frac{p}{mc\omega^2}\left(c_{xx}\frac{\dot{x}}{c\omega}+k_{xx}\frac{x}{c}+c_{xy}\frac{\dot{y}}{c\omega}+k_{xy}\frac{y}{c}\right)=\frac{e}{c}\cos\omega \qquad (2)$$ [5]

With assumption that the negligible stiffness effect and shaft mass, the above equation while it is free in x direction is:

$$\mathbf{X}=\frac{e}{[\sin(\omega)+(0.5\times\cos(\omega))]} \qquad (3)$$

The studied bea ring parameters substituted in equation to use the results in plotting it VS the rotational using Matlab program to find the graph in Figure7 that shows the oil film damping effect spectrum on rolling element.

The optimum bearing with good damping coefficient:

The bearing with good damping coefficient means the bearing is running according to motor speed and oil film damping effect. This can explain by adding the plot in Figure 6 to plot in Figure 7 and take the result in Figure 8. The amplitude in plot (8) is almost less than 50% of the free vibration amplitude this can illustrate the new well manufactured bearing status. From this stage the operator or the condition monitoring responsible should start to observe the condition change in vibration amplitude readings without needing to specific analyzing software for explaining the results. Only we can use the bearing manufacturing data for rolling element weight and dimensions to estimate the amplitude height to compare it with vibration measurement device reading to evaluate its status.

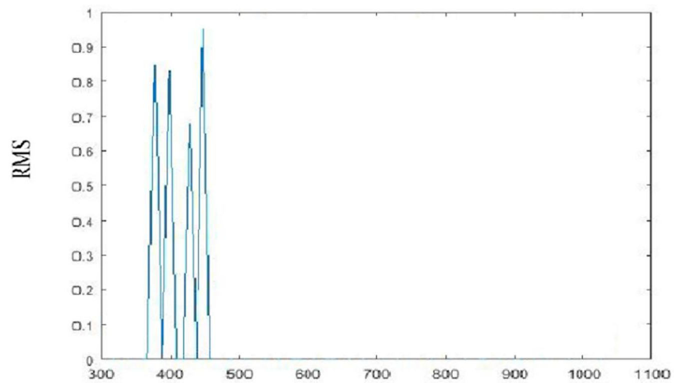

Figure 7: oil film damping effect spectrum on rolling element X axis

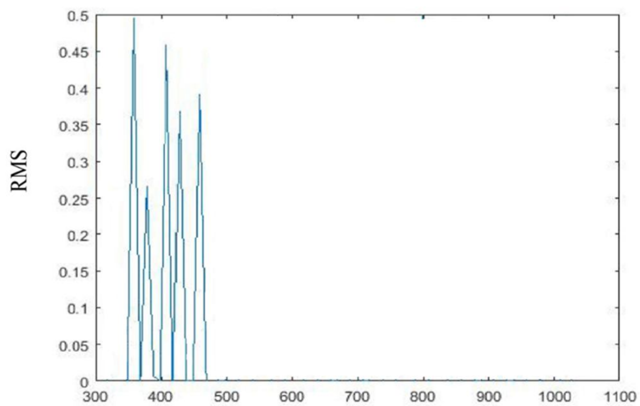

Figure 8: Optimum bearing with good damping coefficient X axis

Experiment analysis of the case study.

Spectrum analysis and deterioration steps:

The first stage:

In the first stage high vibration levels at the high frequency zone only around (150000 rpm) when the shaft speed was (3970 rpm) that less than 5000 rpm has been shown in the spectrum. The assumption is the good contact between the rolling elements and the bearing inner race the vibration root source at high frequency is equal to the diameters ratio between rolling element and the bearing inner race:-

Rolling element frequency = inner race frequency $\times \frac{inner\ race\ radius}{rolling\ ellement\ radius}$

In this stage vibration amplitude rising only in the high frequency because the rolling element still controlled no surface loss in the inner race, outer race or cage and the lubricant oil penetrating easily in between the raceway and its components (cage and rolling elements).

The second stage:-

In this stage there is no change in shaft zone or low frequency zone, but high vibration level appears in wide range in the high frequency zone. This can be explained as multi balls their damping has forgotten with slightly different values.

The last stage of deterioration:

In this stage the vibration level at high frequency zone increase also the vibration readings at low frequency zone has grown up the interpretation is the rolling elements are moving freely with different frequencies between (100000cpm) to (150000 cpm) because the weight losses in moving elements contact areas in the inner race, oil film and outer race. As it is known all rotating component has impact in shaft frequency, the increasing of amplitude in low frequency evidence to unbalancing rotation in bearing's inner race. In this spectrum the vibration levels have increased in high and low frequencies which means the bearing has missed its function.

This is the interpretation of spectrums to explain the bearings failure stages before enough time to damage. The maintenance department is prepared to repair this pump bearing.

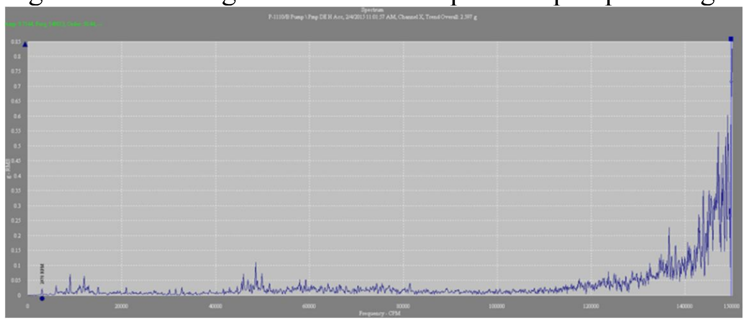

Figure 9: The first stage of deterioration spectrum

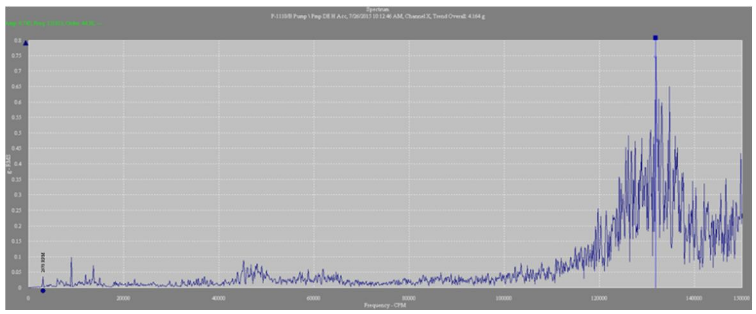

Figure 10: The second stage of deterioration spectrum

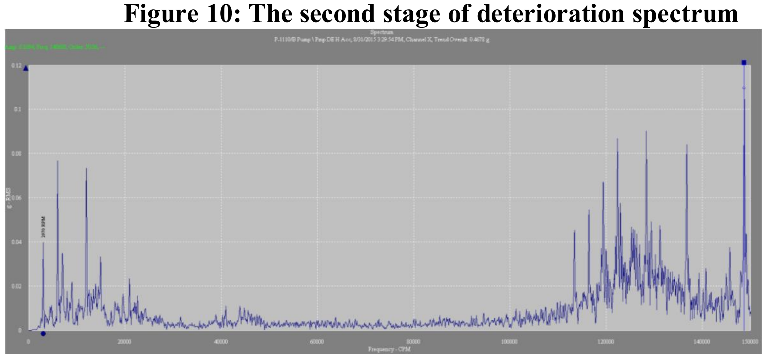

Figure 11: The last stage of deterioration spectrum

***Results*:**

The research calculations find that free bearings rolling elements has the maximum vibration amplitude that equal to (1RMS) , when the cage and lubricant oil working with the best damping coefficient the best vibration damping could be reached ,it equal to (1RMS). After plotting the damping and vibration amplitude VS time in four different frequency zones the research has reached that in all zones the two plots summation give the best vibration readings that equal to (0.5 RMS) after control the rolling elements random movement by the cage and oil damping . That means the good cage and compatible lubricant oil can damp the vibration amplitude 50% if we compare it with in the rolling elements vibration.

CONCLUSION

After the experimental and modeling I compare the results with the calculations results that have been taken from the mathematical model and plotted using Matlab program the results was so similar with close magnitudes values and deviation almost 20%. This deviation was result to calculations and experiment & measurement tools accuracy. Following to these results we can use the mathematical model as technique to evaluate the bearing performance to take correct decision in selecting the bearing changing time because it can illustrate the actual bearing status and its deterioration steps through functional damping coefficient losses this can lead the user to estimate the equipment bearings residual life to prepare the specific maintenance tools, spare parts, expertise technicians and the optimum down time. This can lead to:

1. Optimal Component Replacement Decisions Using Vibration Monitoring
2. The known repair date helps to have stable operation condition.
3. Good production plan which has impact in companies reliable deals and market share.
4. Optimum spare part inventory system that can help the procurement authority to save a lot of money when it pay only for consumable and predictable spare parts that can providing the best storage condition for it with minimum cost.

As per the above reasons that can lead to reduce the plant down time that takes place according to unpredictable maintenance by studying and analyzing vibration signals which can be picked up and their spectrums.

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