VIBRATION SIGNATURE ANALYSIS OF THE BEARINGS FROM FAN UNIT FOR FRESH AIR IN THERMO POWER PLANT REK BITOLA

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Abstract: This paper shows the procedure for prediction of rolling bearings defect in the motor-fan machinery for fresh air used in Thermo Power Plant REK Bitola, R. Macedonia, the biggest energetic institution for manufacturing the electrical power in the country. Detecting the condition of periodical measuring the vibration condition allows early detection and timely intervention renounced by replacing damaged parts and given as a contribution on this part of the predicting maintenance of the Thermo Power Plant.

It will decrease the cost’s for maintains the bearings, save many and avoided some possible failure of the bearings and the whole part of the fan unit for fresh air and increase the probability and efficiency of the Thermo Power Plant.

Keywords: Vibration, Bearing, FFT analysis, bearing failure, thermo power plant

1. Introduction

Mining and Power Company ELEM REK “Bitola” is the largest power generating capacity in Macedonia. It provides more than 70% of power and part of the coal needed for the industry and households in Macedonia.

Social economic importance of the energetic, especially in the last decades, after the oil shock crisis, gets special importance and needs special attention from all citizens and employees to the energy savings from one side and from the employed into energy sector on the other side from which depends safety and reliability of the operation of the energetic plants, equipment life cycle and the price of generated power (1).

Power Generation as all other technologies is a common effort of multidisciplinary specialists. Its efficient operation mostly depends from the quality and reliability of the technology and from the people operating and maintaining the equipment.

Power production process in ELEM REK “Bitola” is organized in two production units: Coal Mine and Thermal Power Plant. Scheme of production technology of electricity from coal mining to the handover of power to consumers is shown in fig 1.

The experience of the maintenance process in REK BITOLA determinates the bearings as mayor sensitive elements in the machinery. Reliability of the rotating machines with rolling bearings strongly depends of the bearings condition. There are many factors influencing bearing life: difference between design and operating conditions, bearing installation, proper lubrication and machines installation(4,5).

Very important is to establish proper method of the monitoring of the bearing condition and to take on time activity to correct situation or to change damaged bearing (2).

Here is presented a method of a rolling bearings condition monitoring and detecting the possible start of failure on the bearing parts.

Bearing vibration analysis can detect lubrication failures, misalignment, out of tolerance running, rubbing, improper gear teeth meshing, out of balance, bent shafts, loose components, worn parts, faulty couplings, improper operating conditions (like pump cavitations) and deflecting support structures (5,7). However to be able to analyses the presence of these type of problems requires a highly skilled person with much experience and exposure to bearing vibration signatures at various stages of failure.

As a fault begins to develop, the vibration produced by the bearing are changes all the time. Every time a rolling element encounters a discontinuity in its path a pulse of vibration results. The resulting pulses of vibration repeat periodically at a rate determined by the location of the discontinuity and by the bearing geometry. These repetition rates are known as the bearing frequencies, more specifically:

• Ball passing frequency of the outer race (BPFO) for a fault on the outer-race
• Ball passing frequency inner race (BPFI) for a fault on the inner-race
• Ball spin frequency (BSF) for a fault on the ball itself
• The fundamental train frequency (FTF) for a fault on the cage.

The bearing frequencies can easily be calculated from the bearing geometry using the formulae given in Fig. 2.
2. Monitoring of bearing vibrations

Monitoring refers to the analysis of vibrations measured at bearings of fan fresh air in REK Bitola. Fan for fresh air consists of the following main parts: body-shaped spiral impeller, shaft with bearings and regulatory apparatus (see fig. 3).

The rolling bearings used in this analysis are bearing number 3: SKF 7338BCBM-angular contact ball bearings, single row (see fig. 4) and bearing number 4: SKF NU 244MA-cylindrical rollers bearings (see fig. 4).

The measuring vibration instrument used the model of SKF CMVA60 working with software PRISM 4. Also, the REK Bitola adopted the ISO 10816 standard by which a certain number and measuring points are set for measuring vibration in horizontal and vertical machines.

To calculate the generate frequency we used the software GEARBOX (see fig. 5).

Fig. 2. Formulas for calculating bearing frequencies

Fig. 3. Fan unit for fresh air BCB type ВДН-32Б

Fig. 4. Bearing SKF 7338 BCBM and bearing NU 244MA

Fig. 5 Calculate general frequency for SKF 7338 and NU244 bearings with the GERABOX
For analysis of the dynamic state of 3VSV-A, a total 99 measurements of vibration were measured on the fan for fresh air branch-A TE Bitola Block 3 (Fig.6). The time period was from 09.06.2003 to 10.03.2011. Each measure has a total of eighteen measurement data. The effective vibration velocity V (rms), measured in an area up to 1000 Hz as defined by ISO standard. Through specter frequency (FFT) are diagnose the causes of vibration (imbalance, misalignments, mechanical looseness, and a combination of them). These reasons are synchronous with the rotor rotation.

Limits on a different machines and setting mechanisms are defined by standard ISO-10816-3.

Analysis of the frequency spectra (FFT) provides signs where defect occurred: at the inner race, of the bearing cage or on the outer race. These defects are not synchronous with the rotation shaft. If there are synchronous peak, it is the same sign that there are mechanical problems present in the spectrum of the vibration speed. Envelope of acceleration technology works in areas up to 1000 Hz. Warning limit is 4 gE, and the limit risk is 10 gE. Vibration acceleration is measured up to 5000 Hz and is useful for early detection of defects in bearings even when in areas of high frequencies.

The following shows just a sample of the overall analysis of the third and fourth bearing. The analysis shows the display of the spectrum in the period before the change bearing (2003-2007), to time when bearing working in failure period (2007-2010), and period after replacement of the bearing with new (2010 till now).

3. RESULTS AND DISCUSSION

Figure 7 shows the spectrum measured for the third bearing SKF 7338, and Fig.8 of fourth bearing NU244MA. Spectrum measured period is on 30 June 2003 until 17 May 2012. This period is classified: 30 June 2003 (period before change bearing), 10 Mart 2010 (period when bearing working in failure period) and 16 May 2011 (period after replacement the bearing with new).

The figure 7 can be seen that the maximum peak in the spectrum is the frequency of 4.25Hz corresponding to the fundamental train frequency FTF=4.24Hz. Also can be noted that the spectrum of frequencies occurs matching first eight FTF harmonics which means that early measurement of vibrations has some damage to the balls of the third bearing and this indicates looseness. This frequency is not encountered very often, but it can occur when some defect affects the rotation of the train. Following are some examples of problems that can cause the generation of the fundamental train frequency.

1. In rare cases when one or more rollers are missing from a bearing, the FTF can be generated. The problem occurs as a pulse at the FTF. The frequency spectra contain a series of harmonics of the FTF. The amplitude of the first harmonic is quite low, the second, third, and fourth harmonics are higher in amplitude as determined by the pulse.

2. Sometimes, attempts to lubricate sealed or shielded bearings can cause the seal or shield to deflect inward. If the cage touches the seal or shield, the FTF and/or two times FTF plus harmonics can be generated.

3. Excessive clearance in an antifriction bearing can cause the generation of a discrete frequency at the FTF and/or modulations of the FTF at rotating speed and harmonics. Except for defects that occur in bearing components during manufacturing, the cage is usually the last component to fail. The typical failure sequence is as follows: defects form on the races, the balls, and then finally the cage. A severely damaged cage can cause constant frequency shifts that are observable with the use of a real-time analyzer. When the cage is broken in enough places to allow the balls or rollers to bunch up, wide shifts in frequencies accompanied by loud noises can occur.
This conclusion can be confirmed by the measured data as shown in fig.7 where noted sizes larger than the value $4gE$ giving warning alarm for mechanical problems in the bearing.

Fig. 7 Spectrum for bearing NU244MA before change period, bearing working in failure period (with FFT) and after replacement the bearing with new

The figure 8 can be seen that the maximum peak in the spectrum is the frequency of 4.35Hz corresponding to the fundamental train frequency $FTF=4.4Hz$. From research it is known that when a bearing is defective rotating elements then the frequency must pass through more mechanical interfaces and balls generate $FTF$ frequency which is rarely presented as a discrete frequency. Frequency modulated in this way generates other frequencies such as frequencies of rotation (10Hz) etc. Since the value of the frequency of rotation is not very large it indicates that the balls are not always loading zone at the outer or inner ring. Also, we may be noted that there is absent the occurrence of BSF frequency because the damaged bearing balls roll is in one direction and spin is in another direction. It can also be observed elevated amplitudes of all other frequencies, suggesting that there are serious problems with the bearing. The spectrum can be seen that the dominant frequency is the rotating frequency of 10Hz with its harmonics, as well as all other frequencies $FTF=4.4Hz$, $BPFI=107.3Hz$, $BPFO=82.7Hz$ and $BSF=38.1Hz$. So, we can observe that after bearings replacement, they are working correct. From presented analysis we can note that the analysis of spectrum requires good professional knowledge of the person performing the analysis, as well as good knowledge of signal processing techniques for analysis.

In further analysis is given vibration condition of the rolling bearing set on the front and the back of the electric motor and a point marked by the point 1 and point 2. Finally, recent measurements at the beginning of 2011 showed a significant increase in the amplitudes of vibration rolling elements and cage (Fig.9). Two types of bearing defects, namely, rolling elements and cage defects were studied. Measurements were carried out on two sets of bearings. The defective bearing was replaced by good bearing after predicting the failure with vibration signal analysis. After dismantling the bearing, the photo showed the place where it caused the failure of the bearing cage (Fig.8).

Defects on the rolling elements (Fig.10) can generate a frequency at twice ball spin frequency and harmonics and the fundamental train frequency. Twice the rolling element spin frequency can be generated when the defect strikes both raceways, but sometimes the frequency may not be this high because the ball is not always in the load zone when the defect strikes and energy is lost as the signal passes through other structural interfaces as it strikes the inner raceway. Also, when a defect on a ball is orientated in the axial direction it will not always contact the inner and outer raceway and therefore may be difficult to detect.

Fig.9. Damaged cage of the SKF 7338 rolling bearing

The bearing cage tends to rotate at, typically, 0.4 times inner ring speed, has a low mass and, therefore, unless there is a defect from the manufacturing process, is generally not visible. Unlike raceway defects, cage failures do not usually excite specific ringing frequencies and this limits the effectiveness of the envelope spectrum. In the case of cage failure (Fig.9), the signature is likely to have random bursts of vibration as the balls slide and the cage starts to wear or deform and a wide band of frequencies is likely to occur.

Fig.10. Damaged rolling elements of the SKF NU244

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4. CONCLUSION

Trend of overall frequencies and vibration spectrum provide useful information to analyze defects in roller bearings. Trend indicates severity of vibration in defective bearings. Vibration domain spectrum identifies amplitudes corresponding to defect frequencies and enables to predict presence of defects on inner race and outer race of roller bearings. The distinct and different behavior of vibration signals from bearings with inner race defect and outer race defect helps in identifying the defects in roller bearings. In the analysis problem, the defect is rolling bearing cage. The bearing cage tends to rotate at typically 0.4 times inner ring
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When deep fatigue spalls are present, pulses are generated. The FTF contains several harmonics when pulses are present. Etching, corrosion, and fluting can also cause many harmonics.

Bearings in rotating machinery should be periodically checked with a frequency spectrum and time signal to detect and study developing defects on the outer and inner races. An accurate method for the calculation of bearing defect length is needed to allow a quantitative determination of the defect severity. With the defect size and progression of development determined, the remaining bearing life can be estimated.

From the conducted analyzes and made the conclusion that the rolling bearings 3 and 4 mounted on the shaft of the impeller fan for fresh air to the steam generators in the Bitola stayed very small life of exploitation in terms of projected (only 25%) which makes them quite uneconomical and quite ineffective in operation.

Given their size, cost and other data related to the cost of assembly-disassembly, disruption of operations, it is evident that haphazardly increased costs of exploitation of the system.

This condition is a consequence of defective assembly for overhaul in 2002, which cause imbalance and misalignment the shaft, further drastically loaded bearings. This is particularly important conclusion for bearing number 4 which is a rolling bearing with cylindrical rolling bodies not suffer greater slope for increased deformation of the shaft. Accordingly, we recommend using the same bearing dimensions but another type of rolling bodies’ breaker, the type that has the ability to automatically compensate for the increased deformations expressed through large slope in the supports.

REFERENCES


