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AN CONTRIBUTION TO THE CONDITION MONITORING OF FANS

Summary. One reason for failure of fan is suppression to float a shaft's bearing in the bearing housings. As a result of this disorder leads to blocking axial displacement of bearing, thrust to the growth of the forces acting on the bearing, and then to his accident. The article deals with the identification of the symptoms of this condition in the data from diagnostic measurements and presents the results of practice.
Keywords. Condition monitoring, fan, statically indeterminate rotor, axial direction.

PRZYCZYNEK DO DIAGNOSTYKI WIBRACYJNEJ WENTYLATORÓW

Streszczenie. Jedną z przyczyn awarii wentylatora jest uszkodzenie podpory łożyska w jego obudowie. W wyniku tej awarii dochodzi do osiowego zablokowania łożyska, wzrostu sił osiowych działających na łożysko, a następnie do jego zepsucia. Artykuł jest poświęcony identyfikowaniu symptomów tego stanu w danych pochodzących z pomiarów diagnostycznych oraz prezentuje wyniki badań empirycznych.
Słowa kluczowe. Diagnostyka wibracyjna, wentylator, wirnik niewyważony statycznie, kierunek osiowy.

1. INTRODUCTION

Ball bearings mounted in pillow blocks are used to support rotating shafts in mechanical equipment. A typical fan has two bearings supporting the shaft. In most cases, the bearing exposed to the highest radial load should be fixed, or axially held, within the housing. Also, the axial force needed to float a bearing is lower when the radial load is lower (axial force to move bearing = coefficient of sliding friction x radial load). The fan's other bearing should be allowed to float internally or within the housing bore, thus accommodating shaft expansion and contraction, [1].

Fig. 1. Overhung centrifugal fans
Rys. 1. Wentylator odśrodkowy

In the case the one bearing do not allowed to float the bearing rotor than the system is statically indeterminate. The axial force increases load of bearing and reduces bearings lifetime. The paper

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deals with problem of identification of statically indeterminate rotor of fan and crusher/hammer mill an presents results of real case study.

At the Fig. 1 is object of our measurement – overhung centrifugal fan exhausts of gas from the filter.

2. EXPERIMENTAL MEASUREMENT AND RESULTS

The fan was included in condition monitoring programme. A SKF Microlog Analyzer CMVA55 and piezoelectric industrial accelerometer was used to measure of overall vibration trend data / effective (RMS) value of vibration velocity. These equipments were used to the special measurements of phase with optical phase reference kit too. Basic information on the processing and interpretation of the measured data in condition monitoring can be found in [2-4].

The measurement on the fan’s bearing houses were done in three direction (horizontal, vertical, axial) according ISO 10816-1. In the Fig. 3 – 9 are depicted overall trend data of velocity and vibration spectrums of velocity in these different directions and in two time:

- Failure stage (failure), axial displacement of rotor was blocked, at 04/28/2011,
- Operate stage (after repair), axial displacement of rotor is able, after 04/28/2011.

The symptoms generalized on the base of measurement or description of failure stage in measure results:

1. Multiple vibration increase of bearings in the axial direction at failure stage. In our case, the increase was approximately four times, in Fig. 2.
   a. In Fig. 2 is trend of electromotor axial vibration by coupling with increasing vibration in this direction from 5,7 mms^{-1} up to 23,3mms^{-1}.
   b. In Fig. 8 is trend of fan bearing vibration by coupling in axial vibration with increasing from 5,2 mms^{-1} up to 20,4 mms^{-1}.
   c. In Fig. 6 is trend of fan bearing vibration by impeller in axial vibration with increasing from 5,1 mms^{-1} up to 21,1 mms^{-1}.
2. The growth of vibration in the rotor bearing houses by coupling other than axial direction is depicted in Fig. 3 and Fig. 4.
   a. In Fig. 3 is trend of fan bearing vibration by coupling in horizontal direction with increasing from 3,9 mms^{-1} up to 8,7 mms^{-1}. The increase however is significantly lower in horizontal direction than in an axial direction.
   b. In Fig. 4 is trend of fan bearing vibration by coupling in vertical direction with increasing from 3,5 mms^{-1} up to 6,2 mms^{-1}. The increase however is significantly lower in horizontal and vertical directions than in an axial direction.
3. A velocity spectrum of fan bearing by coupling in axial direction at 04/28/2011 is in Fig. 7. The dominant peak is at fan speed frequency. This peak has caused an increase in the trend in axial direction.
4. A velocity spectrum of fan bearing by coupling in vertical direction at 04/28/2011 is in Fig. 5. The spectrum is interesting occurrence harmonic frequency fan speed. This fact can be interpreted as the growth of vibration in the radial direction (the response of the system) involved non-linear properties and not fan speed frequency as in Fig. 8. This can be regarded as the accompanying symptom of failure.
5. The high peak of the effective velocity in spectrum (at fan speed frequency) can be caused by imbalance for examples. However, for the imbalance.
   a. A dominant peak at fan speed frequency occurs in the spectrum in a non-axial direction too. This was not observed in the spectra.

b. Trend data are usually not characterized in a sharp increase in the level of vibration.

For the same fault condition as a fan has been done to measure the crusher, according ISO 10816-1. The crusher was also, as in the fan made in three perpendicular directions for Each bearing rack according to ISO 10816-1. In Figures 9 to 18 are displayed velocity spectrums of rms value, at failure condition and after repair. In Tab. 1 are presented values of amplitude s and phases for crusher speed frequency. The trends of velocity vibrations are distorted due to irregularities diagnosis and frequent service calls that have been associated with replacing failure hammers. This dependence is therefore not listed.

The results of measurements (for failure and repair) on the crusher can be interpreted as follows.

1. The peak at crusher speed frequency (at 6.5 Hz) in axial direction is symptom of described failure, Fig. 11 (failure) and Fig. 12 (after repair) for non drive end (NDE).
2. For drive end (DE) of crusher are peak at crusher speed frequency approximately the same, different are second harmonics in Fig. 9 and Fig. 10 in axial direction.
3. In Fig. 13 and Fig. 14 are spectrums for DE and Fig. 15 and Fig. 16 are spectrums for NDE in horizontal direction. The changes of amplitudes at crusher speed frequency are very small.
4. In Fig. 17 and Fig. 18 are spectrums for DE in vertical direction. The change of amplitude at crusher speed frequency is very small too.
5. The measurement of phase at crusher speed frequency was done too. These results served to completion information about considered failure of crusher. The results are in Tab. 1. These results can be interpreted as follows.
   a. The peak at crusher speed frequency is symptom for residual static unbalance too. However, the peak and phase shift after repair is insignificant (the change of phase angle from 30 (horizontal direction) to 260 (vertical direction) and changes amplitudes from 3% (horizontal direction) to 20% (vertical direction).
   b. The problem of shaft bending can be excluded. The points in axial direction at position 9:00 and 15:00 moving in phase.

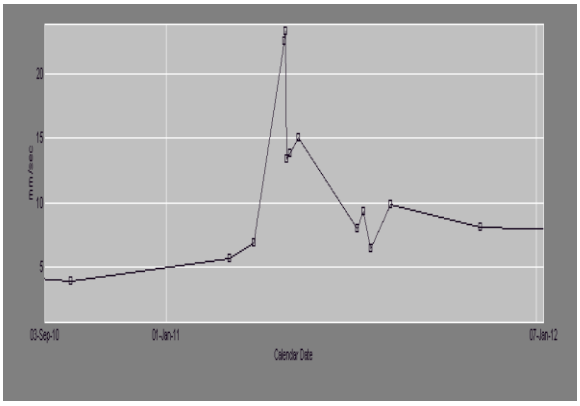

Fig. 2. Trend of electromotor axial vibration by coupling with increasing vibration from 5,7 mms^{-1} up to 23,3 mms^{-1}

Rys. 2. Trend szybkości drgania silnika elektrycznego przy sprzęgle w kierunku osiowym, ze wzrostem wibracji z 5,7 mms^{-1} do 23,3 mms^{-1}

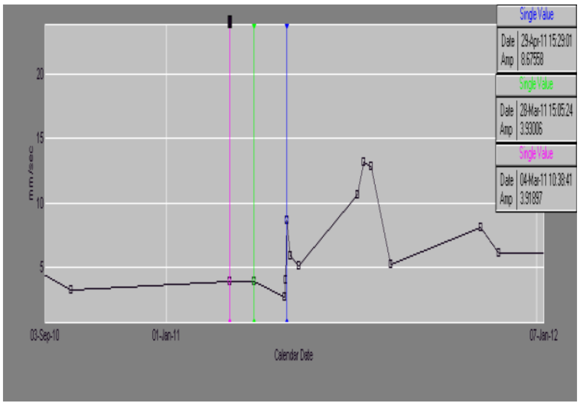

Fig. 3. Trend of fan bearing vibration by coupling in horizontal direction with increasing from 3,9 mms^{-1} up to 8,7 mms^{-1}

Rys. 3. Trend szybkości drgania łożyska wentylatora przy sprzęgle w kierunku poziomym, ze wzrostem wibracji z 3,9 mms^{-1} do 8,7 mms^{-1}

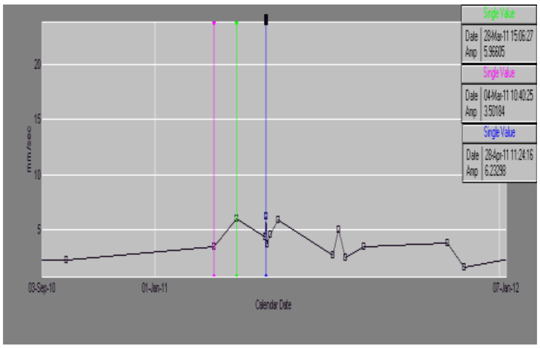

Fig. 4. Trend of fan bearing vibration by coupling in vertical direction with increasing from 3,5 mms^{-1} up to 6,2 mms^{-1}

Rys. 4. Trend szybkości drgania łożyska wentylatora przy sprzęgle w kierunku pionowym, ze wzrostem wibracji z 3,5 mms^{-1} do 6,2 mms^{-1}

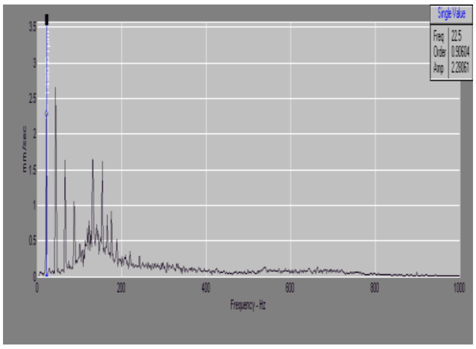

Fig. 5. A velocity spectrum of fan bearing by coupling in vertical direction at 04/28/2011, failure

Rys. 5. Widmo szybkości drgania łożyska wentylatora przy sprzęgle w kierunku pionowym z 28.04.2011, awaria

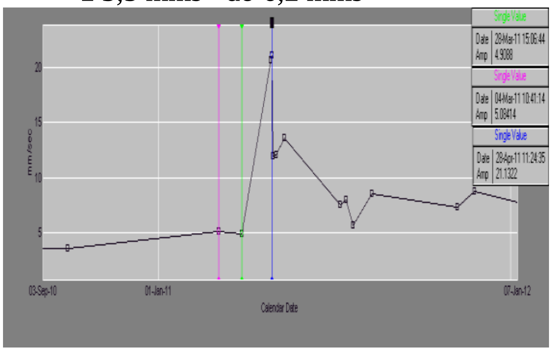

Fig. 6. Trend of fan bearing vibration by impeller in axial vibration with increasing from 5,1 mms^{-1} up to 21,1 mms^{-1}

Rys. 6. Trend szybkości drgania łożyska wentylatora przy kole obiegowym w kierunku osiowym, ze wzrostem wibracji z 5,1 mms^{-1} do 21,1 mms^{-1}

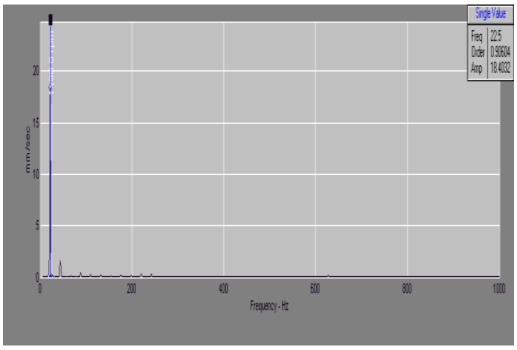

Fig. 7. A velocity spectrum of fan bearing by coupling in axial direction at 04/28/2011. The dominant peak is at fan speed frequency

Rys. 7. Widmo szybkości drgania łożyska wentylatora przy sprzęgle w kierunku osiowym z 28.04.2011. Dominujący szczyt znajduje się przy częstotliwości obrotowej wentylatora

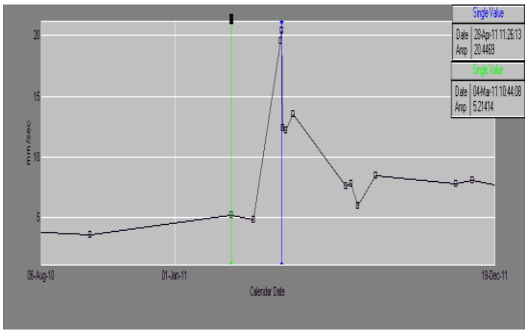

Fig. 8. Trend of fan bearing vibration by coupling in axial vibration with increasing from 5,2 mms^{-1} up to 20,4 mms^{-1}

Rys. 8. Trend szybkości drgania łożyska wentylatora przy sprzęgle w kierunku osiowym, ze wzrostem wibracji z 5,2 mms^{-1} do 20,4 mms^{-1}

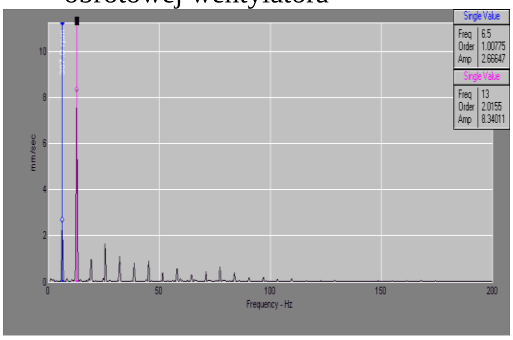

Fig. 9. The crusher, spectrum in axial direction of rotor bearing, DE with harmonics, failure

Rys. 9. Rozdrabniacz, widmo w kierunku osiowym na łożysku wirnika, napędzana część układu mechanicznego, awaria

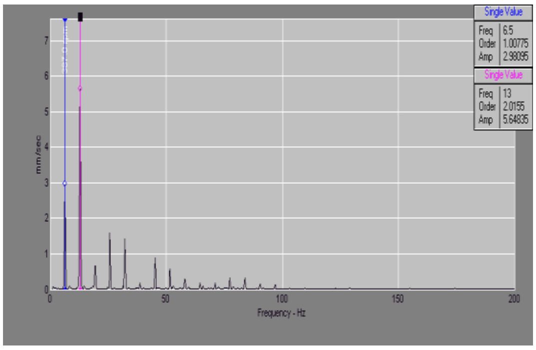

Fig. 10. The crusher, spectrum in axial direction of rotor bearing, DE with harmonics, after failure. Decrease in the second harmonic

Rys. 10. Rozdrabniacz, widmo w kierunku osiowym na łożysku wirnika, napędzana część układu mechanicznego, po naprawie. Spadek drugiej harmonicznej

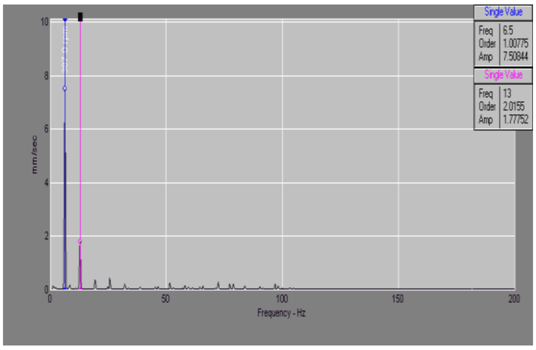

Fig. 11. The peak at crusher speed frequency (at 6.5 Hz) in axial direction of rotor bearing, NDE, failure

Rys.11. Szczyt w widmie przy częstotliwości obrotowej rozdrabniacza (na 6,5 Hz) w kierunku osiowym na łożysku wirnika na nienapędzanej części układu mechanicznego, awaria

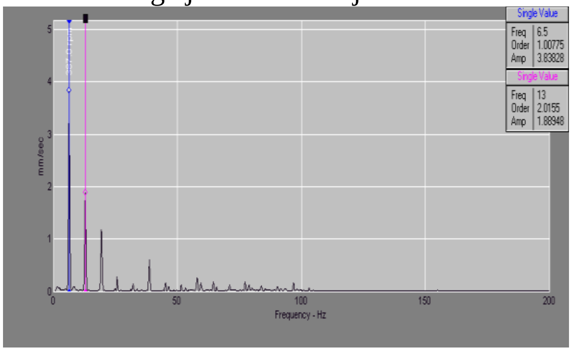

Fig. 12. The peak at crusher speed frequency (at 6.5 Hz) in axial direction of rotor bearing, NDE, after repair

Rys. 12. Szczyt w widmie przy częstotliwości obrotowej rozdrabniacza (6,5 Hz) w kierunku osiowym na łożysku wirnika na nienapędzanej części układu mechanicznego, po naprawie

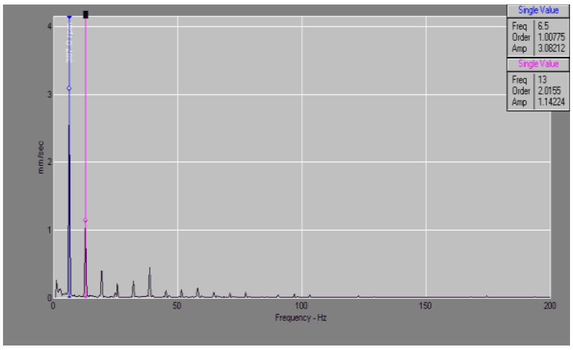

Fig. 13. The crusher, spectrum in horizontal direction of rotor bearing, DE with harmonics, failure

Rys. 13. Rozdrabniacz, widmo w kierunku poziomym na łożysku wirnika, napędzana część układu mechanicznego, awaria

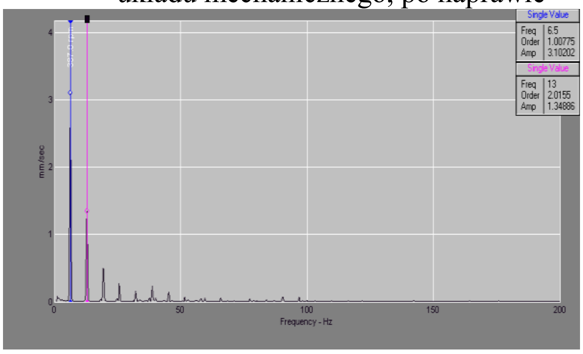

Fig. 14. The crusher, spectrum in horizontal direction of rotor bearing, DE with harmonics, after repair

Rys. 14. Rozdrabniacz, widmo w kierunku poziomym na łożysku wirnika, napędzana część układu mechanicznego, po naprawie

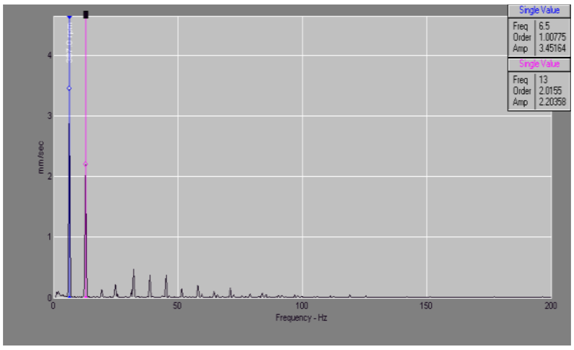

Fig. 15. The crusher, spectrum in horizontal direction of rotor bearing, NDE with harmonics, failure

Rys. 15. Rozdrabniacz, widmo w kierunku poziomym na łożysku wirnika, nienapędzana część układu mechanicznego, awaria

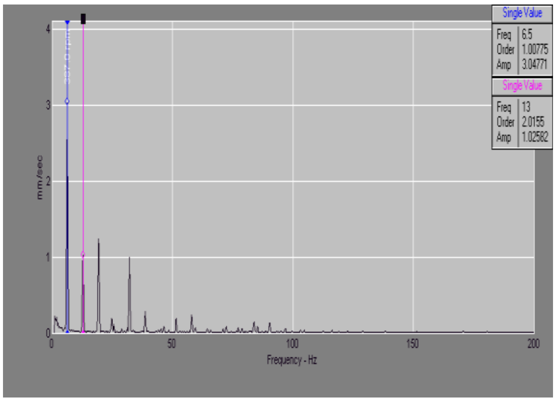

Fig. 16. The crusher, spectrum in horizontal direction of rotor bearing, NDE with harmonics, after repair

Rys.16. Rozdrabniacz, widmo w kierunku poziomym na łożysku wirnika, nienapędzana część układu mechanicznego, po naprawie

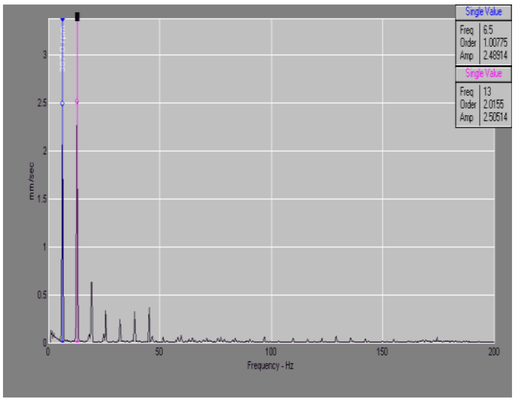

Fig. 17. The crusher, spectrum in vertical direction of rotor bearing, DE with harmonics, failure

Rys. 17. Rozdrabniacz, widmo w kierunku pionowym na łożysku wirnika, napędzana część układu mechanicznego, awaria

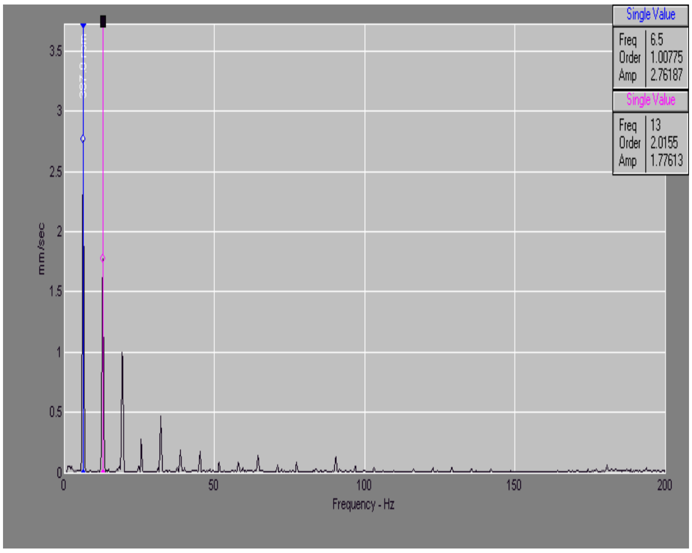

Fig. 18. The crusher, spectrum in vertical direction of rotor bearing, DE with harmonics, after repair.

Rys. 18. Rozdrabniacz, widmo w kierunku pionowym na łożysku wirnika, napędzana część układu mechanicznego, po naprawie

Table 1

The phase measurement on the crusher

		Non drive end				Drive end			
		Direction Position							
		$Axial_{9:00}$	$Axial_{15:00}$	Horizontal	Vertical	$Axial_{9:00}$	$Axial_{15:00}$	Horizontal	Vertical
Failure	Amplitude [mms^{-1}]	2,89	2,87	3,1	2,7	8,45	7,8	3,4	3,3
	Phase [deg]	62	57	341	227	97	87	347	262
After repair	Amplitude [mms^{-1}]	-	-	3,2	2,98	-	-	3,1	2,7
	Phase [deg]	-	-	344	253	-	-	344	256

3. CONCLUSION

An aim of this article was to describe the symptoms of bearing vibration machine with the locked axial motion of the bearing. It is a fault condition characterized by an increase in axial load bearing. An accompanying symptom is an increase in the operating temperature. These conditions led to a rapid reduce in the lifetime of the monitored machine.

It has been shown that the symptoms of this stage to be found in the axial direction, a sharp increase in the measured values of overall trend data of effective velocity of vibration. Trend analysis must be supplemented by the analysis of frequency spectra. In these spectrums should be considered the peak at speed frequency (and second harmonic too). In all directions is presence of harmonics speed frequency. This last symptom can be due to non linearity in system.

In next step we consider useful to make measurements of the fault conditions associated with unbalanced and simulate both conditions in ADAMS program package for different levels of the coefficients of friction bearings and housings in the axial direction and different values of rotor unbalance. In Department of Applied Mechanics of the University of Zilina are experience with such analyzes [5-8].

The paper is contribution to the development of a methodology for evaluating correct identification of faults already under routine vibration diagnosis of rotating machines with roller bearings and in evaluating their spectra. It also provides instructions for programming expert systems with automated multi-parametrical approach to the evaluation of the measured data.

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