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Electromechanical Vibrations of the Vertical Lathe Machining Centre Caused by the Fault of the Working Table Rolling Bearing

In the circumstance of the commissioning of a vertical machining lathe centre, since a previous overhaul due to operative inattention, the rolling bearing of the working table was damaged. When this damaged part was changed, a detection of the machine dynamic response by vibration measurements was also performed. An inacceptable level of vibrations was detected which negatively influences the machining quality. Although it was initially suspected that the higher level of vibrations was solely due to a combination of mechanical causes, an additional analysis performed on frequencies, showing more complex vibrational spectra, suggested a deep interaction between electronical and mechanical criticalities.

Keywords: *Vertical lathe machining centre, electromechanical oscillations, rolling bearings, characteristic frequencies, measurements.*

1. INTRODUCTION

Rolling bearings, along with journal bearings, represent nowadays the most usual way for supporting the moving of machine parts, as shafts in appropriate housing. They consist in inner and outer rings, between which rolling elements are inserted. Rolling bodies are simple geometric shapes, guided in a cage, which prevents their mutual contact, and, at the same time, are uniformly distributed around the circumference of the bearing.

The main task of rolling bearings is to minimize friction and power loss associated with relative motion of machine parts [1, 2].

One of the most common uses of rolling bearings is in machining centres, Figure 1, supporting the vertical lathe machining centre working table, Figure 2. In this and similar applications, they help to guarantee precise [3], stable [4, 5] and controllable movements [6].

These machines are powered by means of electric motors with control units which leave the possibility to change the frequency of stator´s rotational field in order to change rotational speed of the working table, depending on its loading (loaded or unloaded).

These machines are also equipped with gearboxes, components designed for the transmission of the mechanical power from electric motor to the working table. Different processing regimes can also be provided by switching different combinations of gears.

The detection and diagnosis of technical problems can be surely supported by monitoring the vibration levels. But, a proper knowledge on machine dynamics is necessary, especially on the mechanisms of formation [7-9] and propagation of vibrations to the mechanical parts, electrical parts and the mechanical and electrical interaction to produce vibrations, more complex.
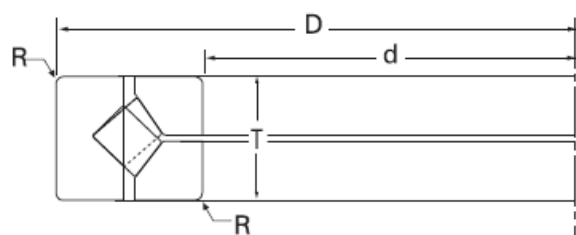

Figure 1. Rolling bearing used on machining centre

Figure 2. Vertical lathe machining centre

The extensive description of the corresponding bearing damage in combination with the benchmark data set for condition monitoring of rolling bearings is given in [10].

Transient response of a flexible Stodola rotor with the flywheel on the free end excited by vibro – impacts due to defective bearing was analysed numerically and experimentally in [11].

The mathematical model which investigates the nonlinear dynamic behaviour of a rotor – bearing

system due to different types of defects of rolling elements, is presented in [12], [13] and [14].

The complexity of rolling bearings vibrations shown through its various sources which can generate characteristic vibration frequencies, often quite difficult to interpret, is described in [15].

Stator and rotor winding faults in three-phase induction machines, measured through the electric current values are reported in [16]. Book [17] and paper [18] consider vibrations connected with electro - motor supply frequency $f = 50$ Hz and its higher harmonics.

Despite the larger number of presented papers connected with mechanically or electrically caused faults in machinery, practically there are not so many papers dealing with coupled mechanical and electrical vibrations, especially connected with rolling bearings and electro - motors as a parts of machining centres.

The theoretical review of relationships that should exist between winding parameters and the mechanical vibrations of AC machines, operating in fault and normal conditions is presented in [19]. Since these machines also uses rolling bearings, its failure can cause the eccentricity in the air gap, which further cause air gap flux density and finally changes in stator current [20]. The results of measurements of vibrations and electric current connected with supply frequency and some machines characteristic mechanical frequencies are given in papers [21, 22] and thesis [23].

In the present paper contribution regarding the explanation of the phenomenon of electro – mechanical vibrations and its experimental identification on the vertical lathe machining centre will be given.

2. MECHANICAL, ELECTRICAL AND ELECTRO – MECHANICAL CAUSES OF VIBRATIONS

In this section the causes of vibrations and relevant appearing frequencies will be described in details.

2.1 Mechanical vibrations caused by rolling bearing faults

Mathematical models to describe the contact between the rolling elements, figure 3 and sliding track as the dominant contribution for the mathematical description of the impact energy that occurs in defective bearing, taking into account the size of the defect, the geometry of the components of the bearing and speed of movement of the bearing shaft.

Figure 3. Mechanical damage of rolling bearing

Frequency content of various bearing faulty conditions is given in equations (1) - (4):

$$f_{FTF} = \frac{1}{2} f_R \left(1 \pm \frac{D_b}{D_c} \cos \beta \right) \qquad (1)$$

$$f_{BSF} = \frac{D_c}{2 D_b} f_R \left(1 - \left(\frac{D_b}{D_c} \cos \beta \right)^2 \right) \qquad (2)$$

$$f_{BPFO} = \frac{N_B}{2} f_R \left(1 - \frac{D_b}{D_c} \cos \beta \right) \qquad (3)$$

and

$$f_{BPFI} = \frac{N_B}{2} f_R \left(1 + \frac{D_b}{D_c} \cos \beta \right). \qquad (4)$$

Sign – in Eq. 1 is true in case when outer ring is stationary, and inner ring is rotating, and + sign is true in case when inner ring is stationary and outer ring rotating. [24]. Also, here it should be noted that in this paper rolling elements taken from the geometric characteristics of ball elements, since the available models in the relevant literature [25, 26] taking into account the taper rolling elements gave almost identical results, i.e. a comparison with the results of experimental analysis, frequency of defects for the ball and the corresponding conical elements are not significantly different.

2.2 Mechanical vibrations caused by shafts and gearbox

Different rotational speeds of the workpiece on the working table can be realized by switching on certain transmission ratios within the gearbox. Thereby, different combinations of gears (which are mounted on the corresponding shafts, Figure 4) are engaged.
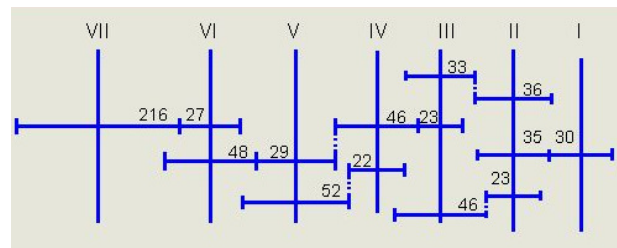

Figure 4. Schematic description of lathe machine gearbox

The frequency content that can be generated due to rotational imbalance on some of these axis coincides with the speed of rotation of shaft misalignment while axis of the two axles usually manifests frequency content in the amount of double speed of rotation, i.e. second harmonic. If there is a defect in the gear teeth, then that expresses frequency content f_{gear} that corresponds to the product of speed and the number of gear teeth which is given in equation (5).

$$f_{gear} = z f_R. \qquad (5)$$

In this procedure, due to unevenness caused by defective gear teeth in the frequency content of the

customary frequency and modulation of this frequency as the vibration signal comprises of a set of additional harmonics at frequencies $\pm kf_R$, which is given as

$$f_{gear} = zf_R \pm kf_R \qquad (6)$$

where $k=1,2,3\ldots$

2.3 Mechanical vibrations caused by machine electrical components faults

Defects of electrical components of machines may occur due to problems in the stator and the rotor of the electric motor. Typical parameters for induction motor is frequency of the input supply voltage f and slip which depends on the difference between frequency of the stator field and the frequency of rotation of the rotor (frequency of the rotor field) and is equal to:

$$s = \frac{f - f_R}{f} . \qquad (7)$$

Symmetrical stator windings supplied by input voltage of the frequency f create stator rotating field of the same frequency (in case of 2 stator poles) which further induces electromotive force of rotor circuits at a frequency sf. The presence of the asymmetry on the rotor side is reflected in the unbalanced rotor currents, which generate the variable magnetic field associated with a component of the counter current at a frequency -sf. This counter sequence current is reflected on the stator side of producing components of the stator current frequency $(1-2s)\,f$. In this way the waveform generated by the disturbance torque and angular velocity at the frequency $2sf$, further modulates the rotary magnetic flux and thus produces two currents and to the earlier mentioned first frequency $(1-2s)\,f$, which reduces the component of the stator current at that frequency, and a second frequency component of the current $(1 + 2s)\,f$, which further propagates disorder. The result of these phenomena at the damage rotor is the new modulation of the power frequency f, in the amount

$$f_{krs} = (1 \pm 2ks)f \qquad (8)$$

on the stator side and in the amount

$$f_{krr} = \pm(1 + 2k)sf \qquad (9)$$

on the rotor side, and k can take any positive integer or zero value, i.e. $k = 0,1,2\ldots$

If there is a defect on the side of the stator windings, there will be the imbalance stator currents, which generate variable rotational magnetic field associated with a component of the counter current at a frequency f. This is the counter sequence current reflected on the rotor side of producing components of the rotor current frequency $(s\text{-}2)\,f$. In this way the waveform disturbance of torque and angular velocity is generated at the frequency $2f$, which then modulates the magnetic flux, thus producing two currents, and to the earlier mentioned first frequency $(s\text{-}2)\,f$, which reduces the component of the rotor current at the same frequency, and a second component current frequency $(s + 2)\,f$, which in the interaction of torque and angular velocity disturbance at

the frequency $2f$ generates electromotive force of the stator to the frequency $3f$, which results in creating two new harmonic frequency $\pm\,3f$. The result of such damage occurrence in the stator, gives the new modulation of the power frequency f, and in the amount of

$$f_{kss} = \pm(1 + 2k)f \qquad (10)$$

on the stator side, and on the rotor side in the amount

$$f_{ksr} = (s \pm 2k)f . \qquad (11)$$

2.4 Mechanical faults of bearings and its influence on frequencies of electrical components of machine

The literature [2] and [10] analyzes cases of interacting vibration signal due to bearing damage and electrical signals measured on an induction machine. If the vibration frequency signal f_{char} can take one of the characteristic frequencies of bearings [1] and Table 1, it acts on the electric motor with wave disturbance torque $\Delta T_i(t)$, which caused a wave disturbance of the angular velocity $\Delta\omega_i(t)$. The angular disturbance causes angular fluctuations of magnetic flux, which in vibratory sense manifests as a component of the torque generated by two electric current components at frequencies

$$f_{be} = |f + kf_{char}| \qquad (12)$$

where $k=0,\pm1, \pm2\ldots$

It is therefore possible to conclude that the defect bearing can generate the stator current at frequency exclusively associated with the frequency of the mechanical properties of defective bearing and the frequency of power supply. This conclusion is true in general, i.e., disturbance in the stator currents can generate electric motors and mechanical disturbances in any of the machine bearings, and not only on the bearing of electric motors, since the mechanical disorder due to deficiency of bearings transmits on the entire structure of the machine, from the deficiency to the power unit.

3. MEASUREMENT PROCEDURE AND MEASURING EQUIPMENT

In the process of measuring the vibration of the lathe's table the three non-contact measuring devices are used (two non-contact displacement sensor IN 085 and one optical phase sensor P-84 measuring spindle speed). The non-contact measuring sensors work on the Eddy current principle, so it is necessary to calibrate them before each measurement (the gap between the top of the sensor and lathe's table should be adjusted). The optical phase sensor work on the principle of emission of light beam and the detection of reflected beam from the reference marker which should be applied on the surface of the lathe table, Figure 5.

To measure the vibrations, eight channel measuring device based on a portable computer and PCI measuring card NI 4472 is used. This measuring card works with the sampling frequency of up to 102.4 kS/s. The measurement results are analyzed in specially prepared

applications within software packages LabVIEW and Matlab/Simulink. The placement of the non-contact sensors (S1 and S2) as well as optical phase sensor (REF) in relation to the lathe working table are shown in the Figure 6.

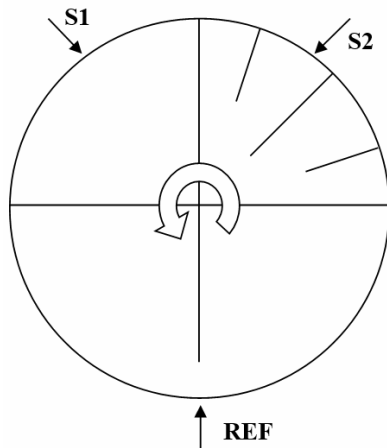**Figure 5. Working table with measuring equipment**

Figure 6. Position of sensors around of working table

4. VIBRATION CHARACTERISTICS OF THE VERTICAL LATHE MACHINING CENTER

Capacity of this vertical lathe machine is 20 t load, diameter of working table is 2 m, power of spindle for turning is 60 kW, for milling and drilling is 22 kW. Lathe machine is numerically controlled, where possible speed of rotation of the working table is in the range of 2-200 rpm. Worktable rolling bearing pitch diameter is equal to 0.88 m and mean diameter of the tapered roller is 0,046 m. Number of rolling elements in the bearing is 44 of which 22 were damaged. Contact angle of the rollers is 45°.

For such defined input parameters, according to equations (1)…(4) it is possible to determine four characteristic natural frequencies for working table speed of 200 rpm, and they are f_{FTF} = 1.728 Hz, f_{BSF} = 31.841 Hz, f_{BPFO} = 35.311 Hz, f_{BPFI} = 38.022 Hz. View of machine speed and gear ratios of individual axle gearbox (figure 4), is given in Table 1. According to information from the user of the machine, frequency of the stator rotating field is equal f = 60,89 Hz. In the case when working table is unloaded, according to equation

(7) slip is equal to s = 0.021, i.e. 2,1%. Frequencies of electromechanical vibrations f_{be} calculated from (12), are shown in Table 2.

Table 1. Rotation speeds and transmission ratios

Label of shaft	Frequency Hz	Gear ratio
n_I	59.51	
		0.857
n_{II}	51.01	
		1.09
n_{III}	55.65	
		0.5
n_{IV}	27.82	
		1.586
n_V	44.13	
		0.6
n_{VI}	26.66	
		0.125
n_{VII}	3.33	

Table 2. Electromechanical vibration frequencies f_{be} in relationship with fundamental train frequency f_{FTF}

Parameter k	Stator rotating field frequency f Hz	Fundamental train frequency f_{FTF} Hz	Frequency of the electromechanical vibrations f_{be} Hz
0			60,89
1			62.62
-1			59.16
2			64.35
-2	60.89	1.728	57.43
3			66.08
-3			55.71
4			67.80
-4			53.98

5. MEASUREMENT RESULT ANALYSIS

Based on the description of damage, the focus of measu–rement was diagnosis of a working table bearing fault, and thus measurements taken at different positions of the bearing housing and on the bearing during the rotation of the unloaded working table. Except mentioned mecha–nical cause it was determined the existence of other exci–tation mechanisms such as the gearbox and drive system, which can be seen in the frequency spectra given below.
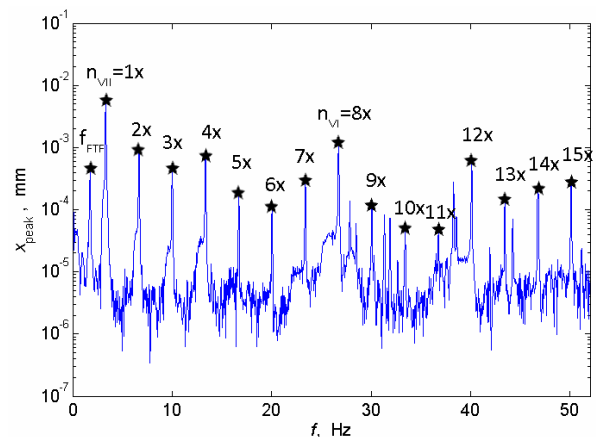

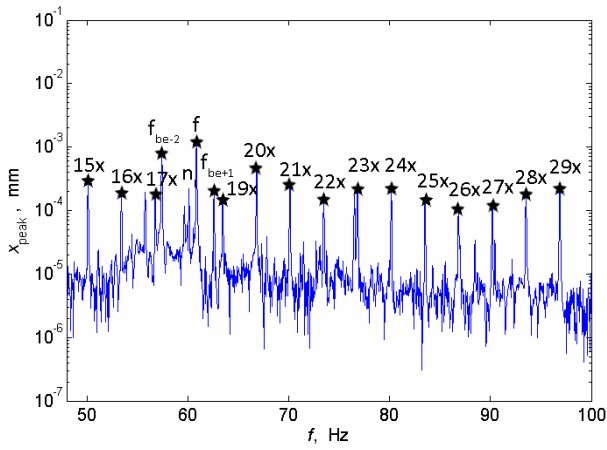**Figure 7. Displacements in frequency range 0-50 Hz**
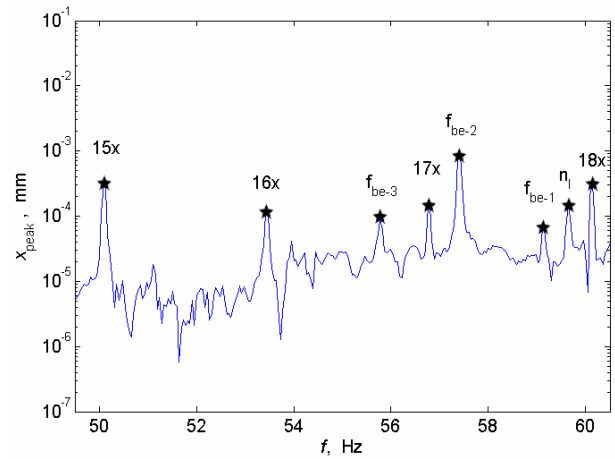
Figure 8. Displacements in frequency range 50-100 Hz

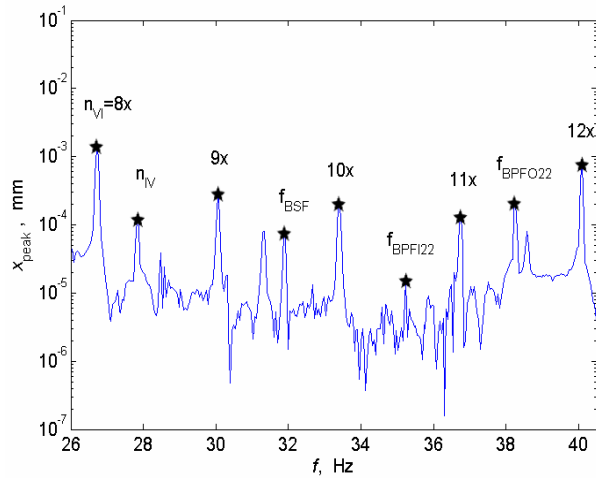**Figure 9. Displacements in frequency range 26-40 Hz**

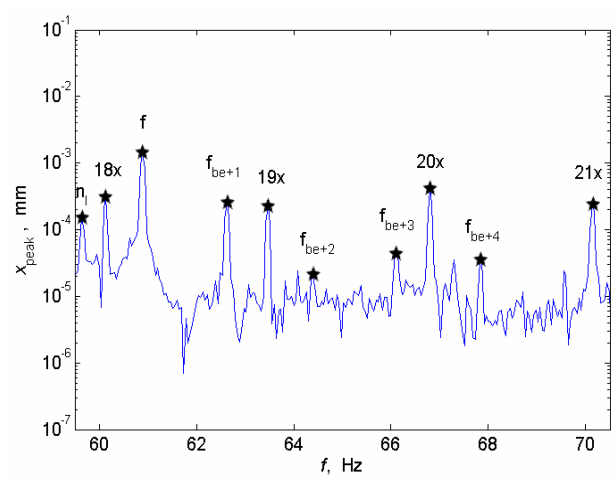**Figure 11. Displacements in frequency range 50-60 Hz**

Figure 12. Displacements in frequency range 60-70 Hz

In particular, on the figures 7 and 8, the overview of the vibration frequencies content from 0 – 100 Hz is shown. The high magnitude of the bearing's f_{FTF} (Fundamental Train Frequency is visible and this frequency will be used as characteristic frequency f_{char}, connected with appearance of coupled electro – mechanical vibrations. Also the rotational frequency of the working table marked as 1x as well as its higher harmonics are show too. Mentioned frequency spectrum also contains the rotational frequencies connected with the specific unbalance of some shafts (n_{VI} and n_I) from the lathe machine gearbox.
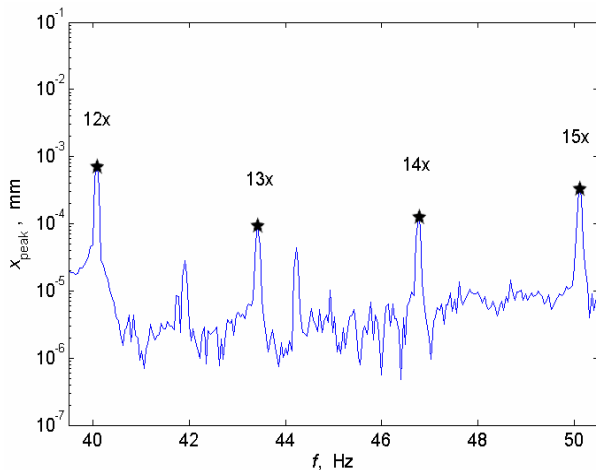

Figure 10. Displacements in frequency range 40-50 Hz

It is also noteworthy, the mentioned frequency spectrums contain a frequencies connected with electro – mechanical vibrations (f_{BE-2} (parameter k=-2 in the table 2), f and f_{BE+1} (parameter k=+1 in the table 2)). These vibrations induced by electro – mechanical cause has very important influence on the whole level of vibrations. The reason why this frequencies has greater importance than those lower connected with the working table rotation is the fact that measurement of vibration displacement (like in this case) emphasize the content of lower frequencies.

Frequency content for identification of damaged stator and rotor windings of electric motors f_{krs}, f_{kss}, f_{krr} and f_{ksr} (basic modulation frequency power due to damage to the stator and rotor windings) was insignificant and therefore no damage to the electric motor is expected. This is primarily related to the frequencies due to damages of rotor's windings, which could appear in the shown frequency spectrum (0-100 Hz). It certainly exceeds frequencies due to damages of stator's windings (above 120 Hz), as well as the frequencies connected with the teeth damages of the gears from the gearbox (above 700 Hz). But, with additional analysis of the higher frequency areas, to which belong symptoms connected with such as failures, it is not noticed any frequency content indicating to this type of faults.

For better insight into the specific vibration content connected to the effects described in the section 2, the

frequency measuring domain is divided to the smaller areas, Figures 9-13. On the Figure 9, showing the displacements in the frequency range of 26-40 Hz, the remaining three characteristic frequencies of roller bearing: BSF (Ball Spin Frequency), BPFO22 (Ball Pass Frequency Outer Race for 22 rolling elements) and BPFI (Ball Pass Frequency Inner Race for 22 rolling elements), are visible. These diagrams contain also the frequencies of two rotating shafts (n_{VI} and n_{IV}) as well as 9-11 harmonics of the working table rotational speed.
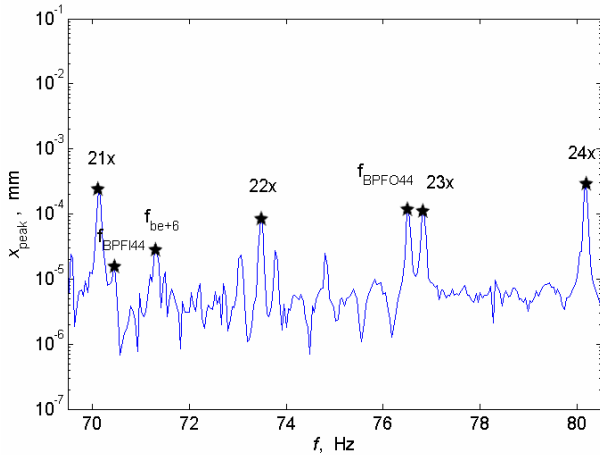

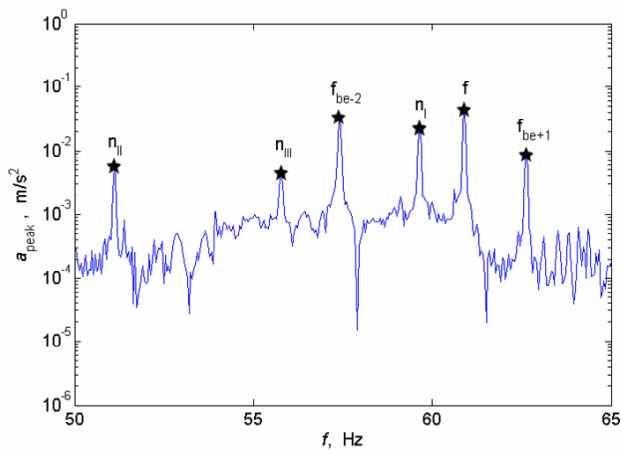**Figure 13. Displacements in frequency range 70-80 Hz**

Figure 14. Accelerations in frequency range 50-65 Hz – contact measurements on foundation

In addition to the above-mentioned harmonics whose values correspond to a situation where the load is transferred over half of rolling elements (22 rollers) it is noticed that harmonic correspond to the case where load is transferred trough all rollers (44 rollers), in while BSF remains the same and BPFO44 and BPFI44 are twice than earlier mentioned BPFO22 and BPFI22. These frequencies are also shown on Figure 13 where the frequency range of 70-80 Hz is reported.

As it is mentioned before, a very important influence show the frequencies connected with electro – mechanical vibrations f_{BE-2}, f and f_{BE+1} which can be found, with significant magnitude, also on the Figures 11 and 12, showing, respectively, the frequency range of 50-60 Hz and 60-70 Hz. To further investigate this phenomenon, additional measurements is performed. Unlike previous described measurements which were performed with non – contact measuring sensors on the working table, in this

case the vibration acceleration is measured on the lathe machine foundation. Two contact sensors Bruel&Kjaer AS020, positioned on foundation near the faulted working table bearing, are used in that measurement.

Measurement of the acceleration frequency spectrum between 50 and 65 Hz, which results are given on the Fig. 14 confirmed the higher frequency content connected with electro – mechanic cause of vibrations, while the rotating shafts frequencies (n_I, n_{II} and n_{III}) are also presented, but to a lesser extent. The appearance of new frequencies connected with the shaft's unbalances, which did not appear in the spectrums before, it is possible to interpret with the fact that the mechanical disturbance is transferred from the place of origin, further through the overall structure of the machine. Therefore, in some specific measuring places, individual characteristic frequencies will be more or less pronounced.

Because of this transfer principle of the disturbance through the entire structure of the machine, disturbances in the electromotor's stator currents can be generated by mechanical disturbances in any of the machine bearings, not only on the electromotor's bearings. It is therefore po–ssible to conclude that the rolling bearing's defect can generate the stator currents at that frequency, exclusively connected with the rolling bearing's characteristic frequ–ency (depends of its mechanical and geometric properties) and the power supply frequency. However, unlike the previously mentioned mechanical and electrical causes of disturbances, whose vibration intensity level is lower value and dependent on the place of measurement, based on such measurements is observed the phenomenon that the vibration resulting from the electro - mechanical causes can be measured in a rather high amounts in all positions on the structure, as well as on contact or contactless manner.

6. CONCLUSION

Damage to individual components processing machinery are the most common causes of excessive vibrations that occur in the machine processing of metals, and result in unacceptably poor surface quality.

By measuring and analyzing the frequency spectra of vibrations it is possible to determine the cause of vibration that is to detect damage to the individual machine component.

In this paper the analysis is done of the vibration system in which the cause of the failure was suspeted to be mechanical, but the analysis of experimental data found interesting fact that the frequency content with the highest level of vibration fit those frequencies which indicated the coupling of mechanical and electrical parts of the system, acting on each other and producing unwanted vibration level. This effect to a lesser extent was theoretically described in the relevant literature, but its experimental confirmation is not so often described.

Further work will be focused on the development of numerical multibody dynamics model capable of simulation of the described phenomena.

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NOMENCLATURE

D_c pitch diametar, mm
D_b ball (roller) diametar, mm
f stator rotational field frequency, Hz
f_{be} forward and backward frequency components, Hz
f_{BPFO} ball (roller) pass frequency of outer race, Hz
f_{BPFI} ball (roller) pass frequency of inner race, Hz
f_{BSF} ball (roller) spin frequency, Hz
f_{char} characteristic frequency of bearing, Hz
f_{FTF} fundamental train frequency or cage rotational frequency, Hz
f_{gear} gear rotational frequency, Hz
f_{krs} modulation of stator frequency due to rotor damage, Hz
f_{kss} modulation of stator frequency due to stator damage, Hz
f_{krr} modulation of rotor frequency due to rotor damage, Hz
f_{ksr} modulation of rotor frequency due to stator damage, Hz
f_r rotor frequency, Hz
f_t rotational frequency of table, Hz
n_i rotational speed of the ith shaft, rpm
N_B number of ball
s slip
T_i shaft torque, Nm
z number of teeths of gear

Greek symbols (Times New Roman 10 pt, bold, italic)

β ball (roller) contact angle, °
ω_i shaft angular velocity, s^{-1}

ЕЛЕКТРОМЕХАНИЧКИ ОСЦИЛАЦИЈЕ ВЕРТИКАЛНОГ МАШИНСКОГ СТРУЖНОГ ЦЕНТРА НАСТАЛЕ ЗБОГ ГРЕШКЕ У РАДНОМ СТОЛУ СА ЛЕЖАЈЕВИМА

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Током пуштања у рад вертикалног машинског стружног центра у претходном ремонту, због непажње, котрљајни лежајеви на радном столу су оштећени. Када је оштећени део замењен, извршено је чишћење са мерењем вибрација. Неприхватљив ниво ниво вибрације је измерен, што представља негативан утицај на квалитет обраде.

Иако се првобитно сумњало да је виши ниво вибрација повезан искључиво са механичким узроцима садржаја фреквенције, детаљнија анализа је показала изглед сложенијих вибрација, такозване електро - механичке вибрације.