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INVESTIGATION OF THE INFLUENCE OF THE WEARING AND THE BELT DRIVE PARAMETERS' CHANGES OVER THE FORCED TORSIONAL VIBRATIONS IN THE SAW UNIT OF A WOOD SHAPER

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ABSTRACT

The proposed study presents an investigation of the influence of the wearing and the belt drive parameters' changes over the forced torsional vibrations in the saw unit of a wood shaper. These changes affect the elastic and damping coefficients of the machines' elements. The research is done on the base of a concrete mechanic-mathematical model for numerical investigations of the torsional vibrations of a wood shaper's saw developed by the authors. The main features in the construction of the wood shapers are rendered an account in this model. Work's conditions in the practice are modeled with the variable moments on the drive electric motor and the wood shaper's saw. The conclusions based on the numerical investigations are confirmed by the real conditions.

The results of the whole investigation are applicable to the direct well-founded recommendations concerning the operation of these machines. These recommendations are important for the increase of the accuracy and quality wood shapers' production. At the same time the results of this research are useful for technical diagnostics of the wood shapers.

Key words: wood shapers, modeling, torsional vibrations.

1. INTRODUCTION

One of the reasons for increasing of the torsional vibrations' level of the machines with belt drives is the wearing and the changes of the belt drives parameters. Practically, the parameters of the belt often become worst although a damage or failure in its visible state has not started. This fact reflects the work of the whole machine. Principally, the investigation of the causes for origin and increase of the torsional vibration of the wood shapers requires understanding of the essence of the dynamic processes in them, when the machine works (Coutinho 2001). It is necessary to conduct purposeful studies in which the machine can be considered as a mechanical vibrating system with known characteristics of its individual elements (Veits *at al*., 1971). For this purpose, firstly it is necessary to have mechanical-mathematical modeling and composing of equations describing the vibration of the elements of the wood shaper. Well-targeted research can be done by solving these equations in different conditions. Some recommendations for the construction's design and the work regimes of the machine are formed on their base (Bachev *at al*., 2012).

The kind of wood shapers that are commonly used in the practice of forestry industry (Filipov 1977) are examined in the proposed study. Fig. 1 shows the general view, and Fig. 2 – a scheme of this type of wood shapers (Obreshkov 1997). The machine body is marked with *1*, *2* is the electric motor, *3* – the belt drive, *4* – the spindle with the bearings, *5* – the arbor with morse cone, *6* – the work table, *7* – wood shaper's saw.

Some special features in the modeling of woodworking machines are examined in previous papers of the authors (Vukov *at al.*, 2010, Vukov 2008).

The aim of this study is to make numerical investigations of the influence of the wearing and the belt drive parameters' changes over the forced torsional vibrations in the saw unit of a wood shaper. This aim requires studying the effect of the changes of the elastic and damping properties of the belt drive on the forming of the torsional vibrations in this machine. The investigation is done on the base of an adequate mechanic-mathematical model for investigation of the torsional vibrations of the wood shapers developed by the authors. The model presents features in the construction of a kind of wood shapers with lower spindle. The real work's conditions are modeled with presence of the variable moments of the drive electric motor and the wood shaper's saw.

Figure 1: Wood shape – general view

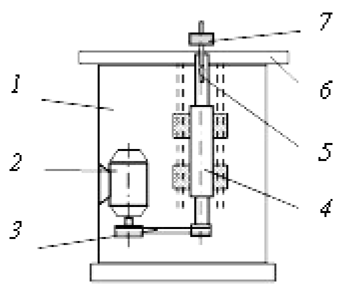

Figure 2: Scheme of the wood shaper

2. MECHANIC-MATHEMATICAL MODEL

A mechanic-mathematical model for investigation of the dynamical processes and vibrations in the wood shaper's saw unit is built by the authors. The model is shown on the fig. 3. This model includes four discrete mass connected with three massless elastic elements. φ_i, $i = 1, 2, 3, 4$ are the angles of the rotation of the corresponding rotor. The elasticity coefficients of the electric motor's shaft, the belt and the spindle are taken into account. The elasticity angular coefficient of the electric motor's shaft is marked with c_1, and this one of the spindle – with c_3 (*N.m/rad*). The elasticity linear coefficients of the two parts of the belt between the belt puller are c_{23} and c_{32} (*N/m*). The damping coefficients are marked with *b* and respective indices. The applied moments M_i on the disks are shown too.

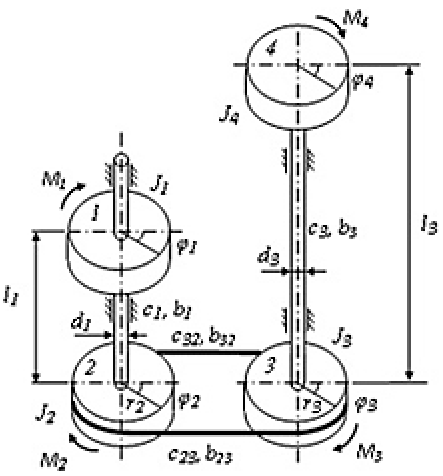

Figure 3: Mechanic-mathematical model

The necessary reduced mass inertia moments ($kg{\cdot}m^2$) are rendered an account (fig. 3): They are J_1 – the mass inertia moment of the electric motor's rotor; J_2 – the mass inertia moment of the belt puller on the electric motor's shaft; J_3 – the mass inertia moment of the belt puller on the spindle; J_4 – the mass inertia moment of the wood shaper's saw with cutter arbor.

Some other symbols on fig. 3 are: d_1, d_3 – diameters of the electric motor's shaft and spindle (m); l_1, l_3 – computing length of the electric motor's shaft and spindle (m); r_2, r_3 – radius of the belt pullers on the electric motor's shaft and spindle (m); G – modulus of shearing.

The investigation of the torsional vibrations of the wood shaper's saw unit requires formulation and solution of the differential equations which describe these processes. The Lagrange's method is used combined with the priority of the matrix mechanics (Angelov and Slavov 2010). This method supposes receiving a system of parametric linear differential equations which describe the small forced torsional vibrations of the saw unit. They are

$$\mathbf{M}.\ddot{\mathbf{q}} + \mathbf{B}.\dot{\mathbf{q}} + \mathbf{C}.\mathbf{q} = \mathbf{Q}, \qquad \textbf{(1)}$$

where:

q is the vector of the generalized coordinates, $\mathbf{q} = [\varphi_1 \quad \varphi_2 \quad \varphi_3 \quad \varphi_4]^T$,

$\mathbf{Q} = [M_1 \quad -M_2 \quad -M_3 \quad -M_4]^T$ – the vector of the generalized loads includes all torsional moments, applied to the rotors,

M, **B**, **C** are respectively the matrix, which characterize the mass-inertial properties of the mechanical system, the damping properties and the elastic properties

$$\mathbf{M} = [a_{ij}], \quad a_{ij} = \frac{\partial^2 E_K}{\partial \dot{q}_i . \partial \dot{q}_j}, \quad \mathbf{M} = \begin{bmatrix} J_1 & 0 & 0 & 0 \\ 0 & J_2 & 0 & 0 \\ 0 & 0 & J_3 & 0 \\ 0 & 0 & 0 & J_4 \end{bmatrix}, \qquad \textbf{(2)}$$

$$\mathbf{B} = [b_{ij}], b_{ij} = \frac{\partial^2 F}{\partial \dot{q}_i . \partial \dot{q}_j}, \mathbf{B} = \begin{bmatrix} b_1 & -b_1 & 0 & 0 \\ -b_1 & b_1 + b_{23}.r_2^2 + b_{32}.r_2^2 & -b_{23}.r_2.r_3 - b_{32}.r_2.r_3 & 0 \\ 0 & -b_{23}.r_2.r_3 - b_{32}.r_2.r_3 & b_3 + b_{23}.r_3^2 + b_{32}.r_3^2 & -b_3 \\ 0 & 0 & -b_3 & b_3 \end{bmatrix}, \qquad \textbf{(3)}$$

$$\mathbf{C} = [c_{ij}], c_{ij} = \frac{\partial^2 E_P}{\partial q_i . \partial q_j}, \mathbf{C} = \begin{bmatrix} c_1 & -c_1 & 0 & 0 \\ -c_1 & c_1 + c_{23}.r_2^2 + c_{32}.r_2^2 & -c_{23}.r_2.r_3 - c_{32}.r_2.r_3 & 0 \\ 0 & -c_{23}.r_2.r_3 - c_{32}.r_2.r_3 & c_3 + c_{23}.r_3^2 + c_{32}.r_3^2 & -c_3 \\ 0 & 0 & -c_3 & c_3 \end{bmatrix}, \qquad \textbf{(4)}$$

where E_K и E_P are respectively the kinetic energy and the potential energy of the mechanical system, F is the dissipative function.

The general solutions of the system of differential equations (1) in the harmonious appearance of disturbing forces and initial conditions $t = 0$, $q(0) = q_0$, $\dot{q}(0) = \dot{q}_0$, written in matrix form, are

$$q(t) = \sum_{r=1}^{4} \frac{2}{g_r^2 + h_r^2}[\mathbf{G}_r\mathbf{M}\dot{q}(0) + (-\alpha_r\mathbf{G}_r\mathbf{M} + \beta_r\mathbf{H}_r\mathbf{M} \quad + \mathbf{G}_r\mathbf{B})q(0)]\,.e^{-\alpha_r t}.\cos\beta_r t +$$
$$+ \sum_{r=1}^{4} \frac{2}{g_r^2 + h_r^2}[\mathbf{H}_r.\mathbf{M}.\dot{q}(0) + (-\alpha_r.\mathbf{H}_r.\mathbf{M} - \beta_r.\mathbf{G}_r.\mathbf{M} + \mathbf{H}_r.\mathbf{B}).q(0)]\,.e^{-\alpha_r t}.\sin\beta_r t + \quad \mathbf{(5)}$$
$$+ \operatorname{Re}\{\sum_{k=0}^{n}\sum_{r=1}^{4} \frac{2}{g_r^2 + h_r^2}\,\frac{\alpha_r.\mathbf{G}_r + \beta_r.\mathbf{H}_r + i.k.\Omega.\mathbf{G}_r}{\omega_r^2 - k^2.\Omega^2 + i.2.k.\sigma_r.\omega_r.\Omega}\mathbf{Q}.e^{ik\Omega t}\}$$

where:

$$g_r = -2\alpha_r\left(\mathbf{V}_r^T.\mathbf{M}.\mathbf{V}_r - \mathbf{W}_r^T.\mathbf{M}.\mathbf{W}_r\right) - 4\beta_r\mathbf{V}_r^T.\mathbf{M}.\mathbf{W}_r + \mathbf{V}_r^T.\mathbf{B}.\mathbf{V}_r - \mathbf{W}_r^T.\mathbf{B}.\mathbf{W}_r;$$
$$h_r = 2\beta_r\left(\mathbf{V}_r^T.\mathbf{M}.\mathbf{V}_r - \mathbf{W}_r^T.\mathbf{M}.\mathbf{W}_r\right) - 4\alpha_r\mathbf{V}_r^T.\mathbf{M}.\mathbf{W}_r + 2\mathbf{V}_r^T.\mathbf{B}.\mathbf{W}_r;$$
$$\mathbf{G}_r = g_r\mathbf{L}_r + h_r\mathbf{R}_r; \quad \mathbf{L}_r = \mathbf{V}_r.\mathbf{V}_r^T - \mathbf{W}_r.\mathbf{W}_r^T;$$
$$\mathbf{H}_r = h_r\mathbf{L}_r - g_r\mathbf{R}_r; \quad \mathbf{R}_r = \mathbf{V}_r.\mathbf{W}_r^T + \mathbf{W}_r.\mathbf{V}_r^T.$$

V is the modal matrix; W – the matrix of the imaginary part of the natural vectors of the damping system; $p_r = -\alpha_r \pm i\beta_r$ – natural values; $u_r = v_r \pm iw_r$ – natural vectors; σ_r – relative damping coefficient; α_r – damping coefficient; β_r – frequency of free damping vibration;

$\alpha_r = \sigma_r.\omega_r$; $\beta_r = \omega_r\sqrt{1-\sigma_r^2}$; w_r – the imaginary part of the natural vector caused by dampening system; v_r, ω_r – natural modes and natural frequencies of the non-damping system.

The developed mechanical – mathematical model allows conducting a number of studies of the forced torsional vibrations in the wood shaper's saw unit. The vibration behavior of the mechanism, when the belt drive wears and its elastic and damping parameters change, are investigated in this study. Work's conditions in the practice are modeled by the variable moments on the drive electric motor and the wood shaper's saw. The unavoidable deviation of the correct shape of the stator and the unbalance of the rotor lead to the occurrence of a variable torsional moment on the electric motor. This moment is modeled by adding to its constant part two components that have the type: $M_{11} \sin \omega_1 t$ and $M_{12} \sin 2\omega_1 t$ (where ω_1 is the frequency of rotation of the rotor; M_{11} and M_{12} are respectively their amplitudes). The variable moment of the wood shaper's saw is modeled as to the concentrated saw's moment M_4 is added a variable component M_{4P}. This component has the type $M_{4P} = M_P \sin 6\omega_4 t$ (where ω_4 is the frequency of rotation of the wood shaper's saw; M_P – the amplitude of this variable component).

RESULTS

The data of the wood shaper, which are necessary for the investigations, are given in the table1. Numerical investigations are conducted for two different work regimes of the mechanism by means of the modern engineering software packages. In the first case it is examined a drive with new belt whose parameters are known. In the second case it is examined a drive whose elastic and damping parameters of the belt get worse by reason of long work but not beyond the bounderies. The calculations are done by taking into account the changes of the coefficients' values – they are given in brackets in the table 1.

Table 1:Machine data

J_1 – inertia moment of the electric motor's rotor (kg·m^2)	0,0102
J_2 – inertia moment of the belt puller *2* (kg·m^2)	0,0740
J_3 – inertia moment of the belt puller *3* (kg·m^2)	0,0060
J_4 – inertia moment of the shaper saw	0,0141
c_1 – stiffness of the electric motor's shaft (Nm/rad)	19938, 6·
c_3 – stiffness of the spindle (Nm/rad)	64434,6
c_{23} – stiffness of the belt (N/m)	4,5.·10^5·(5,5.·10^5)
c_{32} – stiffness of the belt (N/m)	4,5.·10^5·(5,5.·10^5)
b_1 – damping coefficient of the electric motor's shaft (Nms/rad)	5
b_3 – damping coefficient of the spindle (Nms/rad)	5
b_{23} – damping coefficient of the belt (Ns/m)	2·(1,5)
b_{32} – damping coefficient of the belt (Ns/m)	2·(1,5)
d_1 – diameter of the electric motor's shaft (mm)	28
d_3 – diameter of the spindle (mm)	44
r_2 – radius of the belt puller *2* (mm)	95
r_3 – radius of the belt puller *3* (mm)	45
l_1 – distance between the belt puller *2* and the electric motor (mm)	240
l_3 – distance between the shaper saw and the belt puller *3* (mm)	460
M_1 – moment of the electric motor (N.m)	9,554
M_{11} – additional moment of the electric motor (N.m)	4
M_{12} – additional moment of the electric motor (N.m)	4
M_2 – moment of the belt puller *2* (N.m)	0,2
M_3 – moment of the belt puller *3* (N.m)	0,15
M_4 – moment of the shaper saw (N.m)	5,6
M_P – additional moment of the shaper saw (N.m)	2.8
ω_1 – frequency of rotation (s^{-1})	314,16
ω_4 – frequency of rotation of the shaper saw (s^{-1})	628,32

The conducted investigations show that the wearing of the belt drive and the changes of its elastic and damping parameters have influence on the form of the torsional vibrations of the wood shaper. The evaluation of this influence on the electric motor and the wood shaper's saw is important. The graphs, which illustrate the free damped torsional vibrations of the electric motor's rotor, when the first and the second belt work, are shown in fig. 4 and fig. 5 respectively. Fig. 6 and fig. 7 illustrate the forced vibrations of these elements, when the first and the second belt work.

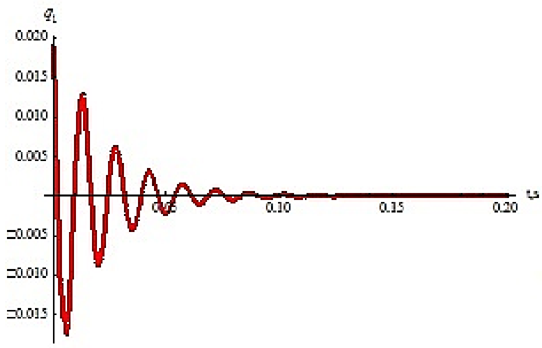

Figure 4: Free damped torsional vibrations of the electric motor's rotor whit first belt

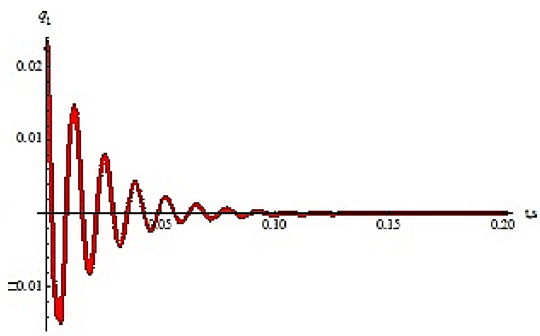

Figure 5: Free damped torsional vibrations of the electric motor's rotor whit second belt

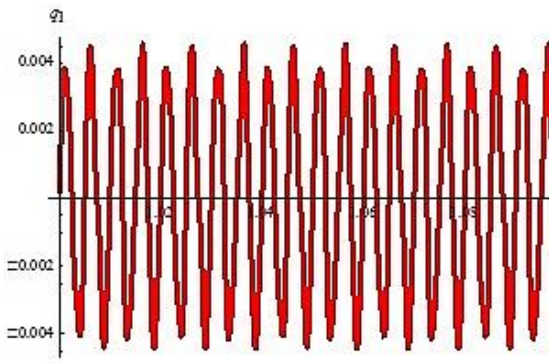

Figure 6: Forced vibrations of the electric motor's electric motor's rotor whit first belt

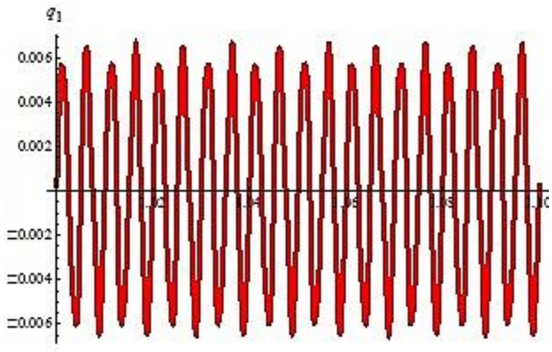

Figure 7: Forced vibrations of the electric motor's rotor with second belt

The graphs, which illustrate the free damped torsional vibrations of the wood shaper's saw, when the first and the second belt work, are shown in fig. 8 and respectively in fig. 9 y. fig. 10 and fig. 11 illustrate the forced vibrations of these elements when the first and the second belt work.

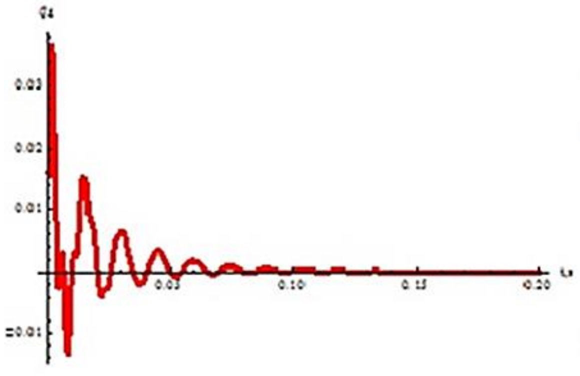

Figure 8: Free damped torsional vibrations of the wood shaper's saw whit first belt

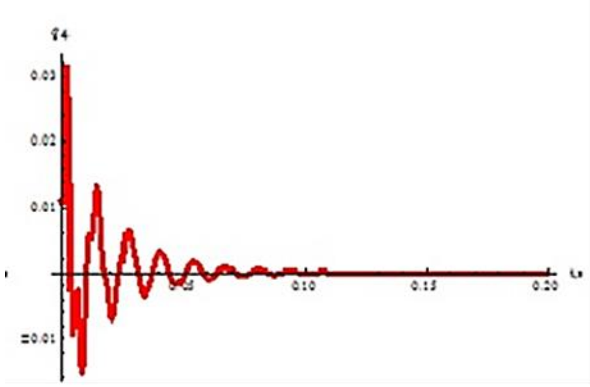

Figure 9: Free damped torsional vibrations of the wood shaper's saw whit second belt

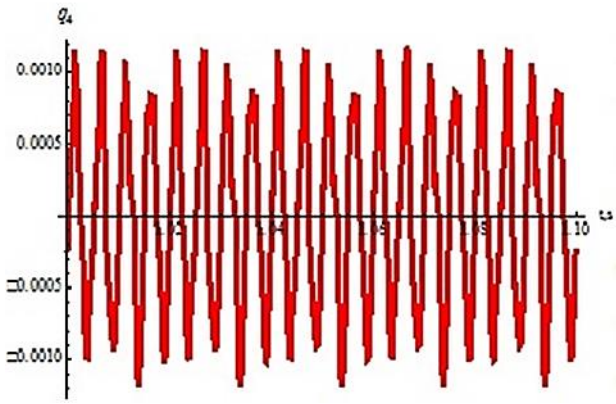

Figure 10: Forced vibrations of the wood shaper's saw whit first belt

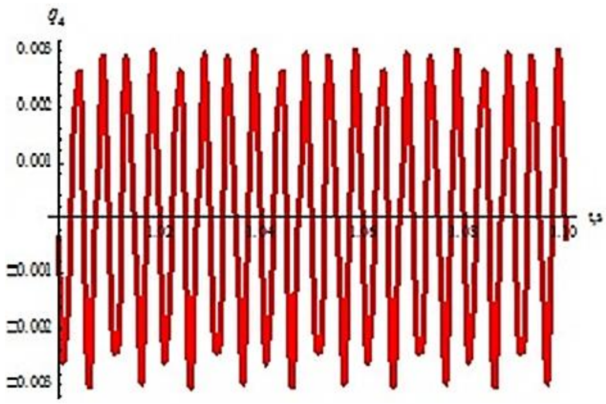

Figure 11: Forced vibrations of the wood shaper's saw with second belt

The analysis of the obtained results leads to some deductions. The damping of the torsional vibrations slows down when the belt drive wears and its elastic and damping properties change. The maximum amplitudes of the forced torsional vibrations of the electric motor's rotor and of the wood shaper's saw increase. This fact practically leads to the disturbing of the uniformly work of the electric motor and its faster amortization. Moreover, bigger amplitudes of the wood shaper's saw vibrations change to worsen the accuracy and quality of the machine's production.

The confirmation of the conclusions from the numerical investigations requires carrying out investigations on a real machine. A universal wood shaper with lower spindle model "FD – 3" is used for that purpose. This machine is made in ZDM – Plovdiv. The general view of the machine assembled with universal roll feeding mechanism model "KAM – LIK" is shown in fig. 1. An examination of the geometry and work precision of the machine is made before each experiment. The good conditions of the assembly correctness of the belt pullers and of the adjustment of the used belts are checked up especially carefully. The tests are made with specializing device Optibelt (Laser Pointer II and TT –3) – Germany. Specimens of black pine and beech prepared in advance are processed. A specimen is shown in fig. 12. Making specimens demands setting their location surface preliminary and then the bars' thickness is processed on a thickness machine.

Figure 12: A specimen

It is necessaty for the wood shaper to work at least 30 min before starting the examinations. In this way reaching the normal work temperature of all machine's elements is guaranteed. The separated investigations are carried out with preliminary determined and fixed feeding speeds and frequencies of rotation of the shaper's spindle. The thickness of the removal layer is 12 mm. After the milling the specimens are prepared for investigations of their surface characteristics (fig. 13). Firstly the roughness of processed surface of the specimens is measured. An electronic profilometer *283 P69* is used to investigate the surface roughness. It is shown in fig. 14.

Figure 13: Specimens prepared for investigations

Figure 14: Profilometer *283 P69*

After that surface characteristics of the specimens are photographed, investigated and analyzed, the electronic microscope ZEISS AxioCam ERc5s is used. The investigation of one of the series specimens with this microscope is shown in fig. 15. The photos with different enlargement of the processing surfaces when the machine has worked with the first and the second belt are shown in fig. 16 and fig. 17. The experimental results are processed by variation statistics. The received data are processed over again with the software product for analysis QstatLab. The analysis of the obtained results shows the elevated roughness of the surface when the belt, whose elastic and damping properties get worse (by 30%), is used. Therefore, the accuracy and quality of the wood shapers' production decrease in this case. The recommendation for watching closely the drive belt's work is imposed. The state of the belt's parameters is especially important because it directly affects the accuracy and quality of the wood shapers' production.

Figure 15: Microscope ZEISS

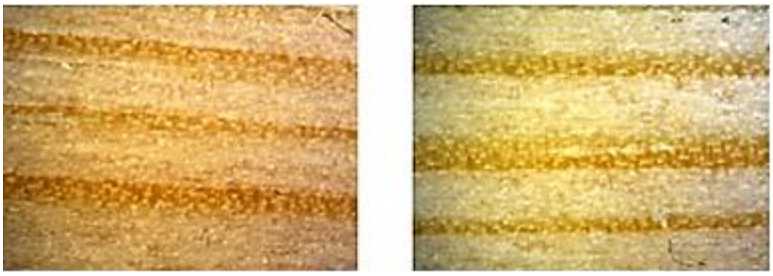

Figure 16: Processing surfaces

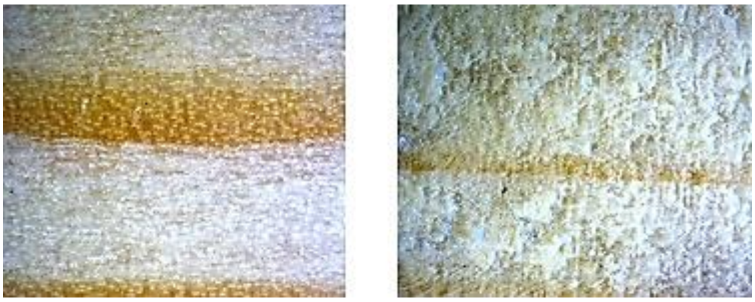

Figure 17: Processing surfaces

CONCLUSION

This study presents investigations that the influence of the wearing and the belt drive parameters' changes over the forced torsional vibrations in the saw unit of a wood shaper. The research is done on the base of a concrete mechanic-mathematical model. The main features in the construction of the wood shapers with lower spindle are rendered an account of this model. Work's regimes and conditions in the practice are modeled by the variable moments on the drive electric motor and the wood shaper's saw. The conclusions based on the numerical investigations are confirmed by the real investigations made on the real machine. The obtained and confirmed results of the whole investigation are applicable to the concrete well-founded recommendations concerning the operation of these machines. The recommendations concern the control of the drive's work and the state of the belt's parameters in exploitation of these machines. These recommendations are important for increase of the accuracy and quality of the wood shapers' production. In addition, the results of this research are useful for technical diagnostics of the wood shapers.

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