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DEFORMATION CHECKS IN BASIC LINKS OF BIG CIRCULAR SAW MACHINES

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ABSTRACT: In this article, deformation checks in basic links of big circular saw machines have been made. For this purpose, the maximum deflections in the most endangered cross sections of the main shaft are calculated. The biggest values are compared with the admissible values for the respective material. The normal operation of the big circular saw machine is guaranteed if the actual values of the deflections are equal or less than the admissible values. In this work, solutions for all cross sections of the main shaft are proposed. These solutions make it possible to carry out deformation checks when designing new circular saw machines.

KEY WORDS: deformation checks, deflections, spatial deformations, circular saw machine, circular saw blade, shafts, cross section, optimization procedures.

1 INTRODUCTION

Big circular saw machines form a specific class of woodworking machines. They are used for sawing different types of wooden material. In order to saw logs, timber sleepers and beams, circular saw blades with big diameters need to be used.

Large static and dynamic loads arise in operating mode. These loads cause large deformations in the main shaft as a result of which the shaft can be destroyed and the circular saw machine to stop working. In this work, to avoid this unfavorable case, an algorithm for carrying out deformation checks of the most endangered cross sections is suggested. For this reason, expressions describing the full deflections of the main shaft can be obtained. These expressions render an account the static and dynamic deformations of the shaft in two mutually perpendicular planes. Dependencies for their calculation were obtained by the author in previous article [1]. The next step is to calculate the actual values of the deflections in the most endangered cross sections. To perform a deformation check of the shaft, the maximum value of the deflection must be determined. Last step is to perform the actual deformation check. In this

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case, the absolute or relative deflections in the most endangered cross sections of the main shaft are used. These values of the deflections must be compared with the admissible absolute or relative deflections. Normal operation of the driving mechanism is ensured if the real deflections are smaller or equal to the limit values established in the technical literature [2–4]. The proposed theory can be used to perform deformation checks in basic links of other classes of woodworking machines to ensure their reliable and safe work in the operating mode.

2 Expose

2.1 Big circular saw machine – principle scheme

Figure 1 shows the scheme of big circular saw machine [5,6]. The following symbols are used: E – electric motor, 1 – circular saw blade, 2 and 3 – belt pulleys, 4 – circular saw shaft, 5 – workpiece, and 6 – timber trolley.

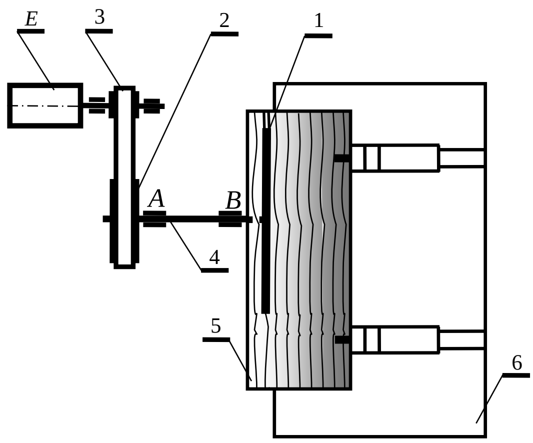

Fig. 1. Big circular saw machine.

2.2 Dynamic model

Figure 2 shows the dynamic model. We choose the following coordinate systems [1,7,8]: O_1xyz – fixed coordinate system; $O_1x_1y_1z_1$ – moving coordinate system. In the initial moment, the axes of the two coordinate systems coincide in between. $C_1x'y'z'$ – coordinate system, beginning at the centre of mass. Its axes are parallel to the axes of the moving coordinate system. $C_1\xi\eta\zeta$ – coordinate system whose axes are principal axes of inertia of the disk.

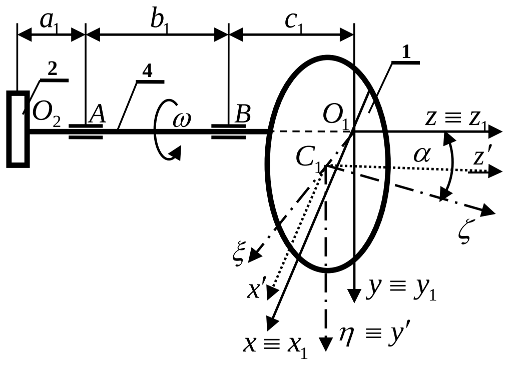

Fig. 2. Dynamic model.

2.3 Deformation checks – analytical solution

Deformation checks can be performed analytically and numerically by using the following algorithm:

- Expressions describing the static and dynamic deformations of the shaft in two mutually perpendicular planes are used. These expressions have been obtained by the author in previous articles;
- The full deformations for the relevant axes are determined;
- The full deflection of the shaft for each part is calculated from the known dependencies;
- The endangered cross-section of the shaft is determined. This is the section where the full deflection is the biggest. We denote this deflection with $f_{\max}$;
- The calculated value of $f_{\max}$ is used for deformation check of the shaft [2–4].

This algorithm is applied to each part.

Part O_2A $\quad 0 \leq z \leq a_1$

We record the expressions for determining the static and dynamic deformations of the shaft in two mutually perpendicular planes.

$$
\begin{aligned}
y^{\text{st}}_{O_2A}(z) &= \frac{1}{EJ}\Big(\frac{P^e_{2y}}{6}z^3 + C_1 z + D_1\Big),\\
x^{\text{st}}_{O_2A}(z) &= \frac{1}{EJ}\Big(\frac{-P^e_{2x}}{6}z^3 + E_1 z + F_1\Big),\\
y^{\text{dyn}}_{O_2A}(z,t) &= \big(\bar{R}_1\cos\omega_0 z + \bar{S}_1\sin\omega_0 z + \bar{V}_1\text{ch}\,\omega_0 z + \bar{W}_1\text{sh}\,\omega_0 z\big)\sin\omega t\,,\\
x^{\text{dyn}}_{O_2A}(z,t) &= \big(\bar{L}_1\cos\omega_0 z + \bar{M}_1\sin\omega_0 z + \bar{P}_1\text{ch}\,\omega_0 z + \bar{Q}_1\text{sh}\,\omega_0 z\big)\cos\omega t\,.
\end{aligned} \tag{1}
$$

The full deformations for the relevant axes are determined by the expressions below.

$$
\begin{aligned}
y_{O_2A\Sigma}(z,t) &= y^{\text{st}}_{O_2A}(z) + y^{\text{dyn}}_{O_2A}(z,t)\,,\\
x_{O_2A\Sigma}(z,t) &= x^{\text{st}}_{O_2A}(z) + x^{\text{dyn}}_{O_2A}(z,t)\,.
\end{aligned} \tag{2}
$$

The full deflection of the shaft is calculated by the dependence below.

$$
f_{O_2A}(z,t) = \sqrt{\left(y_{O_2A\Sigma}(z,t)\right)^2 + \left(x_{O_2A\Sigma}(z,t)\right)^2}\,. \tag{3}
$$

We transform the expression (3) and obtain the dependence for the full deflection of the main shaft in following form:

$$
\begin{aligned}
f_{O_2A}(z,t) = \Big\{&\text{var}\,O_2A + 2(\bar{R}_1\cos\omega_0 z + \bar{S}_1\sin\omega_0 z + \bar{V}_1\text{ch}\,\omega_0 z + \bar{W}_1\text{sh}\,\omega_0 z)\\
&\times\left[y^{\text{st}}_{O_2A}(z)\sin\omega t + x^{\text{st}}_{O_2A}(z)\cos\omega t\right]\Big\}^{1/2},
\end{aligned} \tag{4}
$$

where

$$
\begin{aligned}
\text{var}\,O_2A = {}&(y^{\text{st}}_{O_2A}(z))^2 + (x^{\text{st}}_{O_2A}(z))^2\\
&+ \big(\bar{R}_1\cos\omega_0 z + \bar{S}_1\sin\omega_0 z + \bar{V}_1\text{ch}\,\omega_0 z + \bar{W}_1\text{sh}\,\omega_0 z\big)^2\,.
\end{aligned}
$$

PART AB $\quad a_1 \le z \le a_1 + b_1$

We apply the above described algorithm:

— static and dynamic deformations of the shaft in two mutually perpendicular planes

$$
\begin{aligned}
y^{\text{st}}_{AB}(z) &= \frac{1}{EJ}\left(\frac{\left(P^e_{2y} + A^{\text{st}}_y\right)}{6}z^3 - \frac{A^{\text{st}}_y a_1}{2}z^2 + C_2 z + D_2\right),\\
x^{\text{st}}_{AB}(z) &= \frac{1}{EJ}\left(\frac{\left(A^{st}_x - P^e_{2x}\right)}{6}z^3 - \frac{A^{\text{st}}_x a_1}{2}z^2 + E_2 z + F_2\right),\\
y^{\text{dyn}}_{AB}(z,t) &= \big(\bar{R}_2\cos\omega_0 z + \bar{S}_2\sin\omega_0 z + \bar{V}_2\text{ch}\,\omega_0 z + \bar{W}_2\text{sh}\,\omega_0 z\big)\sin\omega t\,,\\
x^{\text{dyn}}_{AB}(z,t) &= \big(\bar{L}_2\cos\omega_0 z + \bar{M}_2\sin\omega_0 z + \bar{P}_2\text{ch}\,\omega_0 z + \bar{Q}_2\text{sh}\,\omega_0 z\big)\cos\omega t\,.
\end{aligned} \tag{5}
$$

— full deformations for the relevant axes;

$$y_{AB\Sigma}(z,t) = y^{\text{st}}_{AB}(z) + y^{\text{dyn}}_{AB}(z,t)\,,$$
$$x_{AB\Sigma}(z,t) = x^{\text{st}}_{AB}(z) + x^{\text{dyn}}_{AB}(z,t)\,. \tag{6}$$

— full deflection of the main shaft;

$$\begin{aligned} f_{AB}(z,t) &= \sqrt{(y_{AB\Sigma}(z,t))^2 + (x_{AB\Sigma}(z,t))^2} \\ &= \Big[\text{var}\,AB + 2(\bar{R}_2\cos\omega_0 z + \bar{S}_2\sin\omega_0 z + \bar{V}_2\text{ch}\,\omega_0 z + \bar{W}_2\text{sh}\,\omega_0 z) \\ &\qquad \times [y^{\text{st}}_{AB}(z)\sin\omega t + x^{\text{st}}_{AB}(z)\cos\omega t]\Big]^{1/2}, \end{aligned} \tag{7}$$

where

$$\begin{aligned} \text{var}\,AB = {} & (y^{st}_{AB}(z))^2 + (x^{st}_{AB}(z))^2 \\ & + (\bar{R}_2\cos\omega_0 z + \bar{S}_2\sin\omega_0 z + \bar{V}_2\text{ch}\,\omega_0 z + \bar{W}_2\text{sh}\,\omega_0 z)^2. \end{aligned}$$

PART BO_1 $\quad a_1 + b_1 \le z \le a_1 + b_1 + c_1$

— static and dynamic deformations of the shaft in two mutually perpendicular planes;

$$\begin{aligned} y^{\text{st}}_{BO_1}(z) &= \frac{1}{EJ}\Big(\frac{P^e_{2y}+A^{\text{st}}_y+B^{\text{st}}_y}{6}z^3 - \frac{A^{\text{st}}_y a_1}{2}z^2 - \frac{B^{\text{st}}_y(a_1+b_1)}{2}z^2 + C_3 z + D_3\Big), \\ x^{\text{st}}_{BO_1}(z) &= \frac{1}{EJ}\Big(\frac{A^{\text{st}}_x+B^{\text{st}}_x-P^e_{2x}}{6}z^3 - \frac{A^{\text{st}}_x a_1}{2}z^2 - \frac{B^{\text{st}}_x(a_1+b_1)}{2}z^2 + E_3 z + F_3\Big), \\ y^{\text{dyn}}_{BO_1}(z,t) &= (\bar{R}_3\cos\omega_0 z + \bar{S}_3\sin\omega_0 z + \bar{V}_3\text{ch}\,\omega_0 z + \bar{W}_3\text{sh}\,\omega_0 z)\sin\omega t\,, \\ x^{\text{dyn}}_{BO_1}(z,t) &= (\bar{L}_3\cos\omega_0 z + \bar{M}_3\sin\omega_0 z + \bar{P}_3\text{ch}\,\omega_0 z + \bar{Q}_3\text{sh}\,\omega_0 z)\cos\omega t\,. \end{aligned} \tag{8}$$

— full deformations for the relevant axes;

$$y_{BO_1\Sigma}(z,t) = y^{\text{st}}_{BO_1}(z) + y^{\text{dyn}}_{BO_1}(z,t)\,,$$
$$x_{BO_1\Sigma}(z,t) = x^{\text{st}}_{BO_1}(z) + x^{\text{dyn}}_{BO_1}(z,t)\,. \tag{9}$$

— full deflection of the main shaft;

$$\begin{aligned} f_{BO_1}(z,t) &= \sqrt{(y_{BO_1\Sigma}(z,t))^2 + (x_{BO_1\Sigma}(z,t))^2} \\ &= \Big[\text{var}\,BO_1 + 2(\bar{R}_3\cos\omega_0 z + \bar{S}_3\sin\omega_0 z + \bar{V}_3\text{ch}\,\omega_0 z + \bar{W}_3\text{sh}\,\omega_0 z) \\ &\qquad \times [y^{\text{st}}_{BO_1}(z)\sin\omega t + x^{\text{st}}_{BO_1}(z)\cos\omega t]\Big]^{1/2}, \end{aligned} \tag{10}$$

where

$$\text{var}\, BO_1 = (y^{st}_{BO_1}(z))^2 + (x^{st}_{BO_1}(z))^2 + (\bar{R}_3 \cos \omega_0 z + \bar{S}_3 \sin \omega_0 z + \bar{V}_3 \text{ch}\, \omega_0 z + \bar{W}_3 \text{sh}\, \omega_0 z)^2 .$$

The following symbols are used in the above expressions: C_j, D_j, E_j, and F_j $(j = 1, 2, 3)$, and $\bar{R}_i$, $\bar{S}_i$, $\bar{V}_i$, $\bar{W}_i$, $\bar{L}_i$, $\bar{M}_i$, $\bar{P}_i$, and $\bar{Q}_i$, $(i = 1,\ 2,\ 3)$ are constants of integration. A^{st}_x, A^{st}_y, B^{st}_x and B^{st}_y are static reactions. The expressions for their calculation were obtained by the author in a previous article in which the spatial deformations and transverse vibrations in the links of big circular saw machines are investigated. This article can be found at the following link: http://rspublication.com/ijst/ARCHIVE.html [1]. In the technical literature, there are other studies in this direction [9–13]. E is the modulus of elasticity, $J = J_x = J_y$ is the axial moment of inertia. The main shaft performs rotation with constant angular velocity ω about the axis of rotation and describes an angle $\phi = \omega t$. The fixed parameter is marked with ω_0. It can be calculated by an expression recorded in [1, 14].

2.4 Deformation checks – numerical solution

The theoretical expressions obtained above are used for numerical solution of the assigned problems. The following initial data are used [5, 6, 15, 16]:

$a_1 = 0.3$ m,	$b_1 = 0.8$ m,	$c_1 = 0.3$ m,	$e = 0.0005$ m,
$\alpha = 0.012$ rad,	$r_d = 0.75$ m,	$r_{w2} = 0.175$ m,	$H = 0.575$ m,
$D = 1.5$ m,	$E = 2.06$ e+011 Pa,	$\omega = 80$ s^{-1},	$m = 0.75$,
$\omega_0 = 0.9446$ m^{-1},	$\beta = 0.52$ rad,	$m_d = 95$ kg,	
$A_{ms} = 38.46$e-004 m^2,	$d = 0.07$ m,	$\rho = 7850$ kg/m^3,	
$K_\Delta = 92.8 \times 10^6$ J/m^3,	$\Theta_a = 0.9774$ rad,	$P^e_1 = 2540$ N,	$P^e_{1x} = 1420$ N,
$P^e_{1y} = 2100$ N,	$R^e_1 = 1900$ N,	$R^e_{1x} = 1580$ N,	$R^e_{1y} = 1065$ N,
$P^e_2 = 10885$ N,	$P^e_{2x} = 9426$ N,	$P^e_{2y} = 5442$ N,	$G^e_1 = 932$ N,

$$J = J_x = J_y = 117.8\text{e-}008 \text{ m}^4.$$

The type of the spatial deformations of the whole shaft can be obtained by concatenating the expressions describing the full deflections of the shaft for all parts, i.e. by using expressions (4), (7) and (10). Figure 3 shows the type of these deformations. Obviously, they are a function of two arguments, the coordinate z and the time t.

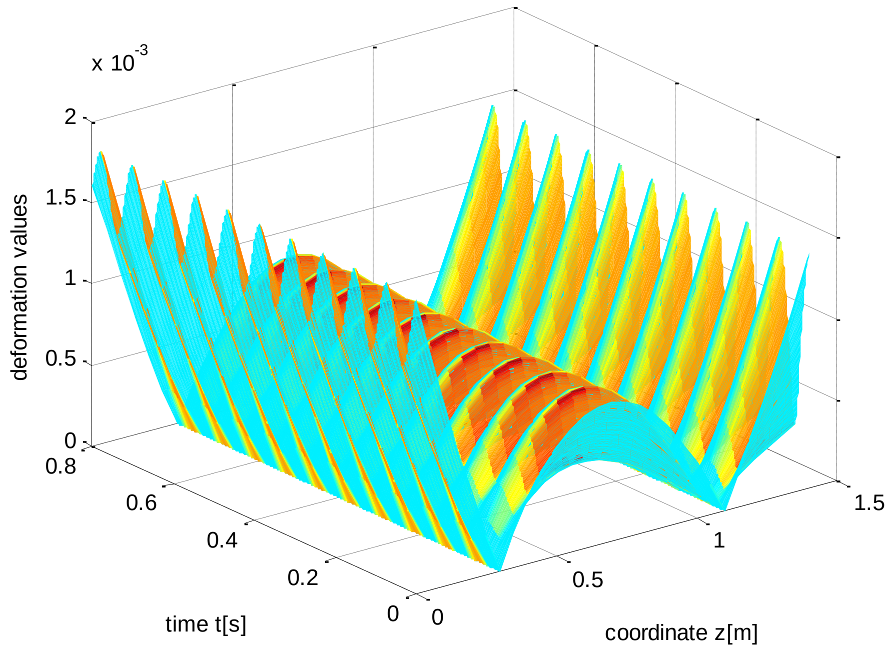

Fig. 3. (Color online) Spatial deformations of the main shaft.

PART $O_2A \quad 0 \le z \le a_1$

We denote $f_{O_2A}(0,\, t) = {}_{\max}f_{O_2A}$. In this case, the coordinate $z = 0$. That's why, the expression (4) can be written as follows:

$$
\begin{aligned}
(11) \quad f_{O_2A}(0,t) &= {}_{\max}f_{O_2A} \\
&= \sqrt{\frac{D_1^2+F_1^2}{(EJ)^2}+(\bar{R}_1+\bar{V}_1)^2+2(\bar{R}_1+\bar{V}_1)\frac{D_1\sin\omega t+F_1\cos\omega t}{EJ}}\,.
\end{aligned}
$$

The maximum value of the upper expression must be determined. This can be performed in different ways, which are listed below:

a) calculation the function values in the domain of local maximum;

b) use of optimization procedures for finding the maximum of the function [17]];

c) determination the values of the argument (the time t), for which the function possesses a maximum value.

Here we will look at the first two ways.

a) First way: Calculation the function maximum values.

In this case, we can proceed in the following way:

Time domain where the function $f_{O_2A}(0,t)$ has a local maximum must be determined. For this purpose, the diagram of the function is shown in Fig. 4. It can be established from this figure that the first local maximum appears in the interval of $0.4 \leq t \leq 0.5$ s.

The function values are calculated in the defined interval. The resulting data are represented as a matrix.

$$f_{O_2A} = \begin{bmatrix} \text{Columns 1 through 5} & & & & \\ 0.00182065 & 0.00176827 & 0.00169176 & 0.00160077 & 0.00150840 \\ \text{Columns 6 through 10} & & & & \\ 0.00143032 & 0.00138215 & 0.00137485 & 0.00141019 & 0.00147988 \\ \vdots & & & & \\ \text{Columns 91 through 95} & & & & \\ 0.00170241 & 0.00177634 & 0.00182519 & 0.00184345 & 0.00182914 \\ \vdots & & & & \\ \text{Columns 151 through 155} & & & & \\ 0.00145082 & 0.00153479 & 0.00162830 & 0.00171625 & 0.00178650 \\ \text{Columns 156 through 160} & & & & \\ 0.00183050 & 0.00184332 & 0.00182359 & 0.00177342 & 0.00169854 \end{bmatrix}$$

The maximum deflection of the shaft is selected from the upper matrix. In this case, this value is $_{\max}f_{O_2A} = 0.00184345$ m. Precise calculations show that the exact value of the maximum deflection is $_{\max}f_{O_2A} = 0.00184345663791$ m. It is not necessary for practical calculations. This value is used to perform the deformation check of the main shaft.

b) Second way: Use of optimization procedures for finding the maximum of the function.

We use the optimization procedure *fminbnd*. This procedure calculates minimum of a function, which depends on one argument. That's why, the expression (11) must be examined with a negative sign. The obtained value for $_{\max}f_{O_2A}$ must be chosen with a positive sign. It is necessary to determine the boundaries in which the time changes, i.e. $t_1 = 0.4$, $t_2 = 0.5$ s.

```
function F=def(x)
[x,fval,exitflag,output]= fminbnd(@def1,0.4,0.5,optimset('TolX',1.e-12))
...
function F1=def1(x)
...
```

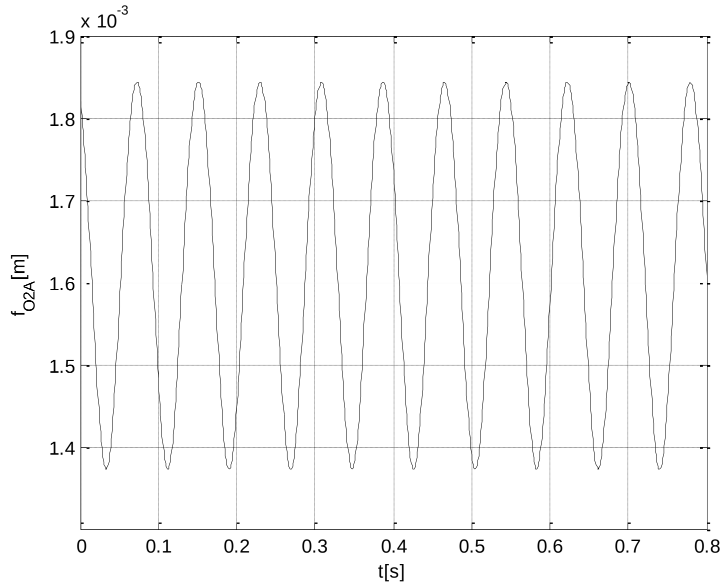

Fig. 4. Diagram of the function $f_{O_2A}(0, t)$.

The following final results are obtained after startup of the procedure *fminbnd*:

$$t = 0.46530516562898 \quad {}_{\max}f_{O_2A} = 0.00184351786154$$

exitflag =1
output =iterations: 8
funcCount: 11
algorithm: 'golden section search, parabolic interpolation'
message: [1x112 char]

The argument "output" contains records of the results from the optimization. The most important record shows the algorithm for finding the maximum of the function.

It is obvious that the calculated result for ${}_{\max}f_{O_2A}$ is the same as this one in the first way to find the maximum of the function. The most endangered cross section may be located in other parts of the main shaft. This depends on various factors such as mass and kinematic characteristics of the mechanical system (masses, mass moments of inertia, linear and angular velocities), other constructive solutions (changing the places of the disks 2, 3 and 5, of the bearing supports A and B), various static and dynamic loads, etc. In this case, we have to do deformation check for these parts.

Part BO_1 $a_1 + b_1 \leq z \leq a_1 + b_1 + c_1$

The supports A and B are perfectly rigid, i.e. they are not deformed. That is why, the biggest deformations are established at the ends of the shaft [$z = 0$ and $z = l$ ($l = a_1 + b_1 + c_1$)]. We use the above algorithm for deformation check of the main shaft and get the following relations:

The expression (10) is transformed and the following dependence for $f_{BO_1}(l,t)$ is obtained.

$$f_{BO_1}(l,t) = \Big[\text{const}BO_1 + 2(\bar{R}_3 \cos\omega_0 l + \bar{S}_3 \sin\omega_0 l + \bar{V}_3 \text{ch}\,\omega_0 l + \bar{W}_3 \text{sh}\,\omega_0 l) \times [y^{\text{st}}_{BO_1}(l)\sin\omega t + x^{\text{st}}_{BO_1}(l)\cos\omega t]\Big]^{1/2}, \tag{12}$$

where

$$\text{const}BO_1 = (y^{st}_{BO_1}(l))^2 + (x^{st}_{BO_1}(l))^2 + (\bar{R}_3 \cos\omega_0 l + \bar{S}_3 \sin\omega_0 l + \bar{V}_3 \text{ch}\,\omega_0 l + \bar{W}_3 \text{sh}\,\omega_0 l)^2,$$

$$y^{\text{st}}_{BO_1}(l) = \frac{1}{EJ}\Big(\frac{P^e_{2y} + A^{\text{st}}_y + B^{\text{st}}_y}{6} l^3 - \frac{A^{\text{st}}_y a_1}{2} l^2 - \frac{B^{\text{st}}_y (a_1 + b_1)}{2} l^2 + C_3 l + D_3\Big),$$

$$x^{\text{st}}_{BO_1}(l) = \frac{1}{EJ}\Big(\frac{A^{\text{st}}_x + B^{\text{st}}_x - P^e_{2x}}{6} l^3 - \frac{A^{\text{st}}_x a_1}{2} l^2 - \frac{B^{\text{st}}_x (a_1 + b_1)}{2} l^2 + E_3 l + F_3\Big).$$

a) First way: Calculation the function maximum values.

Figure 5 shows the diagram of the function $f_{BO_1}(l,t)$.

The first local maximum appears in the interval of $0 \leq t \leq 0.1$ s. We calculate the function values in this interval. The data are presented as a matrix.

$$f_{BO_1} = \begin{bmatrix} \text{Columns 1 through 5} & & & & \\ 0.00145498 & 0.00138887 & 0.00127144 & 0.00110862 & 0.00090972 \\ \text{Columns 6 through 10} & & & & \\ 0.00068965 & 0.00047818 & 0.00035457 & 0.00042784 & 0.00062577 \\ \vdots & & & & \\ \text{Columns 16 through 20} & & & & \\ 0.00146890 & 0.00144122 & 0.00135967 & 0.00122824 & 0.00105374 \\ \vdots & & & & \\ \text{Columns 151 through 155} & & & & \\ 0.00054593 & 0.00076556 & 0.00098067 & 0.00116861 & 0.00131684 \\ \text{Columns 156 through 160} & & & & \\ 0.00141734 & 0.00146512 & 0.00145792 & 0.00139608 & 0.00128255 \end{bmatrix}$$

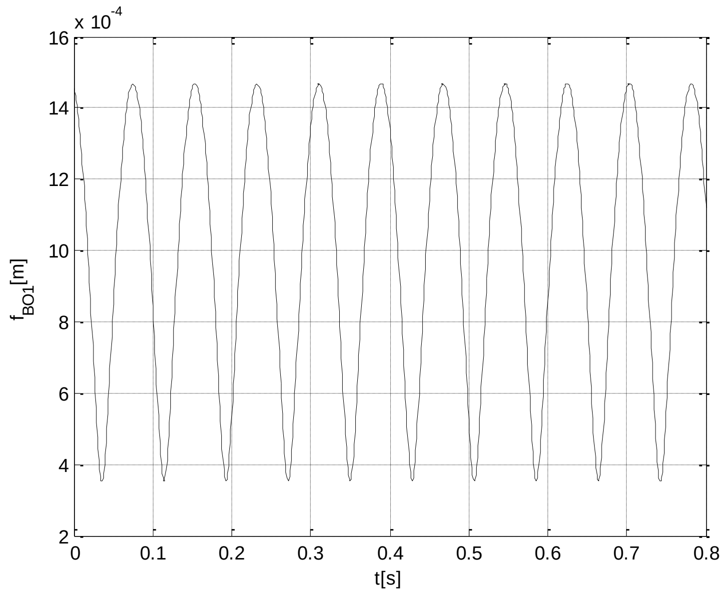

Fig. 5. Diagram of the function $f_{BO_1}(l,t)$.

In this case, the maximum deflection of the shaft is equal to ${}_{\max}f_{BO_1} = 0.00146890$ m. Precise calculations show the exact value of the maximum deflection: ${}_{\max}f_{BO_1} = 0.00146890885430$ m. This value should be used to perform a deformation check of the shaft, if it is the largest measured value.

b) Second way: Use of optimization procedures of finding the maximum of the function.

In this case, we also use the optimization procedure *fminbnd* described above. The expression (12) must be examined with a negative sign. The value obtained for ${}_{\max}f_{BO_1}$ must be chosen with a positive sign. It is necessary to determine the boundaries, in which the time ranges, i.e. $t_1 = 0$, $t_2 = 0.1$ s. The function possesses a maximum value in these boundaries.

```
function F=def(x)
[x,fval,exitflag,output]=fminbnd(@def1,0,0.1,optimset('TolX',1.e-12))
...
function F1=def1(x)
...
```

The following final results are obtained after startup of the procedure *fminbnd*:

$$t = 0.07499027195342 \quad {}_{\max}f_{BO_1} = 0.00146890895912$$

exitflag =1
output = iterations: 9
funcCount: 12
algorithm: 'golden section search, parabolic interpolation'
message: [1x112 char]

The maximum values of the deflection are equal to each other.

PART AB $a_1 \leq z \leq a_1 + b_1$

We use the dependence (7) for a part AB. Obviously, this function depends on two arguments – the longitudinal coordinate z and the time t.

a) First way: Calculation the function maximum values.

Figure 6 shows the deformation of the main shaft in a part AB. This surface is a function of the time t and the coordinate z and it is indicated as $f_{AB}(z,t)$.

This figure shows that the maximum deflections for this shaft part can be measured at the coordinate $z = l/2$. Figure 7 shows the diagram of the function $f_{AB}(l/2,t)$.

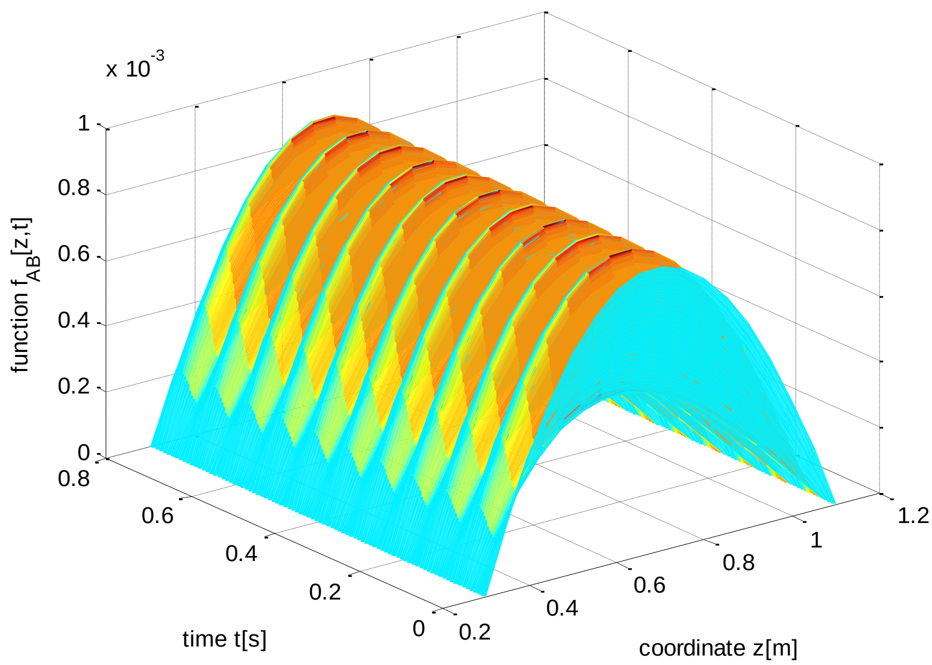

Fig. 6. (Color online) Function $f_{AB}(z,t)$.

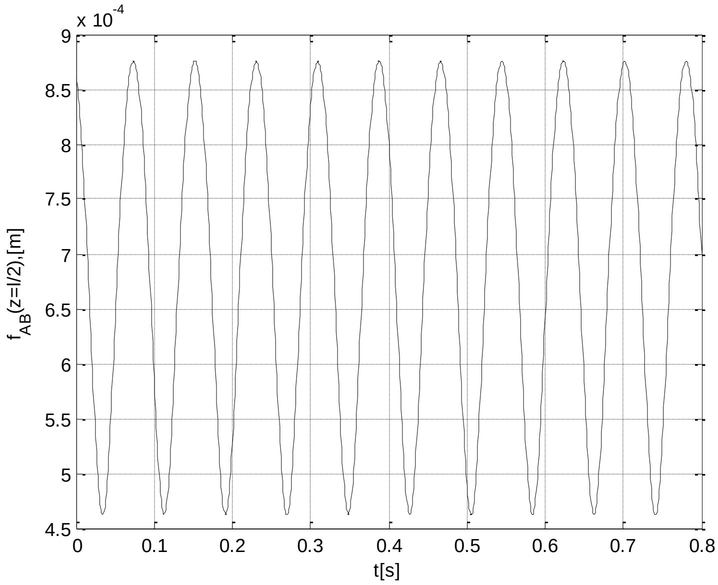

Fig. 7. Diagram of the function $f_{AB}(l/2, t)$.

The first local maximum appears in the interval of $0.5 \leq t \leq 0.6\,[s]$. We record the values of the function $f_{AB}(l/2, t)$ as a matrix.

$$f_{AB} =$$

$$= 1.0\text{e}{-003}\begin{bmatrix} \text{Columns 1 through 5} \\ 0.86261033 \quad 0.82554719 \quad 0.76728402 \quad 0.69306423 \quad 0.61112995 \\ \text{Columns 6 through 10} \\ 0.53387330 \quad 0.47846257 \quad 0.46265868 \quad 0.49269539 \quad 0.55721 \\ \vdots \\ \text{Columns 106 through 110} \\ 0.69888997 \quad 0.77217749 \quad 0.82903334 \quad 0.86440086 \quad 0.87548848 \\ \vdots \\ \text{Columns 151 through 155} \\ 0.53038582 \quad 0.60698600 \quad 0.68902170 \quad 0.76385891 \quad 0.82307629 \\ \text{Columns 156 through 160} \\ 0.86129953 \quad 0.87547861 \quad 0.86454869 \quad 0.82932767 \quad 0.77259410 \end{bmatrix}$$

The maximum of the function is equal to ${}_{\max}f_{AB} = 0.87548848e - 003$ m. Precise calculations show the exact value of the maximum deflection: ${}_{\max}f_{AB} =$

0.87548848367995e-003 m. This value should be used to perform a deformation check of the shaft, if it is the largest measured value.

b) Second way: Use of optimization procedures for finding the maximum of the function.

We use the optimization procedure *fmincon*. This procedure performs multidimensional optimization with constraints by finding the minimum of the function. That's why, the expression (7) for $f_{AB}(z,t)$ must be examined with a negative sign. It is necessary to determine the initial approximations of the coordinates z and t, for which the function possesses a maximum value. These initial approximations are determined by Fig. 6 and Fig. 7. We define the following values: $x_0 = [z,\, t] = [0.7\, 0.54]$. The procedure requires the interval boundaries, in which the coordinates z and t change their values to be determined. In the present case, these intervals are: lower bounds $lb = [0.3\, 0]$, upper bounds $ub = [1.1\, 0.8]$.

```
function F=def(x)
x0=[0.7 0.54];
lb=[0.3 0]; ub=[1.1 0.8];
ops=optimset('LargeScale','off');
[x,fval,exitflag,output]=fmincon(@def1,x0,[],[],[],[],lb,ub,[],ops)
...
function F1=def1(x)
...
```

The following final results are obtained after startup of the procedure *fmincon*:

$$z = 0.69977753752319 \text{ m} \quad t = 0.54470472321467 \text{ s}$$

$$_{\max}f_{AB} = 8.755784560406159\text{e-}004 \text{ m}$$

```
exitflag =5
output = iterations: 1
funcCount: 7
stepsize: 1
algorithm: 'medium-scale: SQP, Quasi-Newton, line-search'
firstorderopt: 2.069272522930987e-004
cgiterations: []
message: [1x172 char]
```

We can determine the most endangered cross sections for the main shaft where the spatial deformations possess maximum values. For this purpose, we use the values obtained for the maximum deflections, which are marked with $_{\max}f_{O_2A}$, $_{\max}f_{BO_1}$

and ${}_{\max}f_{AB}$. The greatest deflection is denoted by $f_{\max}$ and can be used to perform a deformation check of the shaft. In this case, the following expressions are used [2]:

$$f_{\max} \leq [f]\,, \quad f_{\max}/l_b \leq [f/l_b]\,, \tag{13}$$

where $[f]$ is the admissible deflection, and $[f/l_b]$ is the admissible relative deflection. The values of this deflection are taken from technical literature. We can choose the most appropriate value for each particular case, i.e. $[f/l_b] = a_b$. The admissible deflection can be calculated in the following way: $[f] = a_b l_b$. The distance l_b accepts different values for the different parts: for part $O_2Al_b = a_1$, for parts AB $l_b = b_1$ and for part BO_1, $l_b = c_1$. The deformation conditions are satisfied if the maximum deflection $f_{\max}$ or the maximum relative deflection $f_{\max}/l_b$ are smaller or equal to the respective admissible deflections. The final expressions are shown below.

$$f_{\max} \leq a_b l_b\,, \quad f_{\max}/l_b \leq a_b\,. \tag{14}$$

3 Conclusion

In this article, an algorithm for performing deformation checks in basic links of big circular saw machines is proposed. The article consists of two main parts. The first part is entitled "Deformation checks-analytical solution". In this part, theoretical expressions by which shaft deflections can be calculated in all parts (O_2A, AB and BO_1) are obtained. For this purpose, analytical expressions, which describe the deformations of the main shaft of the big circular saw machines in two mutually perpendicular planes are used.

The second part is entitled "Deformation checks - numerical solution". In this part, two main ways of determining the most endangered section are considered. Each a way possesses its advantages and disadvantages. These ways require relevant calculations to be performed, as well as diagrams and surfaces to be drawn and optimization procedures to be included. For this purpose, the computer programs need to be created. These programs allow solving various optimization problems taking into account various parameters. The purpose of these solutions is the spatial deformations of the transmissions to have minimum values.

In conclusion we can say that this study can be applied for deformation checks in the transmissions of other classes of woodworking machines. The developed theory can be used in the design process of new woodworking machines. These machines must be characterized by high reliability and safety in operating mode.

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