The Flexion Movement of the Rotors Afferent to the Centrifuges with Vertical Axis

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This paper presents a study model on the bending vibration of vertical centrifuges with a console mounted basket. The shaft - basket assembly is moulded as a discrete system with two degrees of freedom. The influence of the basket weight force and of its centre of mass positioning upon its natural pulsations is herein studied. The influence of the gyroscopic moment is also considered.

Keywords: Vertical centrifuge, vibrations, natural frequencies, critical speeds

The bending vibrations of the horizontal centrifuges with the console mounted basket are studied in [1]. It is studied also the influence of the constructive and functional factors over the own pulsations.

For technological reasons, in practice, vertical centrifuges with a console mounted basket are used on a large scale.

As regards the centrifuges, there are, in general, two issues:

a. the rigidity of the system, on the basis of which it is determined the angular speed of the shaft, and its comparison with the critical angular speed that results from the determination of its own pulsation (this item was solved in [2]);

b. the flexion movement determined by placing the masses on the shaft (by placing the basket).

In some papers from technical literature [3], [4] the centrifuges are studied, but from the construction of basket point of view only. The problem of the dynamic behaviour of centrifuges is not approached. In [5] the rotors being part of the process equipment are studied, but only from the shafts stress calculation point of view only.

The calculus of the rotor rigidity of vertical centrifuges implies among others the study of the bending vibrations of this system as well. The hereby document deals with the calculation of the critical angular speed (the critical speed) of the shaft. By knowing these critical speeds, it will be avoided the operation of the centrifuge at these speeds or around this kind of speed.

This paper presents a study model for the vibrations of vertical centrifuges for the three situations that might occur. Figure 1 schematically presents these three cases: the fixing point of the basket (O) is located between the centre of mass (C) and the bearing (fig.1,a); the centre of mass coincides to the fixing point of the basket (fig.1,b); the centre of mass is located between the fixing point of the basket and the bearing (fig.1,c). G is the weight of the basket together with the operating material (fluid).

The modelling of the centrifuges in order to study the bending vibrations can be made in two ways:

-discrete system with a finite number of degrees of freedom, composed of a flexible shaft with a negligible mass positioned on two bearing blocks and a basket fixed on the shaft (between the bearing blocks or console mounted);

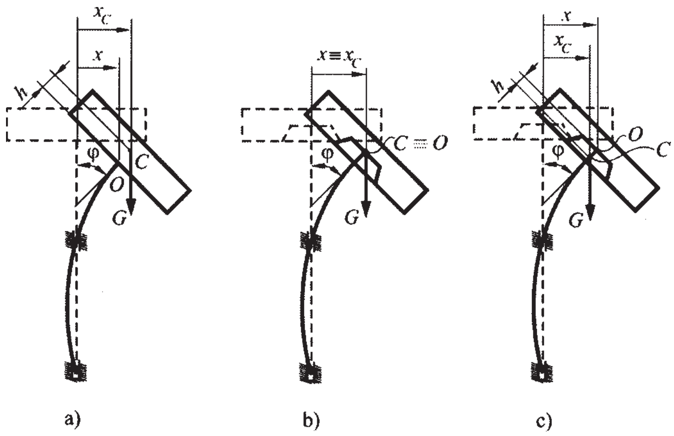

Fig. 1. The position of the fixing point of the basket with respect to the centre of mass in the case of a centrifuge with vertical axis

-flexible shaft considered to be a continuous medium (with an infinity of degrees of freedom) positioned on two bearing blocks and a basket fixed on the shaft (between the bearing blocks or console mounted).

In this paper the model with two degrees of freedom is used. Though it is a simplified model of the centrifuge, it offers the possibility of a good and quick qualitative study for the dynamic behaviour of the centrifuge.

The bending vibrations in plane xOy will be studied, just as in the case of a horizontal centrifuge [1] (fig. 2).

As opposed to the horizontal centrifuge, when studying the bending vibrations of a vertical centrifuge, one should consider the weight of the basket (plus the operating material) (fig.2,c). The weight force of the basket realizes mechanical work and contributes to the oscillatory movement. This paper studies the extent in which the weight force influences the values of the natural frequencies. The weight force of the basket is reduced in its fixing point on the shaft, O, at a force $G = mg$ and a moment $M_G = mg \cdot h \sin \varphi$ (fig.2,c). Because the angle φ takes low values, the approximation $\sin \varphi \approx \varphi$ can be made. Thus, the elements of reducing the weight force in the fixing point of the basket on the shaft are (fig. 2,d):

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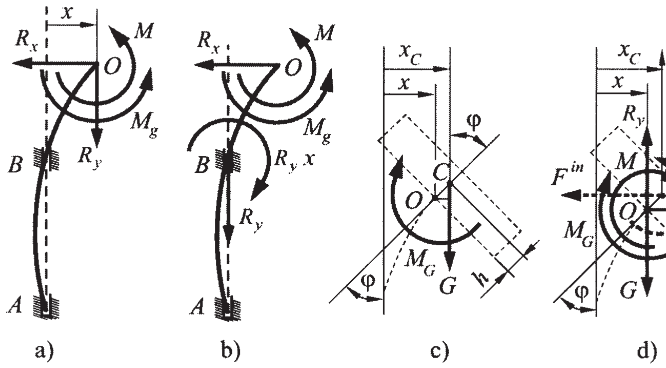

Fig. 2. The calculus of reactions in the fixing point of the basket on the shaft. The action of the gyroscopic moment

$$\tau_O^G \begin{cases} G = mg \\ M_G = mgh\varphi \end{cases} \quad (1)$$

Because there is an interest in the determination of the critical speeds, only the undamped vibrations are studied: free and forced vibrations. If the critical speeds cannot be avoided and the centrifuge is operating at almost critical speeds, the amplitudes of the bending vibrations can be reduced by installing a damper or a vibration dynamic absorber. The possibility to control the bending vibrations via these methods shall make the topic of a separate work.

The disturbance force may appear due to the fact that the centre of mass of the basket is not found on the rotation axis. This might occur as well during the operation due to uneven material deposits on the basket or due to damages occurred during the operation. Thus, just like in the case of horizontal centrifuges [1], the elements of reducing the disturbance force in point *O* are obtained:

$$\tau_O^p \begin{cases} F_p = me\Omega^2 \cos \Omega t \\ M_p = meh\Omega^2 \cos \Omega t \end{cases} \quad (2)$$

where:

m - the mass of the basket;

Ω - the angular speed of the shaft - basket system;

e - the eccentricity (OC = e).

One factor that influences the values of natural angular frequencies is the gyroscopic moment. When the basket is in console mounted the gyroscopic phenomenon takes place. The gyroscopic moment (fig.2,a) modifies the bending deformation, and thus the natural angular frequency and the critical speed as well.

Just like in the case of horizontal centrifuges with a console mounted basket, if the flexion plane (the plane in which the basket realizes bending vibrations) and the shaft rotate in the same direction with the same angular speed $\omega = \Omega$, the movement is forward synchronous rotation. The module of the gyroscopic moment is [6] $M_g = (J_y - J_z)\Omega^2\varphi$. If the flexion plane and the shaft rotate in opposite ways with the same angular speed $\omega = \Omega$, the movement is backward synchronous rotation. The module of the gyroscopic moment is [6] $M_g = (J_y + J_z)\Omega^2\varphi$.

The study of free vibrations

<u>The fixing point of the basket is found between the centre of mass and the bearing</u>

The model used for study is that of two degrees of freedom (fig. 1). The basket is embedded in the shaft. It is considered that the basket has a plane-parallel movement: The two parameters are: x_C– horizontal displacement of the basket centre of mass; φ– rotation around the axis *Oz* of the median plane of the basket.

Just like in the case of the horizontal centrifuge with a console mounted basket, in order to write the differential equations of the movement, the method of influence coefficients is used. First of all, the reactions in point *O* must be calculated (fig.2,a). For this purpose, the basket is isolated and the d'Alembert's principle is applied [7] (fig.2,d) for the calculus of the reactions R_x, R_y and M in *O*. The following expressions are obtained:

$$\begin{cases} R_x = m\ddot{x}_c \\ R_y = G \\ M = J_{cz}\ddot{\varphi} + m\ddot{x}_c h\cos\varphi - Gh\sin\varphi \end{cases} \quad (3)$$

Passing from the displacement x_C of the basket centre of mass to the displacement x of the theoretical fixing point of the basket on the shaft (fig.2,c) and taking into account the fact that the angle φ takes low values, the following expressions are obtained:

$$\begin{cases} R_x = m\ddot{x} + mh\ddot{\varphi} \\ R_y = mg \\ M = mh\ddot{x} + (mh^2 + J_{cz})\ddot{\varphi} - mgh\varphi \end{cases} \quad (4)$$

The reaction R_y has the effect of amplifying the bending strain and the angle φ. That is why the reaction R_y is reduced in point **B** at a force equal to R_y and a moment with the size of $R_y x$ (fig. 2b).

a) The case of forward precession (fig.2,d)

When the gyroscopic effect is taken into account, the gyroscopic moment M_g in chapter 1 must be considered. In the case of forward precession the gyroscopic moment M^g tends to stiffen the shaft, therefore it has the same direction with the resultant moment of the inertia forces (fig.2,a).

Using the influence coefficients method we can write:

$$\begin{cases} x = \delta_F(-R_x) + \delta_M(-M - M_g) + \delta_{MB}R_y x \\ \varphi = \alpha_F(-R_x) + \alpha_M(-M - M_g) + \alpha_{MB}R_y x \end{cases} \quad (5)$$

where δ_F and δ_M are the influence coefficients which represent the displacements in point *O* produced by a force and, respectively, a moment equal to one unit; α_F, α_M - influence coefficients which represent the rotations in point *O* produced by a force and, respectively, a moment equal to one unit; δ_{MB} and α_{MB} - influence coefficients which represent the displacements in point *O* produced by a moment equal to one unit placed in point *B*.

The expressions for the influence coefficients can be found in [8]. Replacing the given R_x, R_y, M and M_g with the above expressions and using the literal notations

$$a_1 = \delta_F + \delta_M h\,; \quad a_2 = mh^2 + J_{Cz}\,;$$
$$a_3 = (J_{Cy} - J_{Cz})\Omega^2 - mgh\,; \quad a_4 = \alpha_F + \alpha_M h$$

we obtain:

$$\begin{cases} x = -a_1 m\ddot{x} - [\delta_F mh + \delta_M a_2]\ddot{\varphi} - \delta_M a_3\varphi + \delta_{MB} mgx \\ \varphi = -a_4 m\ddot{x} - [\alpha_F mh + \alpha_M a_2]\ddot{\varphi} - \alpha_M a_3\varphi + \alpha_{MB} mgx \end{cases} \quad (6)$$

where:

Ω - the angular speed of the shaft;

J_{Cz} - the mechanical moment of inertia of the basket with respect to the *Cz* axis which passes through the centre of mass and which is perpendicular to the drawing plane;

J_{Cy}- the mechanical moment of inertia of the basket with respect to the *Cy* axis which passes through the centre of mass and which is the theoretical rotation axis.

We search synchronous and in phase solutions of the form:

$$\begin{cases} x = X\cos(\omega t - \psi) \\ \varphi = \Phi\cos(\omega t - \psi) \end{cases} \quad (7)$$

Replacing within relations (6) we obtain the linear and homogeneous algebraic system for the unknown variables × and Φ.

$$\begin{cases} [a_1 m\omega^2 + \delta_{MB}mg - 1]X + \{[\delta_F mh + \delta_M a_2]\omega^2 - \delta_M a_3\}\Phi = 0 \\ [a_4 m\omega^2 + \alpha_{MB}mg]X + \{[\alpha_F mh + \alpha_M a_2]\omega^2 - \alpha_M a_3 - 1\}\Phi = 0 \end{cases} \quad (8)$$

In order that the system (8) should allow nonzero solutions, the determinant of the system must be zero. Following this condition we obtain the equation of natural angular frequencies in the form of:

$$B_1\omega^4 - B_2\omega^2 + B_3 = 0 \quad (9)$$

where:

$$\begin{aligned} B_1 &= a_5 m J_{cz} \\ B_2 &= [\delta_F m + a_6 mh + \alpha_M a_2 + a_5 a_3 m] - \delta_{MB} mg a_8 + \alpha_{MB} mg a_7 \\ B_3 &= \alpha_M a_3 + 1 + \alpha_{MB}\delta_M a_3 mg - \delta_{MB} mg(\alpha_M a_3 + 1), \end{aligned} \quad (10)$$

in which the following literal notations have been used

$$a_5 = \delta_F\alpha_M - \delta_M\alpha_F; \quad a_6 = \alpha_F + \delta_M$$

Solving this equation we obtain:

$$\omega_1 = \sqrt{\frac{B_2 - \sqrt{B_2^2 - 4B_1B_3}}{2B_1}} \quad (11)$$

$$\omega_2 = \sqrt{\frac{B_2 + \sqrt{B_2^2 - 4B_1B_3}}{2B_1}} \quad (12)$$

In the case ofq forced vibrations the elements of reduction of the disturbance forces system F_p and M_p intervenes relationship (2). Thus the expressions (6) become:

$$\begin{cases} x = \delta_F(-R_x + F_p) + \delta_M(-M - M_g + M_p) + \delta_{MB}R_y x \\ \varphi = \alpha_F(-R_x + F_p) + \alpha_M(-M - M_g + M_p) + \alpha_{MB}R_y x \end{cases} \quad (13)$$

Replacing R_x, R_y, M, F_p, M_p, M_g with the previously established expressions and separating the terms, the non-homogeneous system with differential equations is obtained:

$$\begin{cases} -me\Omega^2 a_1\cos\Omega t = -a_1 m\ddot{x} - a_7\ddot{\varphi} - (1 - \delta_{MB}mg)x - \delta_M a_3\Omega^2\varphi \\ -me\Omega^2 a_4\cos\Omega t = -a_4 m\ddot{x} - a_8\ddot{\varphi} + \alpha_{MB}mgx - [1 + \alpha_M a_3\Omega^2]\varphi \end{cases} \quad (14)$$

in which the following literal notation have been used:

$$a_7 = \delta_F mh + \delta_M(mh^2 + J_{Cz}); \quad a_8 = \alpha_F mh + \alpha_M(mh^2 + J_{Cz})$$

Just like in the case of horizontal centrifuges, the permanent case of movement is studied. We propose solutions in the form of:

$$\begin{cases} x = X\cos\Omega t \\ \varphi = \Phi\cos\Omega t \end{cases} \quad (15)$$

By replacing them we obtain the non-homogeneous algebraic system:

$$\begin{cases} [a_1 m\Omega^2 + \delta_{MB}mg - 1]X + \Omega^2\{a_7 - \delta_M a_3\}\Phi = -me\Omega^2 a_1 \\ (a_4 m\Omega^2 + \alpha_{MB}mg)X + \{a_8\Omega^2 - \alpha_M a_3\Omega^2 - 1\}\Phi = -me\Omega^2 a_4 \end{cases} \quad (16)$$

which, when solved, leads to the expressions of the amplitudes of the movements in the case of forced vibrations:

$$X = \frac{me\Omega^2(a_1 - a_5 mgh) + me\Omega^4 a_5(a_3 - J_{Cz})}{-m\Omega^4 a_5(a_9 - J_{cz}) + \Omega^2 a_{12} + a_{10}m^2g^2h - a_{11}mg + 1} \quad (17)$$

$$\Phi = \frac{me\Omega^2[(1 - \delta_{MB}mg)a_4 + \alpha_{MB}a_1 mg]}{-m\Omega^4 a_5(a_9 - J_{cz}) + \Omega^2 a_{12} + a_{10}m^2g^2h - a_{11}mg + 1} \quad (18)$$

in which the following literal notations have been used:

$$\begin{aligned} a_9 &= J_{Cy} - J_{Cz}; \\ a_{10} &= \alpha_M\delta_{MB} - \alpha_{MB}\delta_M; \\ a_{11} &= \delta_{MB} + \alpha_M h \\ a_{12} &= a_9(\alpha_M - a_9 mg) - ma_1 - \\ &- a_8(1 - \delta_{MB}mg) + a_5 m^2 gh - \alpha_{MB}a_7 mg \end{aligned}$$

By comparing the expressions of natural angular frequencies, (11) and (12), and the amplitudes of the movement in the case of forced vibrations, (17) and (18), with the expressions obtained in the study of the vibrations of the horizontal centrifuge, we can notice that in the case of a vertical centrifuge some terms show up that contain the weight force of the basket, *mg*, and the influence coefficients δ_{MB} and α_{MB} which are specific for the vertical centrifuge. The presence of these terms may lead to bigger or smaller differences between the results obtained in the study of vibrations for the two types of centrifuges.

In the numerical example in this paper it is studied if the vertical positioning of the centrifuge leads to significant modifications of the natural angular frequencies.

b) The case of backward precession

In the case of backward precession the gyroscopic moment M_g tends to amplify the bending strain of the shaft and thus to reduce the stiffness of the shaft. Therefore the direction of the gyroscopic moment is contrary to that of the resultant moment of the inertia forces system.

The study of vibrations in the case of backward precession is made as for the horizontal centrifuge [1], meaning that the expression of the gyroscopic moment $M_g = (J_y - J_z)\Omega^2\varphi$ is replaced with $M_g = -(J_y + J_z)\Omega^2\varphi$. This means that in all the expressions $a_3 = (J_{Cy} - J_{Cz})\Omega^2 - mgh$ is replaced with $a_3 = -(J_{Cy} - J_{Cz})\Omega^2 - mgh$ and $a_9 = J_{Cy} - J_{Cz}$ is replaced with $a_9 = -(J_{Cy} + J_{Cz})$.

The fixing point of the basket on the shaft coincides with the centre of mass

If the fixing point of the basket on the shaft coincides with the centre of mass, then $h = 0$. Thus, both in the case of forward and backward precession, the relations from which we determine the natural angular frequencies and the amplitudes of the permanent movement (in the case of forced vibrations) are obtained from the relations (11) and (12), through customization $h=0$.

Table 1

THE VALUES OF FUNDAMENTAL NATURAL FREQUENCY IN THE CASE OF HORIZONTAL AND VERTICAL CENTRIFUGES

	Horizontal centrifuge [Hz]	Vertical centrifuge [Hz]	Error [%]
Negligible gyroscopic effect	150.4840	150.4816	0.0016
Forward precession	152.3540	152.3517	0.0015
Backward precession	143.5029	143.5005	0.0017

<u>The centre of mass is located between the fixing point of the basket on the shaft and the bearing</u>

The diference comparated to the case in which the fixing point of the basket is found between the centre of mass and the bearing, is that $h < 0$. Thus, both in the case of forward and backward precession, the relations from which we determine the natural angular frequencies and the amplitudes of the permanent movement (in the case of forced vibrations) are obtained from the relations (11) and (12), by replacing h with $-h$.

Application

It is considered the case of a vertical centrifuge with a console mounted basket.and that the basket has a cylinder form. The geometrical and mechanical features of the shaft-basket assembly are identical with those of a horizontal centrifuge with a console mounted basket [1]:

-length of the shaft: 0.8 m;

-diameter of the shaft: 0.08 m;

-diameter of the basket: 0.5 m;

-length of the basket: 0.12 m;

-distance from the bearing to the fixing point of the basket: 0.05 m;

-density of the shaft and basket material: 7800 Kg/m^3;

- longitudinal modulus of elasticity (Young's modulus): $21x10^{10}$ N/m^2;

-distance from the longitudinal axis of the basket to the centre of mass (e): 0.003 m.

For the influence coefficients the following values have been obtained:

$\delta_F=1.3877x10^{-9}$ m·N^{-1}; $\delta_M=-2.5904x10^{-8}$ N^{-1}; $\alpha_F=-2.5904x10^{-8}$ N^{-1}; $\alpha_M=5.2055x10^{-7}$ m^{-1}·N^{-1}, $\delta_{MB}=3.1588x10^{-8}$ N^{-1}, $\alpha_{MB}=6.3176x10^{-7}$ m^{-1}N^{-1}.

The mechanical moments of inertia:

$J_{Cy}=1.8755$ kg·m^2; $J_{Cz}=1.0628$ kg·m^2.

It has been chosen for this example a centrifuge with geometrical and mechanical features identical with those of the centrifuge considered in the study of the vibrations of a horizontal centrifuge, shown in [1], in order to investigate to what extent the weight force of the basket influences the values of the natural angular frequencies of the vertical centrifuge.

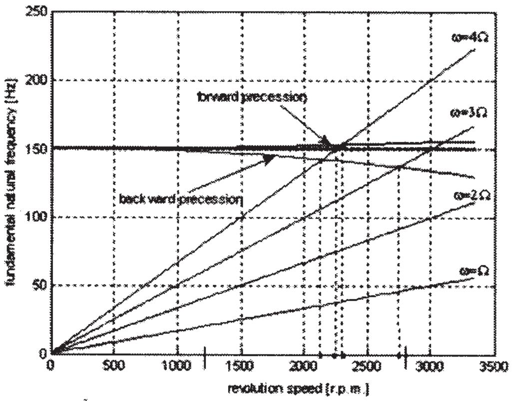

Fig. 3. Variation of fundamental natural frequency depending on the rotation speed for a basket wall thickness of 0.01 m

It is considered first of all the case in which the fixing point of the basket is found between the centre of mass and the bearing. Based on the relationships (11) and (12) the natural angular frequencies have been determined. The first natural angular frequency is interesting, meaning the fundamental natural angular frequency (ω_1), because the second natural angular frequency has a very high value.

Table 1 includes the values of the fundamental natural frequency for the horizontal and the vertical centrifuge for a speed of 2000 r.p.m. The distance from the centre of mass to the fixing point of the basket is 0.0605 m and the lateral wall thickness of the basket 0.01 m. It can benoticed that, in principle, the fundamental natural frequency is smaller in the case of a vertical centrifuge due to the action of the weight force of the basket.

By comparing the values of the fundamental natural frequency in the case of a horizontal centrifuge with those obtained in the case of a vertical centrifuge it can be said that the vertical positioning of the centrifuge does not influence the values of natural frequencies. The errors are very small, so the results obtained in the case of a horizontal centrifuge may be considered valid for the vertical centrifuge as well.

A rather higher error is obtained in the case of backward precession. This is due to the fact that the gyroscopic moment amplifies the shaft strain and thus the moment of reaction R_y with respect to the point B increases.

The small errors found in the table are due to the high rigidity of the shaft and that is why the weight force of the basket does not significantly influence the values of natural frequencies. The distance from the fixing point of the basket on the shaft to the near bearing, meaning the part of the shaft which is console mounted, is rather small and thus the weight force of the basket cannot influence significantly the values of natural frequencies. Furthermore, by analyzing the values of coefficients B_2 and B_3 (10) within the natural angular frequencies equation (9), it is noticeable that the terms containing the weight force of the basket and those containing the influence coefficients δ_{MB} and α_{MB} are 10^6 times smaller than those common in the natural angular frequencies equation for the horizontal centrifuge.

Figure 3 represents the variation of the fundamental natural frequency $\nu_1 = \omega_1 / 2\pi$ depending on the rotation speed. In order to determine the critical speeds the first 4 harmonics of the disturbance force have been represented. It is noticeable that within the considered functioning field of the centrifuge (1200 ÷2800 r.p.m.), critical rotation speeds correspondent to the third and fourth harmonics show up. The results are practically identical with those obtained in the case of a horizontal centrifuge. It is noticeable as well that the forward precession leads to the increase of the critical speeds, while the backward precession leads to the decrease of critical speeds.

Figure 4 represents the variation of the fundamental natural frequency depending on the distance from the centre of mass to the fixing point of the basket for a speed of 2000 r.p.m. The values for h have been considered negative when the fixing point of the basket is found

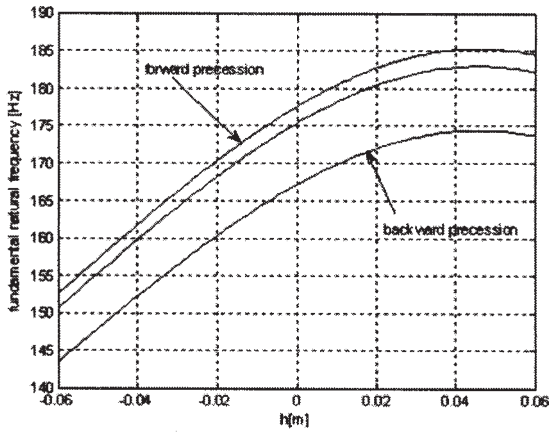

Fig. 4. Variation of fundamental natural frequency depending on the position of the fixing point of the basket on the shaft in comparison with the centre of mass

between the centre of mass and the bearing. The variation field considered for *h* is -0.06÷0.06 m, which covers all the cases studied theoretically. It is noticeable that the smallest values for the natural frequency are obtained when the centre of mass is found between the fixing point of the basket on the shaft and the bearing. The diagram is practically identical with the one traced for the horizontal centrifuge with a console mounted basket. While the fixing point of the basket gets closer to the centre of mass and goes beyond it, the fundamental natural frequency increases. In the case of forward precession the increase is of 16% and in the case of backward precession the increase is of 15%.

If the rotation speed varies as well, then it is obtained the diagram in figure 5 which represents the variation of the fundamental natural frequency depending on the rotation speed and on the distance from the centre of mass to the fixing point of the basket, for a basket wall thickness of 0.01m. It is noticed a significant decrease of the natural

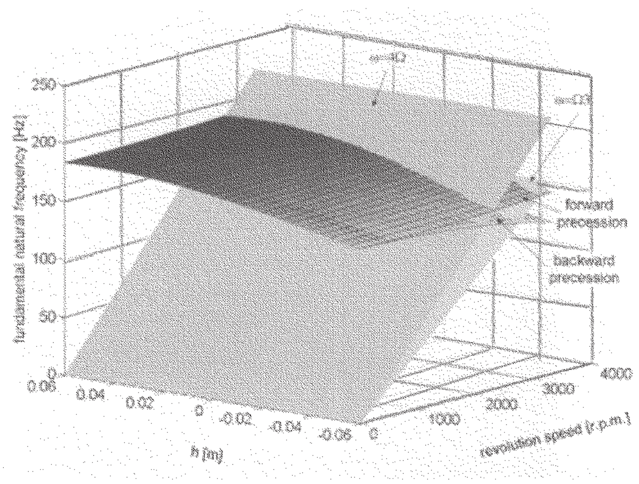

Fig. 5. Variation of fundamental natural frequency depending on the rotation speed and on the position of the fixing point of the basket on the shaft with respect to the centre of mass

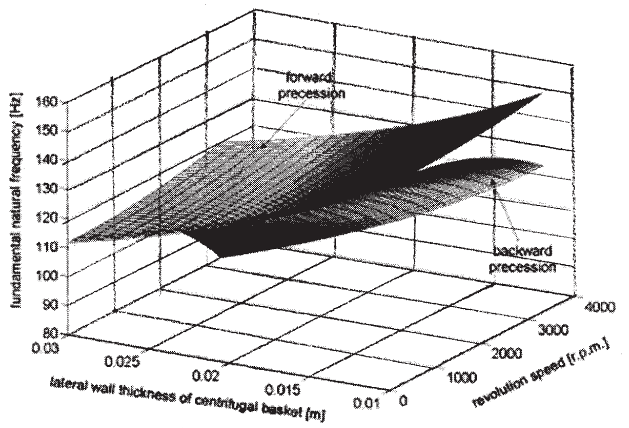

Fig. 6. Variation of fundamental natural frequency depending on the rotation speed and on the thickness of the side wall of the basket

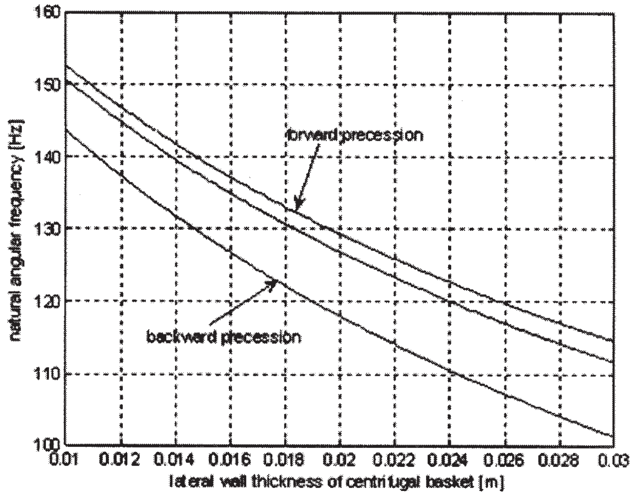

Fig. 7. Variation of fundamental natural frequency depending on the thickness of the side wall of the basket

frequency, corresponding to the backward precession, at high speeds and negative values for the distance between the centre of mass and the fixing point of the basket, when the fixing point of the basket is found between the centre of mass and the bearing.

While the centrifuge is in operation, the mass and the mechanical inertia moments of the basket will be modified as a result of the deposits on the side walls of the basket.

The variation of the fundamental natural frequency depending on the rotation speed and on the thickness of the side wall of the basket is represented in figure 6. It is obtained a diagram nearly identical with the one found in the case of a horizontal centrifuge. It is noticeable that, both in the case of forward and backward precession, the fundamental natural frequency decreases as the thickness of the side wall of the basket increases. For example, figure 7 represents the variation of the fundamental natural frequency depending on the thickness of the side wall of the basket for a speed of 2000 r.p.m. The variation field is

Table 2

THE VALUES OF FUNDAMENTAL NATURAL FREQUENCY IN THE CASE OF HORIZONTAL AND VERTICAL CENTRIFUGES

	Horizontal centrifuge [Hz]	Vertical centrifuge [Hz]	Error [%]
Negligible gyroscopic effect	111.6113	111.6085	0.0025
Forward precession	114.5187	114.5159	0.0024
Backward precession	101.1597	101.1565	0.0032

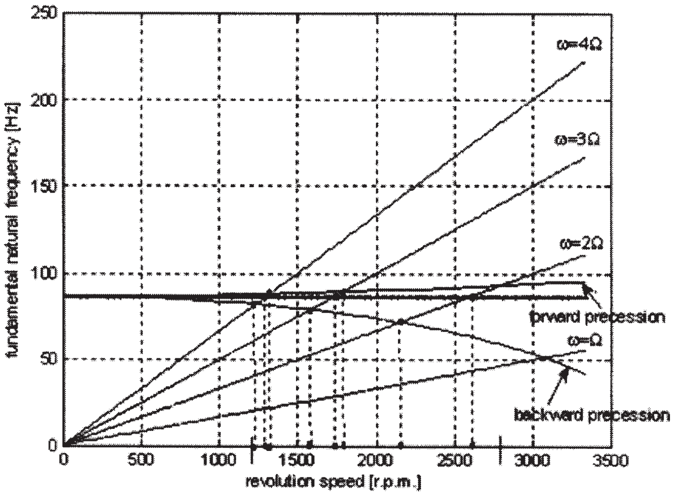

Fig. 8. Variation of fundamental natural frequency depending on the speed, for a basket wall thickness of 0.03 m.

0.01÷0.03 m. It is noticeable the decrease of the fundamental natural frequency as the thickness of the side wall of the basket increases. The decrease is of approximately 29% in the case of backward precession and of approximately 24% in the case of forward precession.

If the values of the fundamental natural frequency are compared, in the case of horizontal and vertical centrifuges, for the same values of speed (2000 r.p.m.) and of thickness of the side wall of the basket (0.03) it is noticeable a very good resemblance (table 2). Though the slight differences which occur are higher than those of the case represented in table 1, because together with the increase of the thickness of the basket wall the weight force which influences the values of natural frequencys increases as well. The influence is low due to the high rigidity of the shaft.

Taking into account the fact that, while the rotation speed increases, in the case of backward precession the natural frequency decreases, the critical speeds might decrease significantly if the thickness of the basket wall increases (through material deposits), thus there is the possibility of critical speeds in the functioning field corresponding to inferior harmonics. Figure 8 represents the variation of the fundamental natural frequency depending on the rotation speed for a thickness of the basket wall equal to 0.03m and the distance between the centre of mass and the fixing point of the basket equal to 0.0605 m, in case the fixing point of the basket is found between the centre of mass and the bearing. It is noticeable that, due to the increase of the basket wall thickness, the fundamental natural frequency decreases, which consequently leads to having in the functioning field critical speeds excited by the second harmonic, thus being more dangerous.

If the bearing near to the basket is flexible and the centrifuge functions near a resonance speed, then the amplitude of the movement aligned with the bearing would allow the moment of the weight force of the basket to increase, resulting in the increase of the movement amplitude. That is why it is recommended that, in case the functioning speed of the centrifuge is near a critical speed and cannot be avoided, at start and shutdown, the centrifuge must quickly pass through the resonance speed.

Conclusions

This dissertation presents a study model on the bending vibrations of vertical centrifuges with a console mounted basket. The study model with two degrees of freedom which is used offers the specialists the possibility to understand the phenomena which occur and to undertake the necessary measures in order to prevent them, or, in case this is not possible, to avoid them during functioning.

Although in the calculation relationships there are terms specific to the vertical centrifuges, which take into account the weight of the basket, their influence upon the results is rather low. Thus the results obtained in the study of the vibrations of horizontal centrifuges with a console mounted basket may be applied as well without problems in the case of vertical centrifuges with a console mounted basket.

The influence of the flexible constant of the bearings upon the horizontal and the vertical centrifuges may represent the topic of another dissertation.

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