

Flexion Movement Analysis of Horizontal Centrifuges Rotors Having the Basket Between the Bearings

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This dissertation presents a study model on the bending vibrations of horizontal centrifuges having the basket between the bearings. The shaft basket assembly is modelled as a discrete system with two degrees of freedom. The influence of the fixing point of the basket and of its centre of mass positioning and of the revolution speed upon its natural pulsations is herein studied. The influence of the gyroscopic couple is also considered.

Keywords: horizontal centrifuge, vibrations, natural frequencies, critical revolution speed

The dissertations [1, 2] study the bending vibrations of the horizontal centrifuges with a console mounted basket and of the centrifuges with a vertical axis as well as the influence of the construction and functional factors upon its natural pulsations.

For technological reasons, in practice, horizontal centrifuges with the basket mounted between the bearings are used.

In this paper the measure, in which the positioning of the basket between the bearings influences the natural frequencies of the system, comparing with the centrifuges with a console mounted basket is studied [1, 2].

- with regard to the centrifuges, in general, two aspects are taken into consideration:

- the system rigidity on which basis the angular speed of the shaft for its comparison with the critical angular speed resulted is determined [3];

- the flexion movement determined by positioning the masses (the basket) on the shaft.

In some technical literature related works [4, 5] the centrifuges are studied considering exclusively the basket construction. These papers do not cope with the problem of the centrifuges dynamic behaviour. In paper [6] the rotors within the process equipment are studied, considering exclusively the shafts resistance calculus.

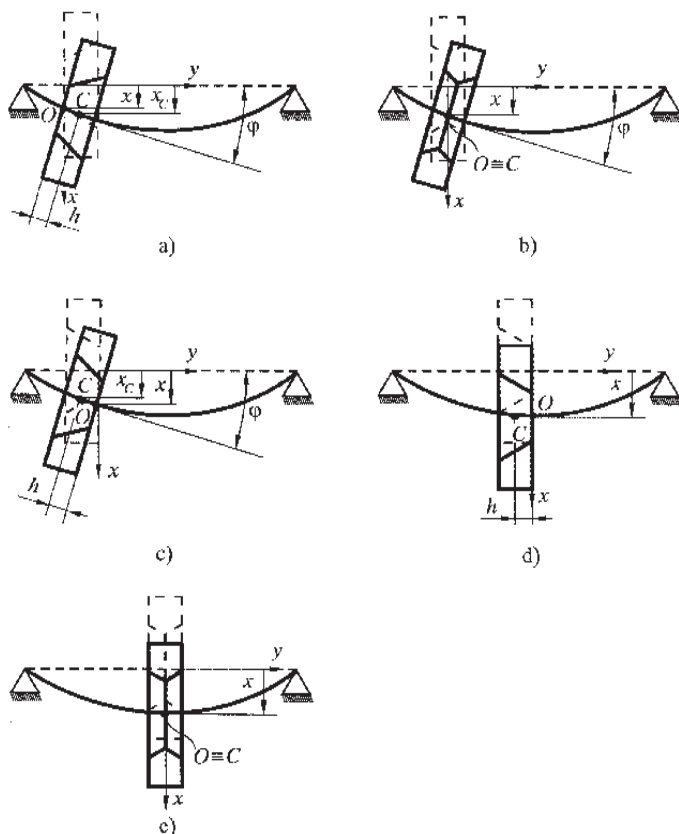


Fig. 1. The position of the fixing point of the basket against the bearings and the centre of mass in the case of a horizontal centrifuge with the basket between the bearings

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The calculus of the rotor rigidity of vertical centrifuges implies among others the study of the bending vibrations of this system as well. This paper has the object of calculating the critical angular speed (critical revolution speed) of the shaft. The data regarding the critical revolution speeds are necessary to avoid running the centrifuge on these revolution speeds or close to them.

This paper presents a study model for the vibrations of horizontal centrifuges having the basket between the bearings for the five situations that might occur. Figure 1 schematically presents these five cases: the basket is not set in the middle of the shaft, while the fixing point of the basket (O) is located between the centre of mass (C) and the bearing (fig. 1,a); the basket is not set on the middle of the shaft, while the centre of mass coincides with the fixing point of the basket (fig. 1,b); the basket is not set on the middle of the shaft, while the centre of mass is located between the fixing point of the basket and the bearing (figure 1,c); the basket is set on the middle of the shaft, while the centre of mass is located between the fixing point of the basket and the bearing (fig.1,d); the basket is set on the middle of the shaft, while the centre of mass coincides to the fixing point of the basket (fig.1,e). G is the weight of the basket together with the work material.

In order to study the bending vibrations the moulding of the centrifuges 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 (in this case between the bearing blocks);
- 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 (in this case between the bearing blocks).

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, as in the case of a centrifuge with a console mounted basket [1, 2] (fig. 1).

As only the critical revolution speeds are to be determined, this paper presents exclusively the undamped vibrations: free and maintained (forced). If the critical revolution speeds cannot be avoided and the centrifuge runs at revolution speed close to the critical ones, the amplitudes of the bending vibrations can be reduced by mounting a dynamic vibration damper or absorber. The possible bending vibrations control through these methods may be the object of another paper.

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 functioning due to uneven material deposits on the basket or due to damages occurred during the functioning. Thus, just like in the case of horizontal centrifuges with a console mounted basket, the elements of reducing the disturbance force in point O are obtained:

$$\tau_o^p \begin{cases} F_p = me\Omega^2 \cos \Omega t \\ M_p = mehe\Omega^2 \cos \Omega t \end{cases} \quad (1)$$

where:

m is the mass of the basket;

Ω - the angular speed of the shaft basket assembly;

e - the eccentricity (distance between the rotation axis and the centre of mass of the basket).

One factor that influences the values of natural pulsations is the gyroscopic moment. The gyroscopic moment modifies the bending strain, and thus the natural pulsation and the critical revolution speed as well. As opposed to the centrifuges with a console mounted basket, when studying the bending vibrations of the horizontal centrifuges with the basket between the bearings, the gyroscopic moment is considered only when the basket is not mounted in the middle of the shaft (fig.1, a, b, c, d).

As in the case of centrifuges with a console mounted basket [1, 2], 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 called forward synchronous rotation. The gyroscopic moment modulus is $M_g = (J_y - J_z)\Omega^2\varphi$ [7]. If the flexion plane and the shaft rotate in contrary directions with the same angular speed $\omega = \Omega$ the movement is called backward synchronous rotation. The gyroscopic moment modulus is [7] $M_g = (J_y + J_z)\Omega^2\varphi$.

The study of free and forced vibrations

The fixing point of the basket is not found in the middle of the distance between the bearings

The fixing point of the basket is found between the centre of mass and the bearing

The model used for study is that of two degrees of freedom (fig. 1). The basket is built 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.

As in the case of the horizontal centrifuge with a console mounted basket, in order to write the differential equations

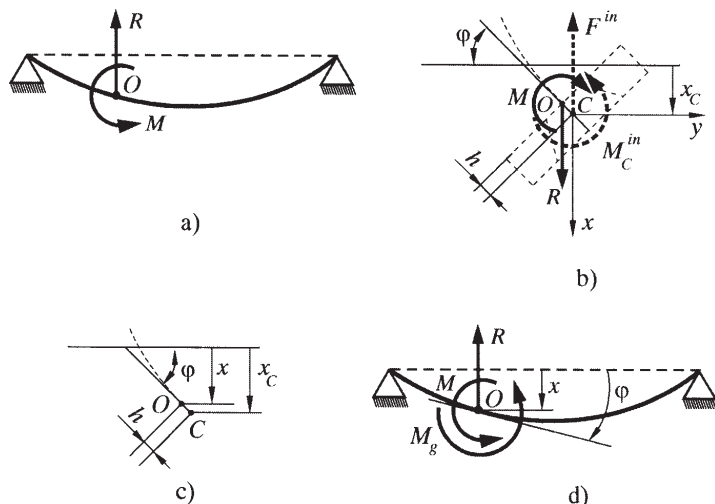


Fig. 2. The fixing point of the basket is not found in the middle of the distance between the bearings. The fixing point of the basket is found between the centre of mass and the bearing. The calculus of the reactions in the fixing point of the basket on the shaft

of the movement the method of influence coefficients is used. For this purpose, the basket is isolated, and the d'Alembert principle is applied (fig. 2,b) for the calculus of the reactions R and M .

For this case the results obtained in [1] can be used. The influence coefficients have to be recalculated though δ_F , δ_M , α_F , α_M [9].

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

Using the influence coefficients method we can write:

$$\begin{cases} x = -a_1 \ddot{m}x - [\delta_F mh + \delta_M a_2] \ddot{\varphi} - \delta_M a_3 \Omega^2 \varphi \\ \varphi = -a_4 \ddot{m}x - [\alpha_F mh + \alpha_M a_2] \ddot{\varphi} - \alpha_M a_3 \Omega^2 \varphi \end{cases} \quad (2)$$

where δ_F and δ_M are influence coefficients which represent the displacements in point O produced by a force and a moment equal to one unit; α_F , α_M - are influence coefficients which represent the rotations in point O produced by a force and a moment equal to one unit;

$$a_1 = \delta_F + \delta_M h;$$

$$a_2 = mh^2 + J_{Cz};$$

$$a_3 = J_{Cy} - J_{Cz};$$

Ω is the angular speed of the shaft;

J_{Cz} - the mechanical moment of inertia of the basket to the axis Cz which crosses the centre of mass and is perpendicular to the sheet plane;

J_{Cy} - the mechanical moment of inertia of the basket to the axis Cy which crosses the centre of mass and which is the theoretical rotation axis.

We search for phase synchronically solutions in the form of:

$$\begin{cases} x = X \cos(\omega t - \psi) \\ \varphi = \Phi \cos(\omega t - \psi) \end{cases} \quad (3)$$

Replacing within relations (2) we obtain the linear and homogeneous algebraic system for the unknown variables X and Ω .

$$\begin{cases} [a_1 m \omega^2 - I] X + [\delta_F mh + \delta_M a_2] \omega^2 - \delta_M a_3 \Omega^2 \Phi = 0 \\ a_4 m \omega^2 X + [\alpha_F mh + \alpha_M a_2] \omega^2 - \alpha_M a_3 \Omega^2 - I \Phi = 0 \end{cases} \quad (4)$$

From the condition for that the system (4) should allow solutions different from the common solution, we obtain the following equation of natural pulsations:

$$B_1 \omega^4 - B_2 \omega^2 + B_3 = 0 \quad (5)$$

where:

$$B_1 = a_5 m J_{Cz}; \quad B_2 = [\delta_F m + a_6 mh + \alpha_M a_2 + a_5 a_3 m \Omega^2];$$

$$B_3 = \alpha_M a_3 \Omega^2 + I,$$

in which the following notation 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 for the natural pulsations:

$$\omega_1 = \sqrt{\frac{B_2 - \sqrt{B_2^2 - 4B_1B_3}}{2B_1}}; \quad \omega_2 = \sqrt{\frac{B_2 + \sqrt{B_2^2 - 4B_1B_3}}{2B_1}} \quad (6)$$

In the case of forced vibrations the elements of reducing in the point O of the disturbance forces, F_p and M_p [1], intervene. Thus the system (4) becomes:

$$\begin{cases} [a_1 m \Omega^2 - I] X + \Omega^2 [a_7 - \delta_M a_3] \Phi = -me \Omega^2 a_1 \\ a_4 m \Omega^2 X + [a_8 \Omega^2 - \alpha_M a_3 \Omega^2 - I] \Phi = -me \Omega^2 a_4 \end{cases} \quad (7)$$

where the following notations have been used:

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

For the permanent case of movement,

$$\begin{cases} x = X \cos \Omega t \\ \varphi = \Phi \cos \Omega t \end{cases} \quad (8)$$

we obtain the expressions of the amplitudes of the movements in the case of forced vibrations:

$$X = \frac{me \Omega^2 a_1 + me \Omega^4 a_3 [a_3 - J_{Cz}]}{-m \Omega^4 a_3 [a_3 - J_{Cz}] + \Omega^2 [\alpha_M a_3 - ma_1 - a_8] + I} \quad (9)$$

$$\Phi = \frac{me \Omega^2 a_4}{-m \Omega^4 a_3 [a_3 - J_{Cz}] + \Omega^2 [\alpha_M a_3 - ma_1 - a_8] + I} \quad (10)$$

- The case of backward precession

Following the same path of calculus as in the direct precession case, we obtain the natural pulsations and the amplitudes of the permanent movement (in case of forced vibrations). The only difference is that in all the equations the expression of the gyroscopic moment $M_g = (J_y - J_z) \Omega^2 \varphi$ is replaced with $M_g = -(J_y + J_z) \Omega^2 \varphi$. In other words, in all the equations we replace $a_3 = J_{Cy} - J_{Cz}$ by $a_3 = -(J_{Cy} + J_{Cz})$.

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

This case is characterized by the fact that $h = 0$. Thus, both in the case of forward and backward precession, the relations from which we determine the natural pulsations and the amplitudes of the permanent movement (in the case of forced vibrations) are obtained from the relations (6), (9), (10) through customization $h = 0$.

The centre of mass is located between the fixing point of the basket on the shaft and the bearing

The study of vibrations is in this case similar with that in paragraph 2.1.1. The difference lies in $h < 0$. Thus, both in the case of forward and backward precession, the relations from which we determine the natural pulsations and the amplitudes of the permanent movement (in the case of forced vibrations) are obtained from the relations in paragraph (6), (9), (10) by replacing h with $-h$.

The fixing point of the basket is in the middle of the distance between the bearings

The fixing point of the basket on the shaft does not coincide with the centre of mass

In this case the influence coefficients δ_F , δ_M , α_F , α_M must be recalculated. The conclusions are $\delta_M = 0$ and $\alpha_F = 0$.

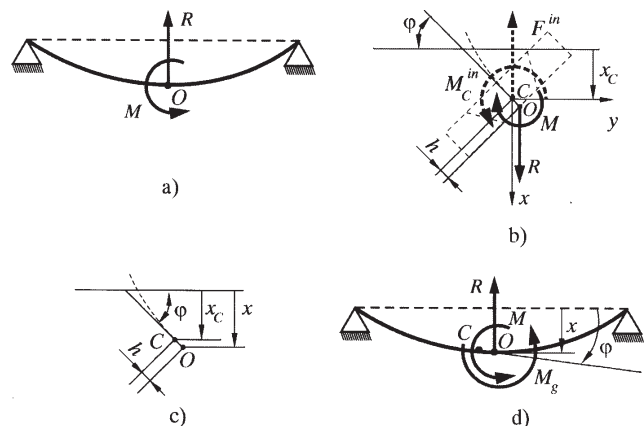


Fig. 3. The fixing point of the basket is set in the middle of the distance between the bearings. The calculus of reactions in the fixing point of the basket on the shaft

For the calculus of the reactions, R and M , from point O (fig. 3.a), we proceed as in paper [1]. The basket is isolated, and the d'Alembert principle is applied (fig. 3,b) taking into account that

$$x_C = x - h \sin \varphi \approx x - h \varphi. \quad (11)$$

We obtain for R and M the expressions

$$\begin{cases} R = m\ddot{x} - mh\ddot{\varphi} \\ M = -mh\ddot{x} + (mh^2 + J_{Cz})\ddot{\varphi} \end{cases} \quad (12)$$

- The case of forward precession (fig. 5,d)

Using the influence coefficients method we write:

$$\begin{cases} x = \delta_F(-R) \\ \varphi = \alpha_M(-M - M_g) \end{cases} \quad (13)$$

Taking into account the equations (12) and the expression of the gyroscopic moment we obtain the equations:

$$\begin{cases} x = -a_1 m \ddot{x} - [-\delta_F m h + \delta_M a_2] \ddot{\varphi} \\ \varphi = -a_4 m \ddot{x} - \alpha_M a_2 \ddot{\varphi} - \alpha_M a_3 \Omega^2 \varphi \end{cases} \quad (14)$$

where: $a_1 = \delta_F$; $a_2 = mh^2 + J_{Cz}$; $a_3 = J_{Cy} - J_{Cz}$; $a_4 = \alpha_M h$.

The equations (14) are different from those in paragraph 2.1.3 as they lack the terms containing the coefficients δ_M , α_F . Thus, we can use the equations obtained in paragraph 2.1.3 in which we replace $\delta_M = 0$ and $\alpha_F = 0$.

The natural pulsations may be determined using the equations in chapter 2.1, (5), (6) in which

$$\begin{aligned} B_1 &= a_5 m J_{Cz}; \quad B_2 = [\delta_F m + \alpha_M a_2 + a_5 a_3 m \Omega^2]; \\ B_3 &= \alpha_M a_3 \Omega^2 + 1. \end{aligned}$$

The notation $a_5 = \delta_F \alpha_M$ has been used.

As in the case of forced vibrations we can use the equations obtained in paragraph 2.1.3, in which we replace $\delta_M = 0$ and $\alpha_F = 0$. Thus we obtain the expressions of the permanent movement amplitudes.

- The case of backward precession

We proceed just like in chapter 2.1. We use the equations for the natural pulsations and for the permanent movements amplitudes established for the forward precession, in which we replace the expression of the gyroscopic moment $M_g = (J_y - J_z)\Omega^2\varphi$ with $M_g = -(J_y + J_z)\Omega^2\varphi$. In other words, in all the equations we replace $a_3 = J_{Cy} - J_{Cz}$ by $a_9 = -(J_{Cy} + J_{Cz})$.

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

In this case the basket has only one degree of freedom. It will be the case of vibrations with only one degree of freedom. The gyroscopic moment does not act anymore (fig. 4.b).

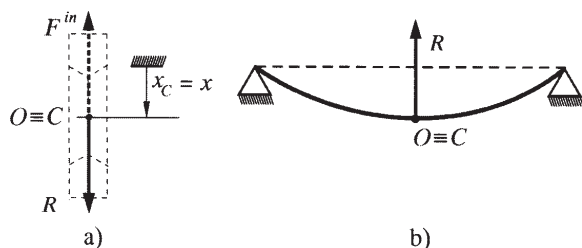


Fig. 4. The fixing point of the basket is in the middle of the distance between the bearings. The fixing point coincides with the centre of mass

The movement parameter will be the displacement of the centre of mass of the basket x (fig. 4.a). For calculating the reaction R the basket is isolated and the d'Alembert principle is applied (fig. 4.a). The differential equation of the movement is written using the influence coefficients method:

$$\delta_F m \ddot{x} + x = 0 \quad (15)$$

and

$$\ddot{x} + \frac{1}{\delta_F m} = 0. \quad (16)$$

Thus, it results the formula for the natural pulsation:

$$\omega = \sqrt{\frac{1}{\delta_F m}}. \quad (17)$$

In the case of forced vibrations the disturbance force intervenes $F_p = me\Omega^2 \cos \Omega t$. The differential equation of the movement becomes:

$$\ddot{x} + \frac{1}{\delta_F m} = e\Omega^2 \cos \Omega t. \quad (18)$$

For the case of the permanent movement, when $x = X \cos \Omega t$, we obtain the following expression for the forced vibrations amplitude:

$$X = \frac{e}{\left(\frac{\omega}{\Omega}\right)^2 - 1}. \quad (19)$$

Application

We consider the case of a horizontal centrifuge with the basket between the bearings. We consider 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], except the basket is not console mounted:

- 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/m³
- transversal elasticity modulus: 21x10¹⁰ N/m²;
- distance from the longitudinal axis of the basket to the centre of mass (eccentricity e): 0.003 m.

For the influence coefficients the following values have been obtained:

$$\delta_F = 1.2737 \times 10^{-8} \text{ m} \cdot \text{N}^{-1};$$

$$\delta_M = 4.0526 \times 10^{-8} \text{ N}^{-1};$$

$$\alpha_F = 4.0526 \times 10^{-8} \text{ N}^{-1};$$

$$\alpha_M = 2.4473 \times 10^{-7} \text{ m}^{-1} \cdot \text{N}^{-1}.$$

The mechanical moments of inertia:

$$J_{Cy} = 1.8755 \text{ kg} \cdot \text{m}^2;$$

$$J_{Cz} = 1.0628 \text{ kg} \cdot \text{m}^2.$$

We have 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 with a console mounted basket [1], in order to investigate in which extent the positioning of the basket between the bearings influences the values of the natural pulsations.

We consider first of all the case in which the fixing point of the basket on the shaft is not found between the centre of mass and the bearing. The fixing point is found between the centre of the mass and the bearing. Based on the equations (6) the natural pulsations have been obtained.

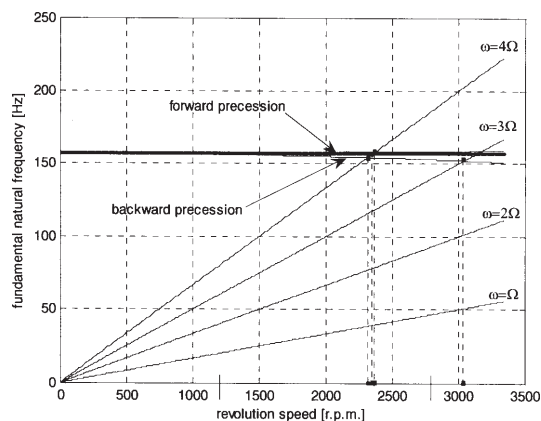


Fig. 5. The variation of the fundamental natural frequency (v_1) with the revolution speed.

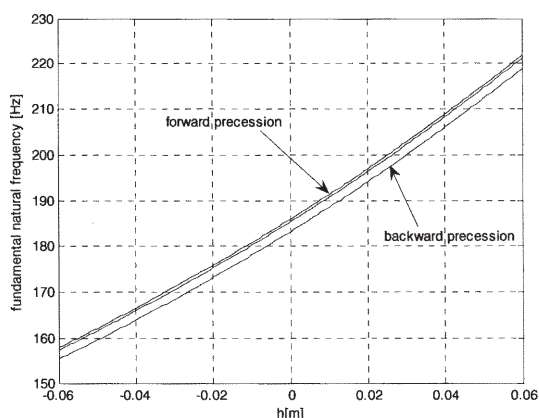


Fig. 6. Variation of fundamental natural frequency (v_1) depending on the position of the fixing point of the basket on the shaft against the centre of mass

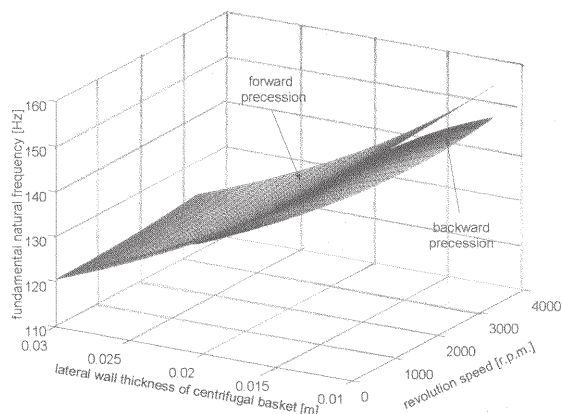


Fig. 7. Variation of fundamental natural frequency (v_1) depending on the side wall thickness and the revolution speed

The first natural pulsation is interesting, meaning the fundamental natural pulsation (ω_1), because the second natural pulsation has a very high value.

In figure 5 the variation of the fundamental natural frequency $v_1 = \omega / 2\pi$ with the revolution speed is presented. This is about 5% higher than in the case of the centrifuge with a console mounted basket [1]. In order to determine the critical revolution speeds the first 4 harmonics of the disturbance force have been represented. We can notice that within the considered functioning field of the centrifuge (1200 ÷ 2800 rot/min), critical revolution speeds correspondent to the harmonic 4, as opposed to the case of the centrifuge with a console mounted basket [1] for which critical revolution speeds correspondent to the harmonics 3 and 4 show up. The influence of the forward and backward precession on the critical revolution

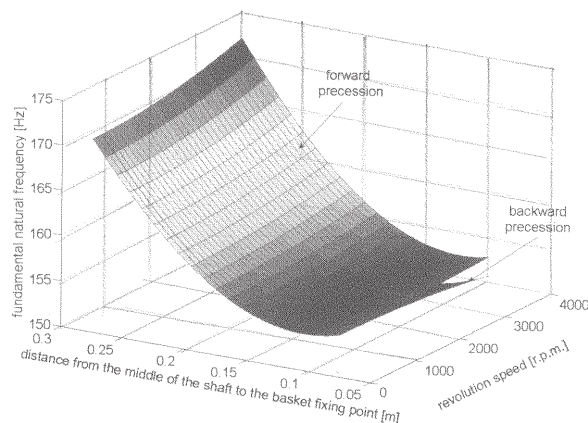


Fig. 8. Variation of fundamental natural frequency (v_1) depending on the distance between the fixing point of the basket and the middle of the shaft and depending on the revolution speed

speeds is smaller relative to the case of a centrifuge with a console mounted basket. This is why the resultant critical revolution speeds are contained in a less wide domain.

Figure 6 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 revolution speed of 2000 rpm. The variation field considered for h is $-0.06 \div 0.06$ m. This way are treated all the cases mentioned in paragraphs 2.1.2. and 2.1.3. The values for h have been considered as negative when the fixing point of the basket is found between the centre of mass and the bearing. We can notice 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 forward and backward precession do not lead to bigger variations of the fundamental natural frequency. While the fixing point of the basket gets closer to the centre of mass and goes beyond it, the fundamental natural frequency increases. The increase is of approximately 40%, much higher than in the case of a horizontal centrifuge with a console mounted basket [1], for which the increase was of 15-16%.

During the centrifuge functioning, through material accumulation on the side walls of the basket the mass and the mechanical inertia moments of the basket change. Figure 7 represents the fundamental natural frequency variation depending on the rotation and on the thickness of the side wall of the basket. The variation field for the basket wall thickness is $0.01 \div 0.03$ m. As opposed to the case of the horizontal centrifuge with a console mounted basket [1], the forward and backward precession do not lead to higher variations of the fundamental natural frequency. Taking into account the fact that while the revolution speed increases, in the case of backward precession the natural frequency decreases, it may happen that, if the side wall thickness increases (through material accumulation), the critical revolution speeds may significantly decrease, thus there is the possibility of critical revolution speeds in the functioning field corresponding to inferior harmonics. For example, at a revolution speed of 2000 rpm we can notice a decrease of the fundamental natural frequency together with the increase of the side wall thickness of the basket. The decrease is approximately 24% in case of backward precession and of approximately 23% in case of forward precession.

Figure 8 represents the fundamental natural frequency variation (v_1) depending on the distance between the fixing point of the basket against the middle of the shaft and depending on the revolution speed. We can notice that,

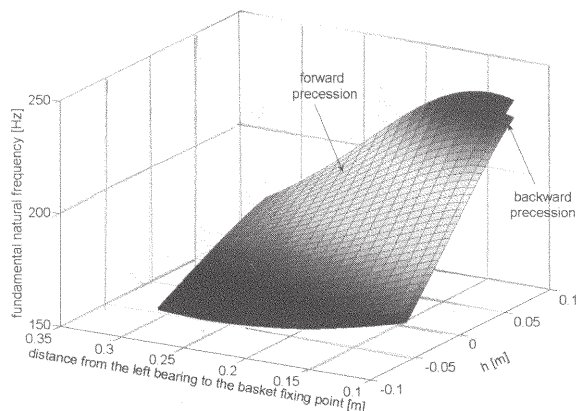


Fig. 9. The variation of the fundamental natural frequency (v_1) depending on the distance from the left bearing to the fixing point of the basket and depending on the positioning of the basket fixing point against the centre of mass

while the fixing point of the basket on the shaft gets closer to the bearing, the fundamental natural frequency increases up to 10% if the distance to the bearing is of 0.2m. The variation with a smaller revolution speed reaches up to 2%. The forward and backward precession do not lead to higher variations of the fundamental natural frequency.

Figure 9 presents the variation of the fundamental natural frequency depending on the distance from the left bearing to the fixing point of the basket on the shaft against the centre of mass. Just like the centrifuge with a console mounted basket, the fundamental natural frequency varies depending on the basket centre of mass positioning against the basket fixing point on the shaft. Thus, while the basket of the centrifuge gets closer to the bearing, the variation of the fundamental natural frequency depending on the basket centre of mass positioning against the fixing point is higher, reaching up to 40%. The negative values of h have been considered when the fixing point of the basket is set between the centre of mass and the bearing. We can notice that the smallest values for the natural frequency are obtained when the basket fixing point on the shaft is set between the centre of mass and the bearing, just like in the case of a horizontal centrifuge with a console mounted basket. As the fixing point of the basket gets closer to the centre of mass and goes beyond it, the fundamental natural frequency increases.

Figure 10 represents the variation of the fundamental natural frequency depending on the revolution speed and on the distance between the centre of mass and the fixing point of the basket, for a basket side wall thickness of 0.01m. The considered variation field for h is $-0.06 \div 0.06$ m, which covers all the cases theoretically treated. We can notice that the smallest values for the natural frequency are obtained when the basket fixing point on the shaft is set between the centre of mass and the bearing. As the basket fixing point gets closer to the centre of mass and goes beyond it, the fundamental natural frequency increases. The position of the centre of mass against the basket fixing point influences the fundamental natural frequency in a higher extent than the revolution speed. The influence of the forward and backward precession on the critical revolution speeds is smaller than in the case of a horizontal centrifuge with a console mounted basket.

Conclusions

This paper presents a study model on the bending vibrations of horizontal centrifuges with the basket between the bearings. Though it is a simplified model, it offers the specialists the possibility to understand the phenomena

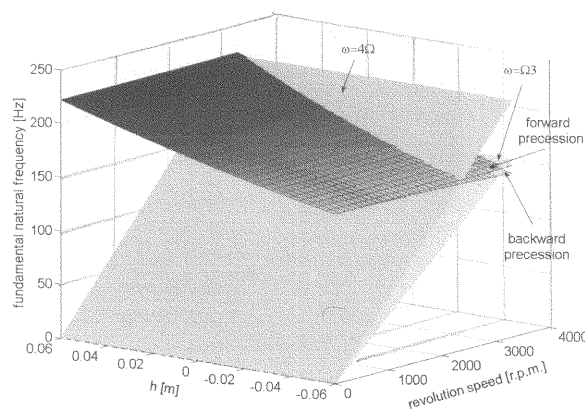


Fig. 10. The variation of fundamental natural frequency (v_1) depending on the revolution speed and the positioning of the fixing point of the basket against the centre of mass.

which occur and to undertake the necessary measures in order to prevent them, or, in case this is not possible, to avoid them during the functioning.

The positioning of the basket between the bearings leads to higher values of the fundamental natural frequency. The variation of the fundamental natural frequency depending on the distance between the centre of mass and the fixing point of the basket on the shaft is higher than in the case of the centrifuge with a console mounted basket. The phenomena of forward and backward precession lead to small variations of the fundamental natural frequency depending on the revolution speed. This leads to a less wide field of critical revolution speeds dissipation.

During the centrifuges functioning, due to their constructive and functional features, mechanical features changes may occur on this equipment, such as the position of the centre of masses and the basket mass. The cause lies in the ununiform deposits of material. This have the effect of changing the dynamic behaviour of the centrifuge, meaning that vibrations with non neglectable amplitudes occur, producing undesirable noises, bearing wearing and decrease of product quality. The fact that upon planning the specialists take into consideration certain set of data which changes during the functioning represents a problem leading to the impossibility of predicting the dynamic behaviour of the centrifuge.

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

References

1. BOIANGIU, M., SIMA, T., ALECU, A., Rev. Chim. (Bucharest), **61**, no. 11, 2010, p.1119
2. SIMA, T., BOIANGIU, M., ALECU, A., Rev. Chim. (Bucharest), **62**, no. 1, 2010, p. 102
3. JINESCU, V. V., Utilaj tehnologic pentru industrii de proces, vol. 3, Editura Tehnică, București, 1988
4. IATAN, I. R., STOICAN, E., BOTE, N., HRISTESCU, C., Rev. Chim. (Bucharest), **36**, nr.12, 1985, p. 1138
5. IATAN, I. R., JUGĂNARU, M., ȘTEFĂNESCU, M., Rev. Chim. (Bucharest), **41**, nr.1, 1990, p. 67
6. JINESCU, V. V., PRODEA, I.-M., Rev. Chim. (Bucharest), **58**, nr.6, 2007, p. 582
7. RĂDOI, M., DECIU, E., VOICULESCU, D., Elemente de vibrații mecanice, Editura Tehnică, București, 1973
8. VOINEA, R., VOICULESCU, D., CEAUȘU, V., Mecanică, Editura Didactică și Pedagogică, București, 1983
9. VOINEA, R., VOICULESCU, D., SIMION, FI. P., Introducere în mecanica solidului, Editura Academiei Române, 1988

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