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Automatic Balancing Simulation of the Centrifugal Grinding-Mixing Unit

Dmitriy Alexandrovich Bushuev, Vasiliy Grigorievich Rubanov and Nicolay Dmitrievich Vorobyov
Belgorod State Technological University Named after V.G. Shukhov,
Kostyukova Street 46, 308012 Belgorod, Russia

Abstract: The study describes the getting of models to estimate changes of the dynamic bearing responses of the centrifugal grinding-mixing unit with automatic balancing and its resultant moment. Using the d'Alembert principle the mathematical model to estimate changes of the dynamic bearing responses and torque depending on the positions of the counterweights shoulders is derived. In the Adams Software, the parametric computer model of balancing of the unit with the same initial parameters is built. For specific optimal positions of the counter weights dynamic responses and torque using the mathematical and computer models are calculated and their identity is shown. Integration of the computer mechanical model into a simulation model of the control systems in the MATLAB/SIMULINK Software is shown. Changing of the dynamic bearing responses of the centrifugal grinding-mixing unit when changing the position of the counter weights and when the material moving through the grinding chambers from the top to the bottom, resulting in a co-simulation using ADAMS and SIMULINK are illustrated. The prospects of using the resulting computer model are described.

Key words: Grinding-mixing unit, dynamic loads, balancing, shoulder counterweight, multi-body dynamics simulation

INTRODUCTION

Energy-efficient Centrifugal Grinding-Mixing Unit (CGMU) used for the production of fine materials and homogeneous composite mixtures (Glagolev *et al*., 2011) is based on the most common type lever slider-crank mechanism (Anis, 2012) used in compressors, internal combustion engines, etc. Due to the kinematic scheme of slider-crank mechanism top working chamber of CGMU performs translational motion, the average-plane-parallel along an elliptical path, and bottom-rotational (Sevostyanov *et al*., 2005).

When links of slider-crank mechanism move with variable speeds due to changing the inertia forces periodic dynamic loads occur which are the source of undesirable oscillations of individual links of the mechanism. These vibrations are transmitted to the joints and machine parts containing slider-crank mechanisms (for example, bearing rod of CGMU) and cause fatigue effects there by reducing their residual life and overall reliability of the units and machines (Bushuev *et al*., 2014).

There are many methods of balancing the slider-crank mechanisms to improve their vibrational state (Arakelian and Smith, 2005). The main ones are: Balancing by a counterweights attached to the links (Artobolevskii, 1968; Shchepetilnikov, 1982). These methods are based on the redistribution of mass of the mechanism by adding counter weights to the moving links where in the center of mass does not change its position and thus disturbing forces does not appear. It should be noted that such balancing may be achieved only by a substantial increase in the mass of the mechanism.

Balancing the necessary order harmonics of the unbalanced forces and their moments of inertia using counter rotating masses (Artobolevskii, 1968; Shchepetilnikov, 1982; Lanchester, 1915; Hirokazu *et al*., 1976). These solutions are based on the harmonic analysis. Reduced inertia effects are primarily achieved by balancing certain harmonics disturbing forces and moments. Unbalanced forces and moments are approximated by the Fourier series and then studied for each frequency order. This approach was successfully applied for balancing internal combustion engines.

Self-balancing using dual mechanism (Arakelian, 2006; Filonov and Petrikovetz, 1987; Turbin *et al*., 1978; Uralskiy and Sevostyanov, 2010). Adding axisymmetric duplicate mechanisms captures the center of mass and thus, balances the disturbing force. This approach involves the construction of self-equilibrated mechanical

Corresponding Author: Dmitriy Alexandrovich Bushuev, Belgorod State Technological University Named after V.G. Shukhov, Kostyukova Street 46, 308012 Belgorod, Russia

systems in which two identical mechanisms perform similar but opposite movement. In this case, the disturbing force is entirely possible to implement mutually exclusive or partial balancing. It should be noted that due to the duplication the original design of the mechanism becomes more complicated.

Balancing by using the effect of self-synchronization (Zamyatin and Dubovik, 2004). Balancing occurs automatically by synchronizing the movement of pendulums and crank.

However, slider-crank mechanism as a part of CGMU (Fig. 1) has following features limiting the application of the above methods:

In the process of grinding as a result of the movement of the stuff the center of the mass is changed which makes it necessary to stop the machine and to change the position of the counter weights.

Different grinding bodies are used in the chambers depending on the nature of the stuff grinding in CGMU, therefore, load factor is varied which leads to the necessity of changing the counterweights position.

Values of the inertia forces are of the same order as the static forces because of the large mass of links with the grinding chamber and a small length of the crank (r_C = 2 cm).

Because of these features grinding-mixing units with automatic balancing were designed (Rubanov *et al*., 2012; Glagolev *et al*., 2014). In which in accordance with the

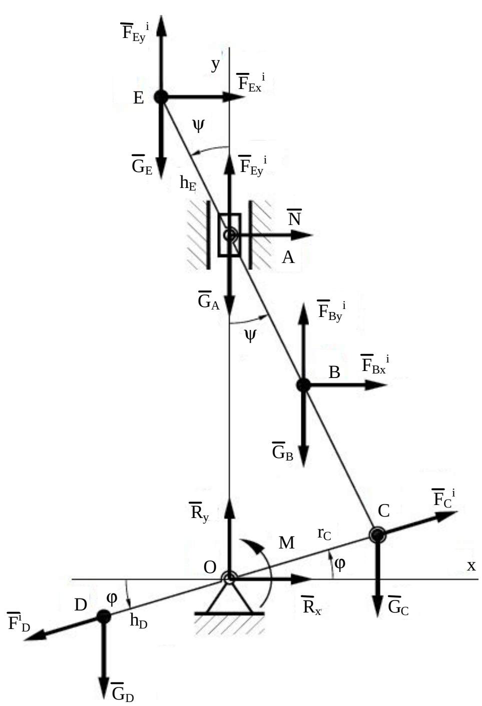

Fig. 1: Force diagram of linkage of the CGMU

command variable from the control systems based on programmable controllers the position of the counterweights is changed due to the existing energy of rotation or external input energy, leading them to the appropriate state of the minimum value of vibration. To carry out simulation and subsequent synthesis of control systems it is necessary to have a model of balancing which would take into account the above features of the CGMU.

Vibratory condition of units and machines is largely determined by unbalanced inertia forces of rotating and reciprocating translational motion of the masses (Skvorchevskiy *et al*., 2009). Therefore, in this research the task of obtaining the mathematical and computer models which can define the dynamic bearing responses of centrifugal grinding-mixing unit with automatic balancing are considered.

MATERIALS AND METHODS

To obtain a mathematical model of CGMU balancing it is necessary to find by using the d'Alembert principle, the analytical expressions for the dynamic loads arising in the bearings and guides.

Consider balancing of the CGMU linkage with two counterweights (the most common case) from the calculation scheme shown in Fig. 1.

Figure 1 A, B and C-CGMU grinding chamber, D and E counter weights, G_A, G_B and G_C-gravity forces of grinding chambers, G_D and G_E-gravity forces of counterweights, F_A, F_B and F_C the inertial forces of grinding chamber, F_D^i, F_E^i-the inertia forces of counter weights, M-torque applied to the link OA, R_x, R_y-dynamic reactions in bearings, N-normal pressure force in the slides, φ the angle determining the current mechanism position.

Mass of the frame and the eccentric shaft is reduced to the respective masses of grinding chambers. Weights of the links DOC and CAE are neglected due to of their smallness compared with weights of the grinding chambers and counter weights. The friction force is also ignored with regard to its smallness as in the guides is used special grease.

Inertial forces of grinding chamber C and counterweight D as DC link makes uniform rotational motion are determined only by the normal component of acceleration directed along the link DC so:

$$F_C^i = m_C\omega^2 r_C \tag{1}$$

$$F_D^i = m_D\omega^2 h_D \tag{2}$$

Where:

m_C and m_D = Masses of chamber C and counterweight D, respectively
[φ] = Angular velocity of the link OC
r_C and h_D = Length of the links OC and OD, respectively

Directions of the inertia forces are shown in Fig. 1. To calculate the inertia forces of other chambers and counter weight define their acceleration. In the Cartesian coordinate system xOy (Fig. 1) we have following trigonometric functions:

$$\begin{aligned} \sin\psi &= \lambda\cos\varphi \\ \cos\psi &= \sqrt{1-\sin^2\psi} \end{aligned} \quad (3)$$

where, $\lambda = r_C/l$, l-length of link CA. Then, from the Eq. 3 follows:

$$\cos\psi = \sqrt{1-\lambda^2\cos^2\varphi} \quad (4)$$

Ratio [lambda] for CGMU is small so, it is advisable to expression Eq. 4 is expanded using the binomial theorem:

$$\begin{aligned} \cos\psi &= \sqrt{1-\lambda^2\cos^2\varphi} \\ &= \left(1-\frac{\lambda^2}{4}-\frac{3}{64}\lambda^4-...\right)-\left(\frac{\lambda^2}{4}+\frac{\lambda^2}{16}+\frac{15}{512}\lambda^4+...\right)\cos 2\varphi+ \\ &\left(\frac{\lambda^2}{64}+\frac{3}{256}\lambda^6+...\right)\cos 4\varphi\pm... \end{aligned}$$

When considering industrial designs of CGMU high degree of accuracy is given by approximation in the following form:

$$\cos\psi = \sqrt{1-\lambda^2\cos^2\varphi} = \left(1-\frac{\lambda^2}{4}\right)-\frac{\lambda^2}{4}\cos 2\varphi \quad (5)$$

Coordinates of the grinding chamber B can be written as follows:

$$\begin{aligned} x_B &= \frac{r_C}{2}\cos\varphi \\ y_B &= r_C\sin\varphi+\frac{1}{2}\cos\psi \end{aligned} \quad (6)$$

Double differentiating the resulting equations over time and considering expression Eq. 5 find:

$$\begin{aligned} \ddot{x}_B &= a_{Bx} = -\frac{\omega^2 r_C}{2}\cos\varphi, \\ \ddot{y}_B &= a_{By} = -\omega^2 r_C\left(\sin\varphi-\frac{\lambda}{2}\cos 2\varphi\right) \end{aligned} \quad (7)$$

In the derivation of Eq. 7 it was taken into account that:

$$\frac{dx}{dt} = \frac{d\varphi}{dt}\frac{dx}{d\varphi}$$

and $d\varphi/dt = \varphi = \omega$. Thus, the inertia force to be added to grinding chamber has projections on the coordinate axes:

$$\begin{cases} F^i_{Bx} = -m_B a_{Bx} \\ F^i_{By} = -m_B a_{By} \end{cases} \quad (8)$$

Similarly, for the grinding chamber A:

$$y_A = r_C\sin\varphi+l\cos\psi \quad (9)$$

(coordinate x_A = const = 0) and taking into account Eq. 7:

$$\ddot{y}_A = a_{Ay} = -\omega^2 r_c(\sin\varphi-\lambda\cos 2\varphi) \quad (10)$$

$$F^i_{Ay} = -m_A a_{Ay} \quad (11)$$

Finally, for counterweight E we have:

$$\begin{aligned} x_E &= -h_E\sin\psi \\ y_E &= r_C\sin\varphi+(l+h_E)\cos\psi \end{aligned} \quad (12)$$

where, h_E is the length of a link AE. Considering Eq. 3 and 5 the acceleration of the counterweight E is obtained:

$$\begin{aligned} \ddot{x}_E &= a_{Ex} = \omega^2 h_E\lambda\sin\varphi \\ \ddot{y}_E &= a_{Ey} = -\omega^2\left(r_C\sin\varphi+\lambda^2(l+h_E)\cos 2\varphi\right) \end{aligned} \quad (13)$$

and inertia forces:

$$\begin{aligned} F^i_{Ex} &= -m_E a_{Ex} \\ F^i_{Ey} &= -m_E a_{Ey} \end{aligned} \quad (14)$$

Terms of zero equivalence in the system of forces can be written as:

- Projected on the axis Ox:

$$R_x+F^i_C\cos\varphi+F^i_{Bx}+N-F^i_D\cos\varphi+F^i_{Ex} = 0 \quad (15)$$

- Projected on the axis Oy:

$$R_y-G_A+F^i_{Ay}-G_B+F^i_{By}-G_C+F^i_C\sin\varphi-G_D-F^i_D\sin\varphi-G_E+F^i_{Ey} = 0 \quad (16)$$

- Sum of the torques relative to the point O:

$$-Ny_A-G_Bx_B-G_Cr_C\cos\varphi+G_Dh_D\cos\varphi-G_Ex_E+m_O(\bar{F}_B^i)+m_O(\bar{F}_E^i)+M=0 \quad (17)$$

- Moments of inertia F_B^i and F_E are calculated according to Varignon theorem:

$$m_O(\bar{F}) = xF_y-yF_x \quad (18)$$

The moments of inertia forces F_C^i and F_D^i relative to the point O are equal to zero. The equation for determining the moment of M is obtained by applying the principle of energy:

$$T-T_0=\sum A_i \quad (19)$$

Consider position of the mechanism corresponding to zero value of the angle [phi] as the initial. At this time, the instantaneous center of CE link speed is at infinity. Consequently, the speed of the chambers A, B, C and counterweight E at the initial time are equal to [φ]×r_C and the speed of the counterweight D in the same time is [φ]×h_D. Velocity modulus of chamber C and counterweight D are constant and equal to the initial values. Thus:

$$T-T_0=\frac{m_Av_A^2}{2}+\frac{m_Bv_B^2}{2}+\frac{m_Ev_E^2}{2}-(m_A+m_B+m_E)\frac{(\omega r_C)^2}{2} \quad (20)$$

Velocity of chambers A, B and counterweight E for an arbitrary angle can be found by differentiating the known relationships of their coordinates Eq. 6, 9 and 12 of the angle [phi], i.e:

$$\dot{y}_A = V_{Ay} = \omega r_C\left(\cos\varphi+\frac{\lambda}{2}\sin 2\varphi\right) \quad (21)$$

$$\dot{x}_B = V_{Bx} = -\omega\frac{r_C}{2}\sin\varphi$$
$$\dot{y}_B = V_{By} = \omega r_C\left(\cos\varphi+\frac{\lambda}{4}\sin 2\varphi\right)a \quad (22)$$

$$\dot{x}_E = V_{Ex} = -\omega h_E\lambda\cos\varphi,$$
$$\dot{y}_E = V_{Ey} = \omega\left(r_C\cos\varphi-\frac{\lambda^2}{2}(1+h_E)\sin 2\varphi\right) \quad (23)$$

Then, determine the work of forces. At the same time take into account, first that the reaction forces do not make research application point O of reaction forces R_x and R_y is fixed and the force N is perpendicular to the displacement of the slider; second, gravity force is a potential so its research can be calculated from the equation:

$$A_{GK} = -m_Kg(y_K-y_{K0}) \quad (24)$$

where, K denotes one of the bodies A, B, C, D or E, g = 9.81 m/sec^2 acceleration of gravity, y_K, y_{K0} the coordinates of the body at a certain angle [phi] = 0 and [phi] = 0, respectively:

Finally, the research of the torque applied to the link and providing rotation of the link with a uniform angular velocity [φ] is calculated by Eq. 25:

$$A_M = \int_0^\varphi M\,d\varphi \quad (25)$$

In view of the mentioned Eq. 19 can be written as follows form:

$$T-T_0 = \sum_{K=A,\dots,E} A_{GK}+\int_0^\varphi M\,d\varphi \quad (26)$$

Whence:

$$M=\frac{d}{d\varphi}\left(T-T_0-\sum_{K=A,\dots,E}A_{GK}\right)=\frac{dT}{d\varphi}-\frac{d}{d\varphi}\left(\sum_{K=A,\dots,E}A_{GK}\right) \quad (27)$$

The first term of the Eq. 27 can be transformed using the ratio:

$$\frac{d}{d\varphi}\frac{mv^2}{2}=mv\frac{dv}{d\varphi}=mv\frac{dv}{dt}\frac{dt}{d\varphi}=\frac{mv}{\omega}a_\tau=\frac{mv}{\omega}\frac{v_xa_x+v_ya_y}{v}=\frac{m}{\omega}\left(v_xa_x+v_ya_y\right) \quad (28)$$

where, a is acceleration so that:

$$\frac{dT}{d\varphi}=\frac{1}{\omega}\sum_{K=A,B,E}m_K\left(v_{Kx}a_{Kx}+v_{Ky}a_{Ky}\right) \quad (29)$$

The second term, taking into account transformations:

$$\frac{d}{d\varphi}mg(y-y_0)=mg\frac{dy}{d\varphi}=mg\frac{dy}{dt}\frac{dt}{d\varphi}=\frac{mg}{\omega}v_y \quad (30)$$

is written as follows:

$$-\frac{d}{d\varphi}\left(\sum_{K=A,\dots,E}A_{GK}\right)=\frac{g}{\omega}\sum_{K=A,\dots,E}m_Kv_{Ky} \quad (31)$$

Eq. 27 thus, takes the final form:

$$M = \frac{1}{\omega}\sum_{K=A,B,E} m_K\left(v_{Kx}a_{Kx}+v_{Ky}(a_{Ky}+g)\right)+g\cos\varphi\left(m_C r_C - m_D h_D\right) \quad (32)$$

Or putting it consistently in the expressions for the velocity and accelerations, finally obtain:

$$M = m_A Z_1 + m_B Z_2 + m_E Z_3 + m_C Z_4 + m_D Z_5 \quad (33)$$

Where:

$$Z_1 = r_C\left(\cos\varphi+\frac{\lambda}{2}\sin 2\varphi\right)\left(g-\omega^2 r_c\left(\sin\varphi-\lambda\cos 2\phi\right)\right)$$

$$Z_2 = \frac{\omega^2 r_C{}^2}{4}\sin\varphi\cos\varphi + r_C\left(\cos\varphi+\frac{\lambda}{4}\sin 2\varphi\right)\left(g-\omega^2 r_C\left(\sin\varphi-\frac{\lambda}{2}\cos 2\varphi\right)\right)$$

$$Z_3 = -\omega^2 h_E{}^2\sin\varphi\cos\varphi + \left(r_C\cos\varphi-\frac{\lambda^2}{2}(1+h_E)\sin 2\varphi\right)\left(g-\omega^2\left(r_C\sin\varphi+\lambda^2(1+h_E)\cos 2\varphi\right)\right)$$

$$Z_4 = r_C g\cos\varphi$$

$$Z_5 = -h_D g\cos\varphi$$

Having the expression for the torque Eq. 33 find an expression for the dynamic pressure in the slide N according to the expression Eq. 17 using Eq. 18:

$$N = \frac{M - G_B x_B - G_C r_C\cos\varphi + G_D h_D\cos\varphi - G_E x_E + x_B F^i_{By} - y_B F^i_{Bx} + x_E F^i_{Ey} - y_E F^i_{Ex}}{y_A} \quad (34)$$

Then, from Eq. 15 and 16 the value R_x and R_y can be determined, respectively:

$$R_x = F^i_D\cos\varphi - F^i_C\cos\varphi - F^i_{Bx} - N - F^i_{Ex},$$
$$R_y = G_A - F^i_{Ay} + G_B - F^i_{By} + G_C - F^i_C\sin\varphi + G_D + F^i_D\sin\varphi + G_E - F^i_{Ey} \quad (35)$$

As the expressions Eq. 33-35 the reactions N, R_x and R_y, being the functions of independent parameters φ, h_E, h_D are bulky nonlinear periodic functions and for simulation of the control systems, for example in MATLAB/SIMULINK Software is inconvenient to use a mathematical model of balancing in this way even when using programmable S-functions and using symbolic computation. While complicating the model, for example, taking into account centrifugal moments of inertia or the process of changing load factor, the adding of external forces it is necessary to compose new equations to rewrite S-functions, etc. Therefore, it is advisable to develop a computer model of CGMU in ADAMS Software which contains a Controls module, used to interface ADAMS with MATLAB/SIMULINK Software.

Simulation model: 3D view of the developed model of CGMU automatic balancing is shown in Fig. 2. The mechanism consists of the following solids in accordance with the design diagram (Fig. 1): CA, AE, OC, OD-(link); A, B, C, D, E-(ellipsoid) which connected by joints and imposed motions which description is shown in Table 1. Location of the counterweight E with shoulder hE' differs from that given in calculation scheme shown in Fig. 1 in which the shoulder of the counterweight is the link AE but the calculation obtained by the mathematical model

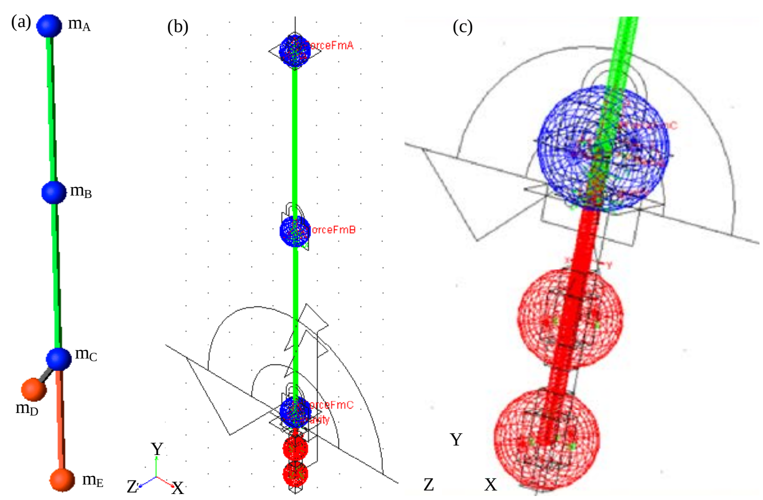

Fig. 2: Multibody computer model of CGMU; a) general 3D view; b) view with joints and motions and c) zoomed 3D view

Eq. 34-35 can find the values of responses for balancing by shoulder CE by reference shoulder in the opposite direction with respect to point C, taking into account the length of rod CA, i.e., for example, hE' = 20 cm corresponds hE = 90 cm at l = 70 cm.

Using parameterization in the model allows changing quickly aggregate values such as weight, length, units, etc.

However, the parameters of the masses cannot be changed during the simulation and to change the load factor for each marker's center of gravity of grinding chamber CM_marker GFORCEs are added which set change in mass. The format of these functions has the form FX = VARVAL (mass)×ACCX (CM_marker) where ACCX-function defining acceleration along the x-axis. VARVAL it is the function which used to exchange values parameters between Adams and Matlab/Simulink through the Controls module.

As a result, the input parameters of the subroutine defining the mechanical part in Simulink are: [φ], h_D, h_E, [Δ]m_A, [Δ]m_B, [Δ]m_C. Output parameters are dynamic reactions R_x, R_y, N and torque M. For another thing forces which are opposing repositioning of the counterweights and used to set the load effects in models of actuators can also be transferred to Simulink.

Mechanical part of the system is integrated into the getup of the model developed in Simulink where using the appropriate subroutines defined algorithms of changing the input parameters (Fig. 3).

Table 1: Description of joints and motions

Joints and motions	No. of joints and motions
Revolute joint	4
Translational joint	3
Fixed joint	3
Rotational motion	1
Translational motion	2

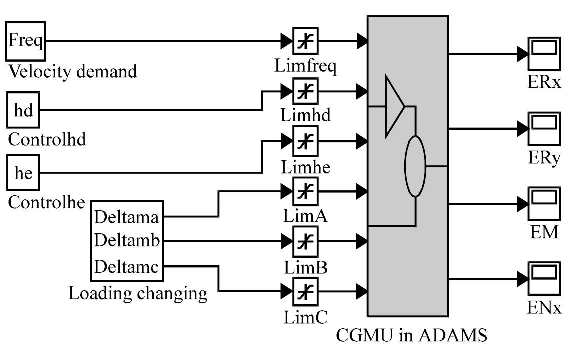

Fig. 3: Co-simulation model of CGMU with automatic balancing

RESULTS AND DISCUSSION

Numerical results: Investigated CGMU has the following parameters: $m_A = m_B = m_C = m_D = m_E = 15$ kg, $r_C = 0.02$ m, l = 0.7 m, AB = BC, [φ] = 40 rad sec^{-1} (382 rpm).

Bushuev *et al.* (2014) the optimality criterion was formulated as the minimum of RMS sum of squares of all reactions or which in formal language can be represented:

$$I = \sqrt{\frac{\int_0^{2\pi}\left[\begin{array}{l} k_{Rx}R_x^2(\varphi,h_D,h_E)+k_{Ry}R_y^2(\varphi,h_D,h_E)+ \\ k_N N^2(\varphi,h_D,h_E) \end{array}\right]^2 d\varphi}{2\pi}} \rightarrow \min_{h_D,h_E} / \begin{cases} 0 \le h_D \le 3[m] \\ -3 \le h_E \le 3[m] \end{cases} \quad (36)$$

where, $k_{Rx} = 10$, $k_{Ry} = 1$, $k_N = 10$ where the influence coefficients. And optimal positions of the counter weights: $h_D = 0.066$ m, $h_E' = 0.254$ m which are ensured the implementation of the above criterion were found.

For given values of counter weights positions graphics of changing the dynamic responses from the angle [φ] calculated by a mathematical model given in the form of Eq. 34-35 have the form shown in Fig. 4b.

Graph of changing torque, built on the basis of Eq. 33 is shown in Fig. 5b. Corresponding calculation of dynamic responses and torque using ADAMS computer model showed that the results (Fig. 4b and Fig. 5b) are identical to the results (taking into account initial phase shift [phi]), obtained with the mathematical model to the nearest millinewtons. It is noteworthy that this accuracy dependent of solver parameters and the number of steps and time of simulation ratio.

When moving the counterweight D with velocity equals to 1 (cm)/[simulation time] without a counter weight E there is a gradual changing of the dynamic reactions R_x and R_y (Fig. 6) and it is clear that their lows correspond to different positions of the counter weight.

If upper chamber fill a certain mass of grinding stuff one-time [Δ]m_A, the change in the dynamic responses R_x and R_y during to the grinding process at constant value of the counter weight position equals to $h_D = 3$ cm and without counter weight E will be as presented on the Fig. 7.

Summary: Thus, in the grinding process taken place the changing of dynamic bearings reactions of CGMU and hence to minimize these reactions it is necessary to carry out a new search using system of automatic balancing built in accordance with the methodology of the design described by Grigorevich (2013). It should be noted that

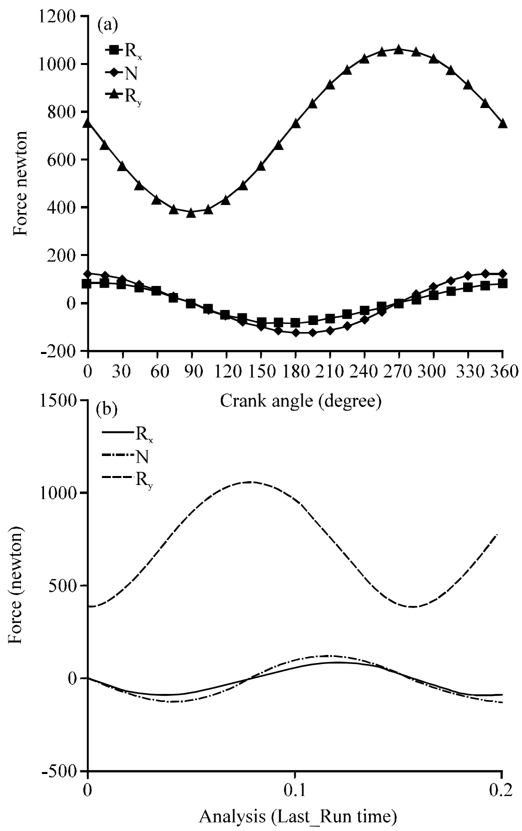

Fig. 4: Dynamic responses of CGMU balanced by two optimally located counterweights calculated by; a) mathematical model and b) computer model

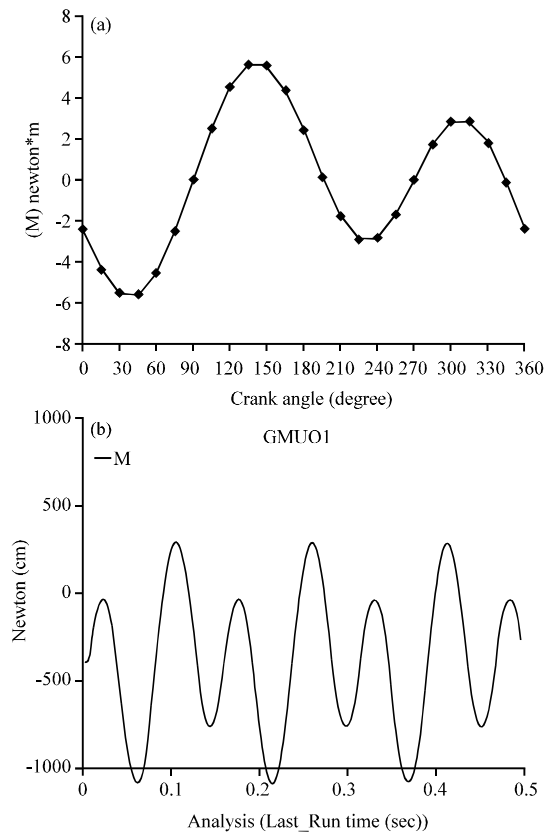

Fig. 5: The dependence of the torque M on the rotation angle [φ] calculated by; a) mathematical model and b) computer model

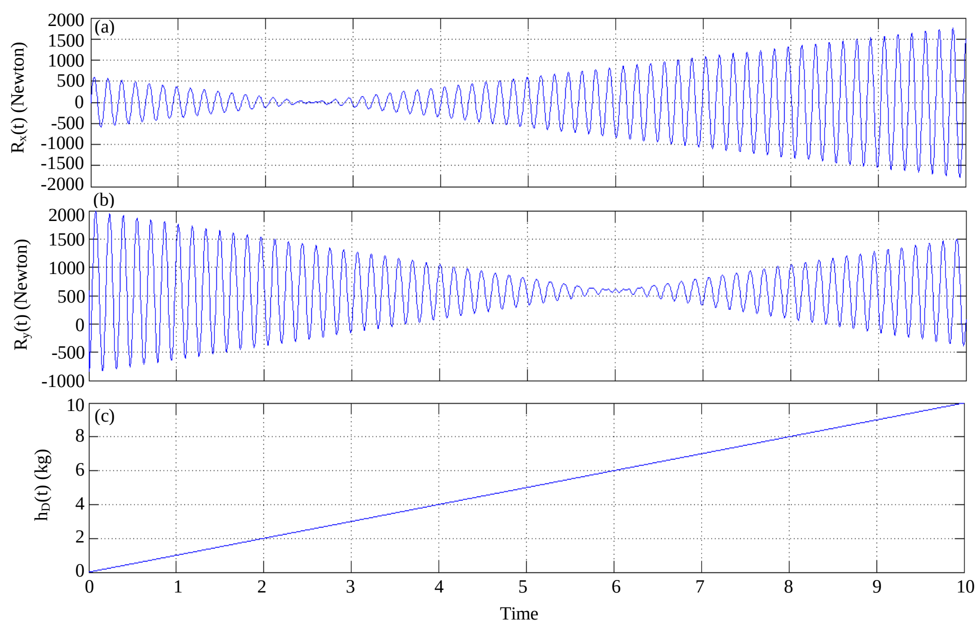

Fig. 6: Changing the dynamic responses caused by the movement of the counterweight D; a) $R_x(t)$ (Newton); b) $R_y(t)$ (Newton) and c) $h_D(t)$ (kg)

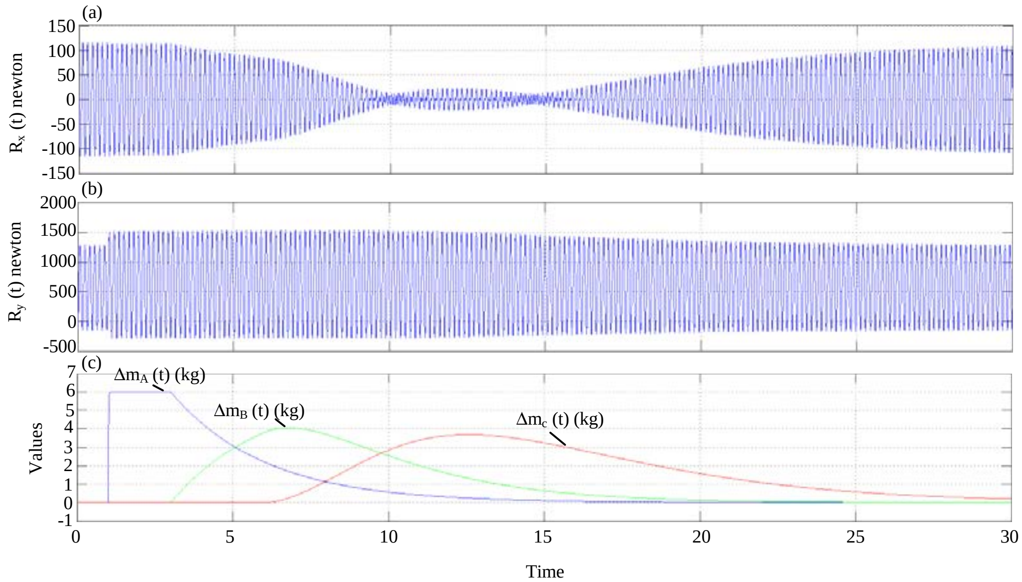

Fig. 7: Changing the dynamic responses due to the loading changing of the grinding chambers

the chosen optimization criterion Eq. 36 is used in this form because minima of dynamic responses and torque are achieved at different positions of the counter weights. And according to the results of research (Stativko and Rubanov, 2013) minimum points of vibration coincide approximately with the minimum points of the chosen criterion with given coefficients of influence.

CONCLUSION

Obtained through the use of d'Alembert principle, mathematical model CGMU with automatic balancing allows calculating the dynamic responses in guides and unit bearing, searching the optimal values of the provisions of counter weights but it is not convenient and flexible for programming, especially when simulating of the dynamics of the automatic balancing systems. As it was stated by means of calculation similar computer model developed in ADAMS software is identical to mathematical model but is more flexible and can be easily integrated into simulation software of control systems. In future, a variety of control algorithms under different operating regimes of CGMU can be tested using the developed computer model.

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