IMPROVEMENT OF OPERATING RELIABILITY OF VIBRATION-CENTRIFUGAL UNIT

V S Sevostyanov¹, V I Uralskij¹, A V Uralskij¹, L S Uralskaja¹, E V Sinitsa¹

¹Belgorod State Technological University named after V.G. Shoukhov, 46, Kostyukov str., Belgorod, 308012, Russia

E-mail: Alexx_1984.10@mail.ru

Abstract. The article describes the operational reliability of a vibration-centrifugal unit. The deformations and stresses of the most loaded structural elements are calculated, and a method of their minimization with the aim of ensuring the operational reliability of the unit is established.

1. Introduction

One of the perspective ways to increase the efficiency of the grinding process can be the combination in one machine of the stages of coarse, thin and ultrafine grinding. Such combination can be determined not only by the shape and size of grinding bodies, but also by different trajectories of the motion of the chambers to provide appropriate modes of operation: for coarse grinding - intensive impact loading and partial abrasion; for fine grinding - impact load with increasing degree of abrasion; for ultrafine grinding - intensive abrasion.

A distinctive feature of the vibration-centrifugal grinding unit is the provision of the possibility of changing the regime of the dynamic action of grinding bodies on the crushed material, namely, the combination of impact and abrasive loads by providing the appropriate trajectories of grinding chamber motion [1, 2, 8,10].

During the operation of the centrifugal grinding unit (CGU), as well as the results of calculations, it was found that the most loaded working elements are the upper grinding chamber, which performs translational motion in the vertical plane, and the mobile frame of the grinding block.

2. Calculation of stresses and deformations in the body of the grinding chamber

Let’s consider the upper and lower chambers in which grinding bodies perform various movements.

In the upper chamber grinding bodies move reciprocating along the vertical axis with periodic impacts onto the inner cylindrical surface of the chamber. In the lower chamber, grinding bodies roll along the inner cylindrical surface. The values of the forces acting on the grinding load on the chamber walls based on the results of the dynamic analysis of the experimental CGU sample were determined [3]. The maximum value of the shock force in the upper chamber is \( F_v = 4800 \, \text{N} \) with a mass of grinding bodies in each chamber equal to 12 kg, in the lower chamber the maximum value of the centrifugal force is \( F_n = 153,6 \, \text{N} \).

From a comparison of the obtained results, it is obvious that the upper grinding chambers are subjected to the greatest dynamic action of the grinding bodies. Consequently, the strength calculations must be performed first for these chambers.
The appearance of plastic deformations in the form of a "barrel-shaped" surface is established in the upper chamber.

Thus, it is necessary to calculate the deformations of these structural elements and to establish methods for minimizing them in order to ensure the operational reliability of the unit.

The body of the grinding chamber is a thin-walled cylinder because the thickness of its wall is less than 1/10 of the average radius of the cylinder. For calculating thin-walled cylinders, it is assumed that in the circumferential direction the stresses are constant along the wall thickness, and there is no stress in the radial direction [4, 9].

The cylindrical body of the chamber with an inner \( r_1 \) (m) and outer \( r_2 \) (m) radii is under the action of an internal load of grinding bodies \( q \) (Pa) distributed over the contact surface [5] (Fig. 1).

![Fig. 1. The scheme of loading from grinding bodies](image1)

We consider an element abcd separated in a ring formed by two sections perpendicular to the axis of the cylinder and located at a distance equal to one to determine the stresses and deformations (Fig. 3).

![Fig. 2. Stresses in element of a cylindrical surface](image2)

In case of an internal distributed load acting on the cylinder according to [6, 7], the circumferential stresses \( \sigma_\Theta \) (Pa) of displacement \( u \) (m) of the cylindrical surface of radius \( r \) (m) are determined by the formulas:

\[
\sigma_\Theta = \frac{r_1^2}{r_2^2 - r_1^2} \left( 1 + \frac{r_2^2}{r^2} \right) q;
\]

(1)

\[
u = \frac{1 - \mu}{E} \cdot \frac{r_1^2}{r_2^2 - r_1^2} q r + \frac{1 + \mu}{E} \cdot \frac{r_1^2 r_2^2 q}{r_2^2 - r_1^2} \cdot \frac{1}{r},
\]

(2)

where \( E \) – modulus of elasticity, Pa; \( \mu \) – Poisson’s ratio.

The greatest value of the circumferential stress will be at the inner surface when \( r = r_1 \):
\[
\sigma_{\theta(\eta)} = \frac{1 + \delta^2}{1 - \delta^2} q.
\]  

(3)

Radial displacement at the inner surface:

\[
u_{(\eta)} = \frac{r_1}{E} \left( \frac{1 + \delta^2}{1 - \delta^2} + \mu \right) q.
\]  

(4)

where \( \delta = \frac{r_1}{r_2} \).

(5)

The presented dependences are valid for a static load action. In fact, the camera body experiences a cyclic impact load.

Stresses \( \sigma_D \) and deformations \( f_D \) arising at a bending impact are determined in accordance to the expressions:

\[
\sigma_D = k_D \sigma_{st}; \quad (5) \quad f_D = k_D f_{st},
\]  

(6)

where \( \sigma_{st} \) – stress of a static load action, Pa; \( f_{st} \) – deformation of a static load action, m; \( k_D \) – dynamic coefficient, defined by formula:

\[
k_D = 1 + \sqrt{1 + \frac{2H}{f_{st}}},
\]  

(7)

where \( H \) – height of load drop, m.

In this case, the kinetic energy of impact of the grinding bodies is equal to the potential energy of the incident load, i.e.:

\[
M_1 g H = M_1 \frac{V_{S1}^2}{2}.
\]  

(8)

Then:

\[
H = \frac{V_{S1}^2}{2g}.
\]  

(9)

The value \( f_{st} \) corresponds to the maximum radial displacement of the inner surface of the cylinder \( u \).

Taking into account (4) and (9), we obtain an expression for determining the dynamic coefficient:

\[
k_D = 1 + \frac{E V_{S1}^2}{q r_1 g \left( 1 + \frac{1 + \delta^2}{1 - \delta^2 + \mu} \right)}.
\]  

(10)

Calculations of stresses and deformations for concrete conditions of the grinding chamber of the experimental CGU sample are performed using the professional finite element analysis complex ANSYS, which allows solving the structural strength problems.

The results are shown in Figures 3 and 4.

After making changes in the design of the body of the grinding chambers of the experimental unit sample, which is made of 40X steel (GOST 4543 -71), the inner diameter of the chamber is \( d_1 = 0.15 \) m, the outer diameter is \( d_2 = 0.16 \) m.

Based on the results of calculations, the maximum stresses \( \sigma_D = 5.7 \) MPa and deformation \( f_D = 3.1 \times 10^{-3} \) mm were obtained.
The probability of failure-free operation of the camera body by the strength criterion can be determined depending on the quantile found by the formula [6]:

\[ u_P = -\frac{\bar{\sigma}_{\text{lim}} - \bar{\sigma}}{\sqrt{S^2_{\text{lim}} + S^2_{\sigma}}}, \]

(11)

where \( \bar{\sigma} = \bar{\sigma}_0 \) – average value of stress, MPa; \( \bar{\sigma}_{\text{lim}} = \bar{\sigma}_B \) – average value of the tensile strength of the body material, MPa; \( S_{\sigma} \) – standard deviation of the value \( \bar{\sigma} \), MPa; \( S_{\text{lim}} \) – standard deviation of the value \( \bar{\sigma}_{\text{lim}} \), MPa.

Performing series of calculations varying the load on the body of the camera and taking into account possible deviations in the strength of the material depending on the heat treatment, we obtain the following values of the quantities entering into the expression (11):

\[ \bar{\sigma} = 5,04 \text{ MPa}; \quad \bar{\sigma}_{\text{lim}} = 591 \text{ MPa}; \quad S_{\sigma} = 0,167 \text{ MPa}; \quad S_{\text{lim}} = 94,0 \text{ MPa}. \]

As a result, \( u_P = -6.23 \), and in accordance with the recommendations [6], we have a probability of failure-free operation by the criterion of strength \( P = 1 \) with a large margin, because \( P = 1 \) already occurs when \( u_P = -3.9 \). During the experimental research and pilot operation of the grinding unit, the control measurements of the diameters of the upper chamber were carried out, as a result of which it was established that there were no deformations.

3. **Calculation of stresses and deformations in the mobile frame of the grinding unit**

Movable frames are integral parts of the grinding unit design and are affected by both static and dynamic loads caused by the forces of gravity of the structure and the grinding load, centrifugal forces. The main load falls on the elements of the lower hinges. According to the results of the kinetostatic analysis of the lever mechanism, it is revealed that the reactions in these hinges take the maximum values at the angle of rotation of the eccentric shaft of the grinding block, equal to \( \varphi_0 = 270^\circ \). The scheme of the forces in this position is shown in Fig. 5.

The movable frame is a welded structure of channel bars No. 8, No. 10. The grinding chambers are fixed to the frame using bolted connections. As a result of the forces in the chambers, the load is transferred to the frame in the places where the chambers are fixed to the frame.

We obtain the design scheme of the mobile frame, shown in Fig. 6.
In the scheme, the points \( m' \) and \( m'' \), \( k' \) and \( k'' \), \( f' \) and \( f'' \) are the attachment points for the upper, middle and lower chambers, respectively.

Transverse forces and moments act at points \( m' \) and \( m'' \):

\[
F_{m'}^y = F_{m''}^y = \frac{ql}{2}; \tag{12}
\]

\[
M_{m'}^{zy} = M_{m''}^{zy} = \frac{ql^2}{12}, \tag{13}
\]

where \( ql \) - distributed load in the upper chamber, Pa; \( l \) – the length of the chamber, m.

Fig. 5. The scheme of the forces in the mobile frame

Longitudinal forces: \( F_{m'}^z = F_{m''}^z = 0 \).

At points \( k' \) and \( k'' \), the transverse forces and moments act:

\[
F_{k'}^z = F_{k''}^z = \frac{ql}{2}; \tag{14}
\]

\[
M_{k'}^{z\alpha} = M_{k''}^{z\alpha} = \frac{ql^2}{12}, \tag{15}
\]

where \( q_2 \) – distributed load in the middle chamber, Pa;

Longitudinal forces: \( F_{k'}^z = F_{k''}^z = 0 \).

At points \( f' \) and \( f'' \), transverse forces and moments act:

\[
R_{f'}^z = R_{f''}^z = \frac{q_3l}{2}; \tag{16}
\]

\[
M_{f'}^{z\alpha} = M_{f''}^{z\alpha} = \frac{q_3l^2}{12}, \tag{17}
\]

where \( q_3 \) – distributed load in the lower chamber, Pa;

Longitudinal forces: \( F_{f'}^z = F_{f''}^z = 0 \).
At the points $C'$, $C''$, and $B'$, $B''$ there are reactions $R_{C'}$, $R_{C''}$ and $R_{B'}$, $R_{B''}$, determined with kinetostatic analysis of the lever mechanism.

Calculations of stresses and deformations for specific loading conditions of the experimental CGU sample were performed using the ANSYS computational complex.

The results obtained are presented in Figures 7 and 8.

4. Conclusion
Based on the obtained results and with the aim of creating a rational design of the grinding chamber, it is advisable to use a less durable material, for example, Steel 10 (GOST 1050-88), for which $\sigma_B = 340$ MPa. It is also possible to reduce the thickness of the shell wall to 3.5 mm, provided it is made of Steel 30 (GOST 1050-88), for which $\sigma_B = 500$ MPa.

Based on the results of calculations of the design of the mobile frame, $\sigma_{D,\text{max}} = 19.7$ MPa; $f_{D,\text{max}} = 0.16$ mm, which is much less than the permissible values. This indicates a significant safety margin of the designed structure.

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References
[1] Gridchin A.M., Sevostyanov V S, Lesovik V S, Uralskii V I, Uralskii A V, Sinitsa E V 2006 The grinding-mixing plant applicant and patent owner of LLC “TC RECYCLE” 2005118705/03 Russian Federation (17) 8.
[2] Gridchin A M, Sevostyanov V S, Lesovik V S, Uralskii V I, Uralskii A V, Sinitsa E V 2010 The grinding-mixing plant applicant and patent holder of the BSTU. V G Shukhova (LLC "TC RECYCLE" 2008109444/03 Russian Federation) 5 (11).
[3] Reshetov D N, Ivanov A S, Fadeev V V, Reshetova D N 1988 Reliability of machines: A manual for engineering specialties of universities (Moskow: High education. School) 238.
[4] Sevostyanov V S, Uralsky V I, Sinitsa E V, Uralsky A V 2008 Questions of dynamic research of a centrifugal grinding-mixing plant Vibration machines and technologies: Collection of scientific. tr.
(Kursk gos.tehn. univ-t. Kursk) 596-601.

[5] Sinitsa E V, Uralsky A V, Pletnev A V 2007 Influence of grinding loading on the dynamics of a centrifugal grinding Scientific researches, nanosystems and resource-saving technologies in the construction industry: coll. reports of the International Scientific and Practical Conference Belgorod: (Publishing house BSTU. V G Shukhova) 188-192.

[6] Pisarenko G S 1986 Resistance of materials Ed. acad. AS USSR - 5th ed. (K.: High school. Head Publishing House) 775.

[7] Bot G U 1966 Some problems of vibration grinding Proceedings of the European Conference on Chopping (Moscow: Stroyizdat) 435-443.

[8] Uralsky A V, Sevostyanov V S 2010 Multifunctional centrifugal unit with parallel grinding blocks Herald of Belgorod State Technological University named after V.G. Shukhov 1 106-112.

[9] Rose H E and Sullivan R M 1961 Vibration Mills and Vibrating Milling (London) 195.

[10] Uralsky V I, Sevostyanov V S, Sinitsa E V Multifunctional centrifugal grinding unit "IOP Conference Series: "Materials Science and Engineering "(MSE), International Scientific and Technical Conference "Modern Problems of Mechanical Engineering ".

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