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MAIN PROBLEMS OF MINE DRUM DESIGN

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Abstract. *The article deals with the problems of mine drums, namely, an adjusted approach to the justification of their design, and suggests ways to improve their performance. The practice of designing mine drums is based on the introduction of additional stiffening elements, such as ribs, rings and scarves. This significantly increases the weight and complexity of the drum manufacture, and also leads to local stresses in the welded areas. This greatly complicates the operation of the structure and leads to the destruction of welds and the appearance of cracks. Reinforcing the welds does not improve the situation, but rather reduces the durability of the drum shell. Designers of mine hoisting machines do not have a theoretical basis for refusing to install rings and stiffeners.*

The calculations of the strength of the drum head using the simplified and refined methods showed that the strength of the head is fully ensured and there is no need to install additional structures in the form of scarves.

Key words: *mine drum, frontal liner, shell, strength, bending moment, stress, scarf, stiffening elements.*

Introduction

Mine drums are the main element of a hoisting machine, so ensuring their reliable operation is quite important. In addition, mine drums have a rather high metal consumption, which is not always justified. To increase the structural rigidity, mine drums are reinforced with rings and stiffeners, which not only increase their weight and complicate the manufacturing process, but also cause significant local stresses that impair the drum's performance.

If the drum shell thickness determined by the strength calculation does not provide sufficient stability, either increase the thickness or reinforce the shell with ribs or stiffening rings.

The first way leads to an increase in the weight of the drum, and the second, if not significantly increasing the weight, worsens the drum manufacturing technology and its performance.

Analysis of the latest research and publications

The problems of mine drums have been studied by many prominent scientists, such as B. Davydov, B. Kovalsky, Z. Fedorova, K. Zabolotny, O. Grigorov [1], V. Malinovsky [2], W. Woernle [3], H. Ernst [4] and others.

B. Kovalsky drew attention to the harmfulness of installing stiffening elements in the rope drum shell and the need to justify those cases when they are necessary.

Cracks in the shell are usually progressive and significantly reduce the strength of the entire drum.

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Considerations of a possible reduction in drum weight when stiffeners are installed are often illusory, as stiffeners generate bending stresses under conditions of sufficiently high concentration. These points are not taken into account by designers, despite the fact that fatigue cracks near stiffeners appear very often, as evidenced by studies of the operation of drums of mine hoisting machines. For example, O. Panasyants notes that excessive finning of the mine drum shell causes premature failure of the structure.

A. Manevich and A. Buchuch considered the stress state of the shells and diaphragms of mine hoists on the example of the mine hoist ШПМ БЦК 8/5x2.7, which is designed to operate at a depth of up to 1200 m. It was noted that cracks appeared in the welds of the headstays and ribs. The cracks were repaired by removing the welded metal and re-welding.

The frequency of repairs was 2...3 months. During an unscheduled shutdown of the machine, it was noted that cracks in the repair seams resumed 10...14 days after the repair and did not develop.

In subsequent repairs, in addition to welds, the diaphragms were reinforced with one-sided linings in the areas of cracking. However, cracks continued to develop in the repair seams and in the base metal of the linings and diaphragms along the edge of the lining. In the overhaul period (2...3 months), one of the cracks in the diaphragm developed over the entire width of the ring with access to the base metal of the shell.

Repeated repairs of cracks and strengthening of the cracked areas in the diaphragm rings did not stop the cracking process.

The authors determined the stress-strain state of the sheath and diaphragms caused by the winding of a loaded rope. The proposed calculation methodology took into account the weakening of previously wound rope turns due to sheath compression during the winding of subsequent turns.

Calculations carried out by A. Manevich and A. Buchuch showed that the stiffness of the diaphragm with respect to cyclic loading and its plane is much higher than the stiffness of the shell. Therefore, the local load close to the junction of the shell with the diaphragm is almost completely absorbed by the diaphragm. It was noted that a more precise determination of the length in the longitudinal direction of the zone within which the cyclic or local load is perceived by the diaphragm is a rather difficult task.

Problem statement

In mine drums, not only the drum shell, but also the lining is loaded. To increase their stiffness, they are reinforced with braids, which not only increases the weight of the drum itself, but also significantly complicates the technology. The most harmful consequence of this design is that, due to the large number of welds, a significant amount of local stresses arises in the material of the lining. At the same time, the strength of the lining decreases, i.e. we have the opposite effect.

In the calculations, the frontal plate is assumed to be a smooth circular plate that is clamped on the outer surface of the hub at a radius *r*. The condition of compatibility of deformations is $y_c = y_l, \theta_c = \theta_l$, where $y_c, y_l, \theta_c, \theta_l$ are the radial displacements and angles of rotation of the drum wall and the edge of the frontal plate, respectively.

Presentation of the main material

Radial displacements and rotation angles occur at the junction of the casing and the header, which depend on the load of the rope being wound, the size of the drum and the distance a between the header and the location of the first turn. The larger this distance is, the lower the bending moment and transverse force values that occur in the head.

We will study the mine drum of the ЦР – 6x3.4/0.6 hoisting machine (Fig. 1).

The stress in welds can be calculated using the formula [5]:

$$\sigma=\frac{M_x}{W}+\frac{Q_x}{F}=\frac{6M_x}{c^2}+\frac{Q_x}{c}, \tag{1}$$

where M_x is the bending moment in the weld zone; Q_x – transverse load; c – is the suture catheter.

Let's consider the case of loading the drum shell in the ring area We perform a more detailed calculation of the drum shell, which is supported by stiffening rings.

Let us consider a rope drum supported by stiffening rings as a mixed variational system whose potential energy is a functional with additional terms.

$$\mathrm{U}=\int_{x_0}^{x_1} \Gamma\left(x,f,f^{`},f^{``}\right)dx+\eta_1\left(x_0,f_0,f_0^{`},f_0^{``}\right)+ +\eta_2\left(x_1,f_1,f_1^{`},f_1^{``}\right),$$

where η_1 is the potential energy of the frontal area; η_2 – potential energy of the stiffening ring.

$$\eta_0=\oint \frac{EJ_{01}}{2D_h^2}Rf^2(x)\cos^2 n\varphi = \frac{EJ_{01}R\pi}{2D_h^2}f^2(x), \tag{2}$$

$$\eta_1=\oint \frac{EJ_{0k}}{2D_h^2}Rf^2(x)\cos^2 n\varphi = \frac{EJ_{01}R\pi}{2D_h^2}f^2(x), \tag{3}$$

where EJ_{01}, EJ_{0k} is the bending stiffness of the frontal plate and the stiffening ring, respectively.

The natural boundary conditions for solving the mixed variational problem will be as follows

$$\left[\frac{d\Gamma}{df^{`}}-\frac{d}{dx}\left(\frac{d\Gamma}{df^{``}}\right)+\frac{d\eta_0}{df}\right]_{x=x_0}, \tag{4}$$

$$\left[\frac{d\Gamma}{df}-\frac{d}{dx}\left(\frac{d\Gamma}{df'}\right)+\frac{d\eta_2}{df}\right]_{x=x_1}. \qquad (5)$$

Then we get a system of two equations:

$$\begin{cases}\frac{D(n^2-1)(2-v)}{R}\frac{df}{dx}-\frac{DR}{2}\frac{d^3f}{dx^3}+\frac{12(1-v^2)J_{0I}R}{\delta^3}f(x)\\ \frac{D(n^2-1)(2-v)}{R}\frac{df}{dx}-\frac{DR}{2}\frac{d^3f}{dx^3}+\frac{12(1-v^2)J_{0k}R}{\delta^3}f(x)\end{cases}.$$

The solution of this system allows us to determine the coefficients C_1 and C_2 of the equation:

$$C_1=\frac{Ae^{-2\rho cos\psi I}e^{-2\pi k\mu\frac{L-I}{h}}\left\{\begin{matrix}2D^2\pi k\mu\times\\ \times\left[R^2\pi k\mu-(n^2-1)(2-v)\right]-\\ -EJ_{0I}R^2h\end{matrix}\right\}}{EJ_{0I}R^2h(1+e^{-2\rho cos\psi l})}$$

$$-\frac{A\pi k\mu e^{-2\rho cos\psi I}e^{-2\pi k\mu\frac{L-I}{h}}}{\rho\psi he^{\rho cos\psi l}(1+e^{-2\rho cos\psi l})sin\,\psi lcos(\rho sin\,\psi l)}\times$$

$$\times\frac{(n^2-1)(2-v)h-\pi k\mu R^2}{\frac{(n^2-1)(2-v)}{R}-\frac{\psi^2R}{2}\left[2\rho\cos\begin{matrix}(\rho sin\psi l)+\\ \rho^2cos2\psi l\end{matrix}\right]\times\\ \times\frac{12(1-v^2)J_{0k}R}{\rho\psi\delta^3sin\psi l}}$$

$$-\frac{2\pi k\mu e^{-2\pi k\mu\frac{L-l}{h}}}{\rho\psi he^{\rho cos\psi l}sin\psi lcos(\rho sin\psi l)(1+e^{-2\rho cos\psi l})}\times$$

$$\times\frac{(n^2-1)(2-v)-\pi k\mu R^2}{\frac{(n^2-1)(2-v)}{R}-\frac{\psi^2R}{2}\left[\begin{matrix}2\rho\cos(\rho cos\psi l)\\ +1+\rho^2cos2\psi l\end{matrix}\right]\times\\ \times\frac{12(1-v^2)J_{0k}R}{\rho\psi\delta^2sin\psi l}}.$$

The annular normal stresses in the shell are determined by the formula:

$$\sigma_\varphi=\frac{w(x)}{R}E, \qquad (6)$$

where $w(x)$ is the radial displacement determined by the formula:

$$w(x)=f(x)cosn\varphi, \qquad (7)$$

Substitute the values of the bending moment and transverse load into formula (1), which are determined by the following formulas:

Fig. 1. Basic design of the drum

The bending stress in the frontal area can be determined by a simplified method [2].

$$\sigma_u=\frac{6M_u}{\delta^2}=\frac{3p(x)}{\beta^2\delta^2}\frac{1}{1+\varphi}, \qquad (9)$$

where

$$\beta=\frac{1,285}{\sqrt{R\delta}},\varphi=2,6\sqrt{\frac{R}{\delta}}\left(\frac{\delta}{\delta_1}\right)^2\frac{1-\frac{r^2}{R^2}}{1+v-(1-v)\frac{r^2}{R^2}}.$$

Using these formulas, we determine the value of the bending moment in the frontal area $\sigma_u=115$ MPa for the drum of the hoisting machine ЦР – 6x3.4/0.6, which is significantly less than the permissible value for steel St. 3 $[\sigma_u]=160$ MPa.

Determine the bending stress according to the revised method [3].

$$M_m=EJ_0\chi_\varphi=\frac{EJ_0}{D_m}f(x)cosn\varphi, \qquad (10)$$

where EJ_0 is the bending stiffness of the frontal plate.

$$D_m=Ei_m,$$

$$f(x)=cos(\rho sin\psi k)\left(C_1e^{\rho cos\psi k}\right)+Ae^{-k\mu\frac{1-x}{h}2\pi},$$

$$C_1=\frac{2+v(n^2-3)-\frac{4\pi^2k^2\mu^2R^2}{h^2}}{J_0R}$$

$$\times\left[e^{-\rho}-\frac{cos\psi L(1+\rho cos\psi L)}{e^{-\rho cos\psi L}\left(cos\psi L+\rho cos2\psi L\right)-e^{\rho(2-cos\psi L)}}\right]+$$

$$+\frac{4\pi^2k^2\mu^2Ae^{-k\mu\frac{L-i}{h}2\pi}}{h^2\rho\psi^2cos\boxed{f_0}(\rho sin\psi L)\begin{bmatrix}e^{\rho cos\psi L}\left(cos\psi L+\rho cos2\psi L\right)-\\-e^{\rho(2cos\psi L)}(cos2\psi L-cos\psi L)\end{bmatrix}}$$

$$C_2=\frac{\begin{bmatrix}\dfrac{4\pi R^2k^2\mu^2}{h^2}\\+2-v\left(n^2-3\right)\end{bmatrix}i_m^2}{J_0R}\begin{bmatrix}\dfrac{cos\psi L+\rho cos2\psi L}{e^{-\rho(2-cos\psi L)}}\\(cos\psi L+\rho cos2\psi L)\end{bmatrix}.$$

The calculation using this method yields a bending stress value of MPa [4].

As we can see, the strength of the headwall is fully ensured without the installation of reinforcing elements. We believe it is appropriate to propose a new design of the mine drum (Fig. 2) [5].

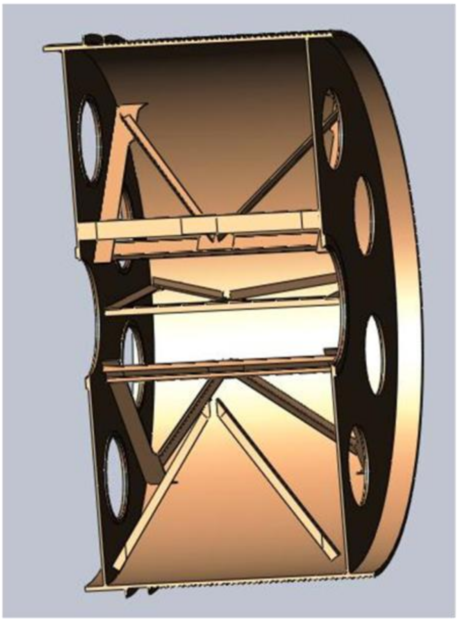

Fig. 2. Modernised design

Conclusions

The calculations of a particular mine drum allowed us to conclude that it is possible to change the design to reduce the weight of the drum and the cost of its manufacture, as well as to improve its reliability.

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Основні проблеми конструкції шахтних барабанів

Анотація. Проблема. *У статті розглянуто питання шахтних барабанів, а саме скоректований підхід до обґрунтування їх конструкції, запропоновано шляхи покращення їх експлуатаційних властивостей. Шахтні барабани забезпечують надійну роботу всієї підйомної машини й нерідко безпеку самих людей, тому тема, яка розглядається в статті, дуже актуальна.* ***Мета.*** *Практика конструювання шахтних барабанів ґрунтується на застосуванні елементної жорсткості, таких як ребра, кільця і косинки. Це значно збільшує вагу та складність виготовлення барабана, а також призводить до появи місцевих напружень у точках приварювання. У шахтних барабанах навантаженими є не тільки обичайка барабана, але й лобовини.*

Методика. *Для підсилення їх жорсткості вони посилюються косинками, що не тільки збільшує вагу самого барабана, але значно ускладнює технологію. У цьому разі міцність лобовин зменшується, тобто маємо зворотний ефект. Це значно ускладнює роботу конструкції та спричиняє руйнування зварювальних швів із виникнення тріщин. Підсилення зварювальних швів не покращує ситуацію, а навпаки, зменшує довговічність обичайки шахтного барабана.* ***Результати.*** *Проведені розрахунки міцності лобовини за спрощеною та уточненою методикою показали, що міцність лобовини повністю забезпечена й необхідності установки додаткової конструкції у вигляді косинок не потрібна.* ***Оригінальність.*** *У дослідженні враховано додаткові фактори, що в попередніх розрахунках не розглядалися, такі, наприклад, як жорсткість самих лобовин і коефіцієнт тертя між канатом і барабаном.* ***Практична цінність.*** *Запропоноване в статті застосування нової методики розрахунків міцності оболонки шахтного канатного барабана дає змогу значно покращити роботу шахтних підйомних машин, зменшити кількість ремонтів і забезпечити їх довговічність. Крім цього, металоємність барабанів значно зменшується, що дозволяє отримати значну економію їх вартості та забезпечити надійну роботу всього процесу шахтного підйому.*

Ключові слова: *шахтний барабан, лобовина, обичайка, міцність, момент згину.*

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