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MODELING OF WATER REGULATION IN WELLS WITH WATERED PRODUCTION IN HETEROGENEOUS OIL FORMATIONS

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ABSTRACT

At a certain stage of oil and gas production, by increasing the efficiency of regulating water volumes in wells with water-cut production, it is possible to reduce water cut and increase current oil recovery. When developing oil fields, in order to maintain reservoir pressure and increase the oil recovery factor, water or various additives are pumped into the objects.

During some stages of oil field development, various types of water (including water injected into the reservoir and external water filtered from the top) enter highly permeable areas and reach the production wells, and the amount of water in their products increases. In this regard, the main problem is the isolation of water entering the well by maintaining oil production.

The presented work describes the development of nanostructured gels that easily penetrate into the water-saturated depth of the formation in flooded wells, effectively blocking culverts, without changing the phase permeability of oil, mathematical modeling of the process, and analysis of the results.

QEYRİ-BIRCİNS NEFT LAYLARINDA MƏHSULU SULAŞMIŞ QUYULARDA SUYUN TƏNZİMLƏNMƏSİNİN MODELLƏŞDİRİLMƏSİ

XÜLASƏ

Neftqazçıxarmanın müəyyən mərhələsində məhsulu sulaşmış quyuların suyunun tənzimlənmə effektivliyini artırmaqla, quyu məhsulunun sulaşmasının aşağı salınması və cari neftvermənin artırılması mümkündür. Neft yataqlarının işlənməsi zamanı lay təzyiqinin saxlanılması və layların neftverməəmsalının artırılması məqsədi ilə obyektlərə su və ya onun müxtəlif əlavələri vurulur.

Neft yataqlarin işlənməsinin bəzi mərhələsində müxtəlif tip sular (o cümlədən, laya vurulan su və yuxari hissədən süzülən kənar su) yüksək keçiricilikli sahələrə daxil olaraq istismar quyularına çatır və onun məhsulunda suyun miqdarı artır. Bununla əlaqədar olaraq neftə görə hasilatın saxlanması şərti hesabına quyuya daxil olan suyun təcrid edilməsi əsas problem kimi mühüm aktuallıq kəsb edir.

Təqdim olunan işdə məhsulu sulaşmış quyularda layın su ilə doymuş dərinliyinə asanlıqla daxil olan, su buraxan kanalları effektiv şəkildə bağlayan, neftə görə faza keçiriciliyini dəyişməyən nanostrukturlu gellərin işlənməsi və prosesin riyazi modelləşdirilməsi, nəticələrin analizi verilmişdir.

Açar sözlər: *metal nanohissəcik, maye şüşə, riyazi model, təcrid, təzyiq, doyma, konsentrasiya*

МОДЕЛИРОВАНИЕ РЕГУЛИРОВАНИЯ ВОДЫ В СКВАЖИНАХ С ОБВОДНЕННОЙПРОДУКЦИЕЙ В НЕОДНОРОДНЫХ НЕФТЯНЫХ ПЛАСТАХ

РЕЗЮМЕ

На определенном этапе добычи нефти и газа,повышая эффективность регулирования воды в скважинах собводненной продукцией,можно добиться снижения обводненности и увеличения текущей нефтеотдачи. При разработке нефтяных месторождений с целью поддержания пластового давления и повышения коэффициента нефтеотдачи пластов в объекты закачивают воду или различные ее добавки.

В некоторых этапах разработки нефтяных месторождений различные виды вод (в том числе вода, закачиваемая в пласт, и внешняя вода, отфильтрованная из верхней части) попадают в высокопроницаемые участки и достигают добывающих скважин, а количество воды в их продуктах увеличивается. В связи с этим основной проблемой является изоляция входящей в скважину воды за счет поддержания добычи по нефти.

В представленной работе даются разработка наноструктурированных гелей, легко проникающих в водонасыщению глубину пласта в обводненных скважинах, эффективно перекрывающих водопропускные каналы, не изменяющих фазовую проницаемость по нефти, математическое моделирование процесса, анализ результатов.

Ключевые слова: *наночастица металла, жидкое стекло, математическая модель, коэффициент изоляции, давление, насыщение, концентрация.*

INTRODUCTION

One of the most important issues of the modern stage of development of the oil and gas extraction industry is the issue of increasing oil and gas production. By increasing the oil yield in non-homogeneous formations, the rapid advance of injected water occurs and causes dilution of the extracted products.

There are a number of methods for preventing premature waterlogging of production wells: physico-chemical methods of creating a barrier to prevent water ingress; hydrodynamic methods calculated to create minimal impact on the well's optimal operating mode and the wetted parts of the productive horizon.

In this regard, the problem of limiting water flow is highlighted. To solve this problem, tamponade solutions, sediment forming solutions, gels, wetting modifiers, etc. There are quite a number of methods that allow solving using [1,2].

For this, various tamponage (adhesive) materials, including emulsion composition, polymer systems, cement mixtures, and gel-forming compounds are introduced into the wetted zone of oil wells.

A comprehensive analysis of the mentioned technologies was carried out, their capabilities were evaluated and their shortcomings were clarified. It was determined that one of the most promising methods of selective isolation of water flow is isolation with a composition based on alkaline silicate gel. Silicate gel formed by the interaction of silicate sodium and carbon agent is the best raw material for water flow isolation. Regardless of the size and geometry of the porous medium, it easily penetrates into the water-saturated depth of the formation and effectively closes the water-releasing channels. The study of the physico-chemical properties of the compositions based on silicate gels allowed to identify the optimal composition for filtration processes and the possibility of increasing the strength of the gel and the resistance to disintegration against water pressure by adding metal nanoparticles to it with a mass fraction of 0.05-0.07% was substantiated [3].

In addition, a mathematical model can be built to prevent the flow of water, and the problem can be solved. In order to solve the mentioned problem, many works have been devoted to the mathematical modeling of the process [4,5]. Despite the conducted research, the problems of prevention of fluidization of the formation and isolation of gels remain unsolved.

From this point of view, the establishment of a mathematical model that can take into account the real conditions and the evaluation of the isolation ability of gels based on it is of both scientific and practical importance.

RESEARCH METHOD

In the presented work, the two-dimensional problem of displacement of oil by water in a non-homogeneous layer is considered. Production and percussive wells are working in the formation. The outer boundaries of the layer are assumed to be impassable. The amount of water injected into the shock wells is given, while the volume of oil and water is predicted in the production wells.

The process of displacement of oil with water and subsequent isolation of the water flow is mathematically modeled with the help of the oil and water filtration equations, the balance equation of the gel distribution in the formation, which provides water isolation in the general flow, and the equilibrium equation:

$$\frac{\partial ms}{\partial t} + div\,(\frac{kf_w(s,c)}{\mu_w(p,c)} grad\ p_w) = Q_w(t)\delta(x,\ y) \qquad (1)$$

$$\frac{\partial m(1-s)}{\partial t} + div(\frac{kf_o(s,c)}{\mu_o(p,c)} grad\ p_o) = Q_o(t)\delta(x,\ y), \qquad (2)$$

$$\frac{\partial}{\partial t}\left[msc + m(1-s)\phi(c) + a(s,c)\right] + div(c\frac{kf_w(s,c)}{\mu_w(p,c)} grad\ p_w + $$

$$+\phi(c)\frac{kf_o(s,c)}{\mu_o(p,c)} grad\ p_o) = (cQ_w(t) + \phi(c)Q_o(t))\delta(x,\ y)$$

$$c(x,y,t) = \begin{cases} 0, t < T \text{until isolation} \\ c_*(x,y,t), t \geq T \quad \text{after isolation,} \end{cases} \qquad (3)$$

$$p_o - p_w = p_k(s,c), \qquad (4)$$

Initial and boundary conditions

$$s(x,y,t)\big|_{t=0} = s_0(x,y), \qquad (0 \le x \le l_x;\ 0 \le y \le l_y),$$

$$(5)$$

$$c(x,y,t)\big|_{t=0} = c_0(x,y), \qquad (0 \le x \le l_x;\ 0 \le y \le l_y),$$

$$\frac{\partial p_w}{\partial x}\Big|_{x=0,l_x} = 0, \qquad 0 \le y \le l_y,$$

$$\frac{\partial p_o}{\partial x}\Big|_{x=0,l_x} = 0, \qquad 0 \le y \le l_y,$$

$$\frac{\partial c}{\partial x}\Big|_{x=0,l_x} = 0, \qquad 0 \le y \le l_y,$$

$$\frac{\partial p_w}{\partial y}\Big|_{y=0,l_y} = 0, \quad 0 \le x \le l_x,$$

$$\frac{\partial p_o}{\partial y}\Big|_{y=0,l_y} = 0, \quad 0 \le x \le l_x, \tag{6}$$

$$\frac{\partial c}{\partial y}\Big|_{y=0,l_y} = 0, \quad 0 \le x \le l_x$$

here, t -time; T - the time from the start of water injection to the entry of water into the operational well; x and y - spatial coordinates; m - porosity; s -saturation with water; c and $\varphi(c)$ - gel volume concentrations in water and oil phases; $a(s,c)$ - amount of gel absorbed per unit volume of porous medium; $(x,\ y)$ - coordinates of wells; $Q_w(t)$, $Q_o(t)$ - - volume consumption of water pumped from the injection well and extracted oil; k_i - coefficient of absolute permeability of the porous medium; $f_w(s,c)$ and $f_o(s,c)$ - relative phase conductivities of water and oil phases in the environment; $\mu_w(p,c)$, $\mu_o(p,c)$ - viscosities of water and oil; p_w, p_o - pressures of water and oil phases; $p_k(s,c)$ -capillary pressure; l_x and l_y -are the length and width of the layer, respectively.

The system of equations (1)-(4) of the filtration model is solved within the conditions (5), (6), and with the help of numerical calculation methods and the Maple program, the unknown quantities s - water saturation, c - gel concentration in the water phase, and p_w - pressure of the water phase are found .

Thus,

$$s_w = \frac{\int_0^{t_o} Q_o(t)dt}{\pi(R^2 - r_c^2)h_1 m_0}, \tag{7}$$

$$Q_o(t) = Q_0 \cdot e^{-\alpha t},$$

$$\phi(c) = c/2, \tag{8}$$

$$p_w(t_o) = p_o(t_o) + Q_o(t_o)\frac{\mu_o h_2 (s_w)^2}{\pi R_w^2 k_2 (1-s_w)^2} \tag{9}$$

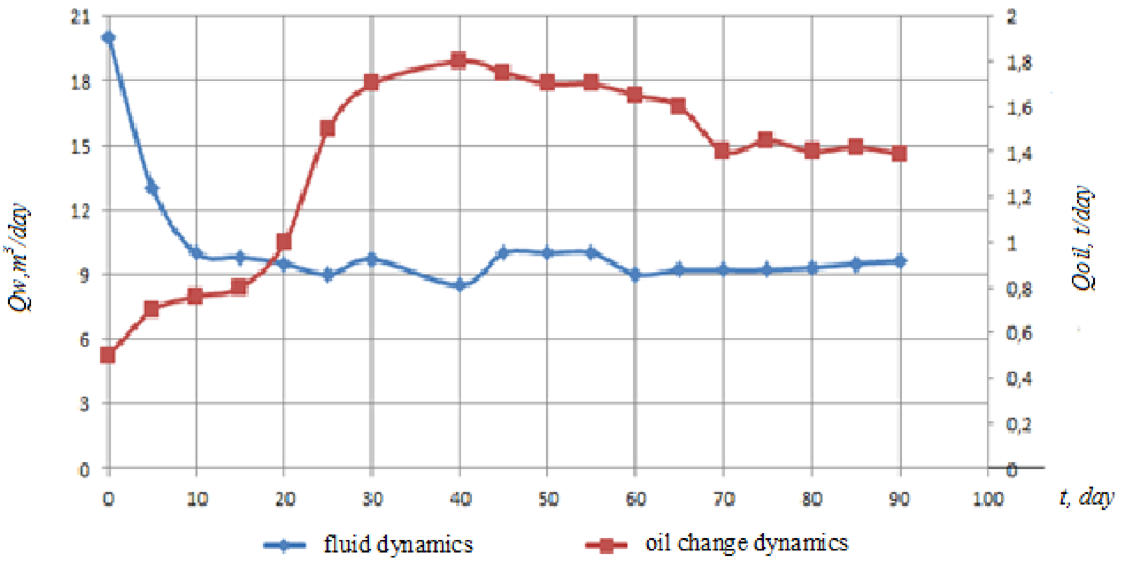
From the following values of reservoir parameters and also from the properties of fluids

$$k_1 = 0.05 \cdot 10^{-12} - 10^{-12} m^2;$$

$$k_2 = 0.05 \cdot 10^{-12} - 10^{-12} m^2;\ s_0(x,y) = 0.2;\ c_0(x,y) = 0;\ a(s,c) = 0; \tag{10}$$

$$\mu_o = 3 \cdot 10^{-9} MPa \cdot s, \quad \mu_w = 1 \cdot 10^{-9} MPa \cdot s$$

calculations were made taking into account formulas (7)-(9) and a graph was constructed based on the obtained results.



Graph. After the selective isolation of the production well from water change in flow rate of fluid and oil

CONCLUSION

After the adaptation of the hydrodynamic model to the drainage area, the possibility of increasing the current oil yield by adjusting the technological process was predicted. The calculation result of the isolation process is presented in the graph above. In Gafik, after the selective isolation of the production well from water, the dynamics of flow depending on the working time of liquid and oil are shown. When the well is put into operation after isolation from water, the production due to oil increases sharply in the initial moments.

The proposed calculation model allows to choose the capillary phenomena in the isolation of the water flow, the degree of isolation and the depth of penetration of the isolation agent, the most optimal moment of action, the necessary degree of isolation (residual resistance factor). The degree of isolation of the water flow significantly affects the efficiency of the process, and by adjusting the degree of isolation, a significant increase in oil recovery is ensured at any stage of reservoir development. As the penetration depth of the isolation agent increases, high isolation efficiency is achieved.

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MOVEMENT OF ANOMAL OIL IN THE ROUND CYLINDRICAL PIPE ACCORDING TO MAXWELL'S LAW OF FRICTION

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ABSTRACT

The article solves a stationary hydromechanical problem about the movement of anomalous oil in a round cylindrical pipe according to the law of friction, i.e., according to the modified Maxwell model. When solving this problem, it is assumed that the direction of oil movement will coincide with the direction of the pipe axis. Between the 1st and 2nd cross sections of the pipe, a part with a length is taken. In this part of the pipe, radius size calculations are taken from the pipe axis. The speed of oil movement depends on the radius and decreases as it increases. At : $r = R : v = 0$ *.*
From the condition of equilibrium of two forces, that is, the pressure force and the friction force, an expression was found for the radius of the flow core. Formulas are presented for the initial pressure ΔP *drop and for the shear stress* τ *. To solve the differential equation of anomalous oil, a technique was used to replace a complex differential with a simple differential. A formula has been derived for the total oil flow rate in a pipe; a formula for pressure loss in the laminar mode of movement of anomalous oil in a pipe has been extracted. When* $\Delta P \leq \Delta P_0$ *the liquid in the pipe does not move, it remains at rest.*

The article solves a new hydromechanical theoretical problem about the rectilinear stationary movement of anomalous oil in a round cylindrical model, that is, according to Maxwell's law of friction [1].

To solve this problem, it was assumed that the direction of oil movement would coincide with the direction of the pipe axis.

In Fig. 1 shows a schematic drawing of a given oil pipe. Between the first and second cross sections of the pipe, a part of it with a length of l . In a given part of the pipe, the radius r dimensions were calculated using the pipe axes.

The speed of movement of visco-plastic oil v depends on the radius r and decreases as r it increases. In the inner surface of the pipe it reaches its lowest value. At: $r = R : v = 0$.

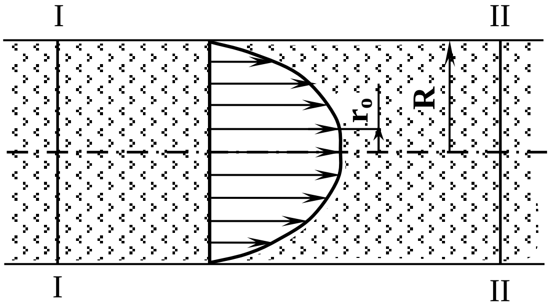

Fig.1. Schematic drawing of a round cylindrical pipe

Due to the fact that as the radius r increases, the speed of oil movement decreases, then the velocity gradient $\frac{dv}{dr}$ receives a negative value (-). Therefore, from the point of view of geometry, it represents an obtuse angle, which is known to be a negative quantity (-).

The modified model – Maxwell's law of friction has the form:

$$\tau = \eta\frac{dv}{dr} + \tau_0 e^{-\frac{t}{T}} \quad \text{.......................................} \quad (1)$$

As you can see, this expression has three constant parameters, which are physical characteristics of the liquid (in this case, oil): η – coefficient of structural viscosity, τ_0 - static shear tightness, T - period of relaxation (weakening).

In formula (1) τ – tangential tightness, e – base of natural logarithm, t – flow time of the technological process.

Equation (1) at $\tau > \tau_0$ expresses the movement of the fluid. During movement, tangen-tial tightness τ must always be greater than the static shear tightness τ_0 and $\frac{dv}{dr}$ as shown abo-ve, it can be a (-) value.

When $r = R$ the tangential tightness in the inner surface of the pipe reaches its maxi-mum value. As the pipe approaches the axis, the tangential tightness decreases and at the radius $r = r_0 : \tau > \tau_0$; that's why $\frac{dv}{dr} = 0$. A flow core with a radius r_0 moves like a rigid body.

Let us determine the value of the radius of the flow core r_0 from the condition of equili-brium of two forces:

1) The pressure forces acting on the lateral surfaces of the flow core are equal to $\pi r_0^2 \Delta P$;

2) The friction forces acting on the surface of the same flow core are equal to $2\pi r_0 l \tau_0$.

Then we get:

$$r_0^2 \Delta P = 2\pi r_0 l \tau_0 \quad \text{..................} \quad (2)$$

hence we have:

$$r_0 = \frac{2l\tau_0}{\Delta P} \quad \text{................................} \quad (3)$$

As noted above, for fluid movement in a pipe, the tangential tightness of the fluid in the pipe must be greater than the ultimate shear tightness, that is to start the movement of liquid in the inner wall of the pipe, it is necessary that $\tau > \tau_0$. At values of $\tau < \tau_0$, the liquid in the pipe does not move.

In fig. 2 shows a graph of the velocity gradient versus tangential tightness. It is part of a general curve in the form of an inclined straight line and intersects with the abscissa axis at the point τ_0 . This graph is expressed by Shulman's friction law. At $\tau < \tau_0$, as can be seen in all its values $\tau\frac{dv}{dr} = 0$. This happens when the speed in the pipe does not depend on the radius and remains constant.

In the inner cylindrical surface $r = R$ (at $\tau < \tau_0$), that is $v = 0$, as a result, the speed does not depend on the radius, in the inner wall of the pipe, the speed is zero at all points. On the surface $r_0 = R$ when $\tau = \tau_0$ the limit equilibrium is not observed and in this case the corresponding value ΔP_0 is determined by the following formula:

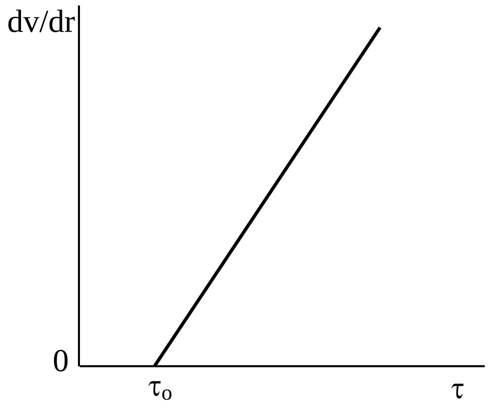

Fig. 2. Gradient modulus graph speed from tangential tightness

$$\Delta P_0 = \frac{2l\tau_0}{R} \quad \text{...} (4)$$

To start the movement of liquid in a pipe, the following condition must be met $\Delta P > \Delta P_0$.

Inside the pipe, we select an annular space with an outer radius r and an inner radius r_0 and create a condition for the equilibrium of the tangential forces that act in the surface of the selected space:

$$2\pi r l \tau - 2\pi r_0 l \tau_0 = \pi(r^2 - r_0^2)\Delta P_0 \quad \text{.................} (5)$$

From formula (3), we substitute the value r_0 in the equilibrium equation (5) and obtain:

$$2\pi r l \boldsymbol{\tau} - \pi r_0^2 \Delta P = \pi r^2 \Delta P - \pi r_0^2 \Delta P \quad \text{........} (6)$$

Thus, the equilibrium condition takes the form:

$$\pi r_0^2 \Delta P = 2\pi r l \tau$$

From here we have:

$$\tau = \frac{r\Delta P}{2l} \quad \text{......................................} (7)$$

From formula (7), we substitute the value τ into Maxwell's equation (1) and obtain:

$$\frac{r\Delta P}{2l} = \eta\frac{dv}{dr} + \tau_0 e^{-\frac{t}{T}} \quad \text{........................} (8)$$

We divide the differential equation (8) into variables and obtain:

$$\frac{\Delta P}{2l\eta} r dr + \frac{\tau_0}{\eta} e^{-\frac{t}{T}} dr = dv \quad \text{...................} (9)$$

We integrate equation (9) within the limits from v to v_0 and from r to r_0 before r and we obtain:

$$\frac{\Delta P}{2l\eta}\int_r^{r_0} rdr + \frac{\tau_0}{\eta}e^{-\frac{t}{T}}\int_r^{r_0} dr = \int_v^{v_0} dv \ ;$$

$$v_0 = v - \frac{\Delta P}{4l\eta}(r^2 - r_0^2) - \frac{\tau_0}{\eta}e^{-\frac{t}{T}}(r - r_0) \ldots\ldots\ldots\ldots (10)$$

From formula (10) we can derive the formula for the pressure drop in the form:

$$\Delta P = \frac{1}{r^2 - r_0^2}\left[(v_0 - v)4l\eta + 4l\tau_0 e^{-\frac{t}{T}}(r - r_0)\right] \ldots\ldots (11)$$

The total flow of abnormal oil in the pipe consists of two parts:

1) fluid flow in the flow core: $Q_1 = \pi r_0^2 v_0$,

2) fluid flow in the annular space around the flow core, that is, in the gradient layer: $Q_2 = \int_{r_0}^{R} 2\pi r dr$.

The total oil consumption will be:

$$Q = Q_1 + Q_2 = \pi r_0^2 + 2\pi\int_{r_0}^{R} rdr \ \ldots\ldots\ldots\ldots\ldots\ldots (12)$$

Substituting the value v_0 from formula (10) in formula (12), we get:

$$Q = \pi r_0^2\left[v - \frac{\Delta P}{4l\eta}(r^2 - r_0^2) - \frac{\tau_0}{\eta}e^{-\frac{t}{T}}(r - r_0)\right] + \pi(R^2 - r_0^2) \ldots\ldots\ldots (13)$$

When $\Delta P \le \Delta P_0$ the oil in the pipe does not move.

Thus, the solution to the problem posed in the article is completed.

CONCLUSIONS

1. The article solves a stationary hydromechanical problem about the movement of anomalous oil in a round cylindrical pipe according to the law of friction i.e. according to the modified Maxwell model.
2. When solving this problem, it is assumed that this direction of oil movement will coincide with the direction of the pipe axis.
3. Between the 1st and 2nd cross sections of the pipe, a part of the pipe with a length of l .
4. In this part of the pipe, radius r size calculations are taken from the pipe axis.
5. The speed of movement of anomalous oil depends on the radius r and with its increase it decreases, at $r = R$; $v = 0$.
6. From the condition of equilibrium of two forces, that is, the pressure force and the friction force, an expression was found for the radius of the flow core.
7. Formulas are presented for the initial pressure drop ΔP_0 and for the tangential tightness τ.
8. To solve the differential equation of motion of anomalous oil in a pipe, a technique was used to replace a complex differential with a simple differential.
9. A formula has been derived for the total oil flow in the pipe.

10. The formula for pressure loss in the laminar mode of movement of anomalous oil in a pipe has been extracted.
11. At $\Delta P \leq \Delta P_0$ the liquid in the pipe does not move, it remains at rest.

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STABILIZATION OF RESIDUAL STRESSES IN WELDED STRUCTURES OF PRESSES BY PULSE UNLOADING BEFORE VIBRATION

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ABSTRACT

This study focuses on the stabilization of residual stresses in welded structures of presses through pulse unloading prior to vibration treatment. Residual stresses in welded components can compromise structural integrity, reduce fatigue life, and lead to premature failure. The proposed approach involves applying controlled pulse unloading to redistribute and stabilize stress fields, creating a more uniform stress state before subjecting the structure to vibration. Experimental and analytical results demonstrate that the combination of pulse unloading, and vibration treatment significantly reduces peak residual stresses and improves the mechanical properties of welded assemblies. This method provides an efficient solution for enhancing the durability and performance of welded structures in industrial applications. The influence of various stabilizing treatments on the properties of steel welded samples is investigated. The efficiency of the vibration method with pulsed unloading for reducing the level of residual stresses in samples by 1.3-2.3 times and ensuring their geometric stability is demonstrated.

Introduction. Stabilization of the level of residual stresses and the dimensions of welded press frames is one of the most important tasks in mechanical engineering. The most probable potential source of such stresses is non-uniform plastic deformation caused by thermal stresses during welding. Various relaxation processes are used to stabilize the level of residual stresses [1]. The most universal of these is thermal action by high tempering of parts (600...650ºC). However, this method requires significant expenditure of natural gas. Therefore, methods of stabilizing residual stresses and the geometry of parts using vibration processing have become widespread at present [2-4].

The most frequently used vibration treatment involves excitation of forced undamped oscillations of a certain amplitude in parts. This allows for a significant reduction in energy costs and labor intensity during stabilizing treatment but is associated with the use of vibrators that create a certain noise level, which is not always acceptable, as well as with the need to select the main parameters of vibration action for each specific part (type of dynamic stress state, required level of dynamic stresses, conditions for fastening the part, etc.). A vibration treatment method has been developed that is free from the above-mentioned disadvantages. Its essence lies in excitation of free damped oscillations in the part because of practically instantaneous (pulse) removal of a previously applied static load (VIR method) [5]. It is advisable to determine the efficiency of this method of vibration treatment of large parts such as crank press frames.

The aim of this work was to evaluate the effectiveness of various types of relaxation treatment to stabilize the level of residual stresses in the weld zone and ensure the geometric stability of welded parts.

Research methodology. Model welded samples made of grade 20 steel were used as objects of study. The samples were unclosed rings with an outer diameter of 240 mm with a transverse weld seam made opposite the cut.

The following relaxation treatment options were investigated:

1. tempering at a temperature of 650 ± 10ºC, 1.5 h;

2. vibration treatment by exciting forced resonant oscillations (VRC method);

3. vibration method using pulse unloading (VIR method).

The initial option was welding without subsequent processing.

Excitation of forced resonant vibrations was carried out using a model 489P installation, consisting of a spring-loaded vibration table with an unbalanced vibrator of the IV-98 type mounted on it and a control panel that allows changing the frequency of the vibration exciter from 20 to 100 Hz.

The experimental setup for processing by the VIR method is a special device for pulsed unloading of samples, installed on a universal machine of the UME-10TM type. The device consists of a loading device, which, with the help of disk springs and a trigger mechanism, throws the punch away from the sample at high speed, because of which free damped oscillations arise in the sample released from the static load (Fig. 1).

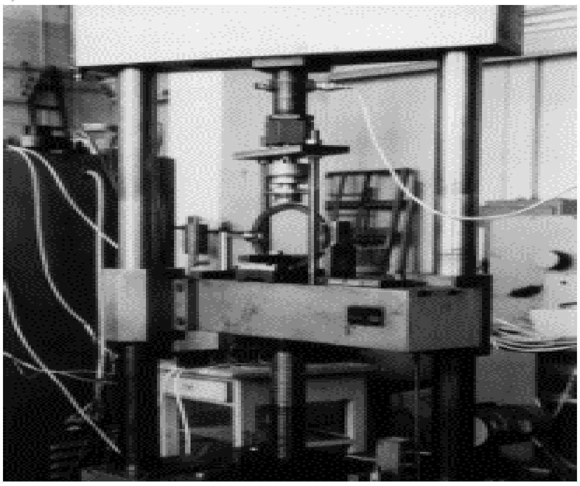

Fig. 1. General view of the installation for VIR processing

Discussion of results. Residual macro stresses were determined on the surface of the samples by the X-ray sin2ψ method on a stationary diffractometer of the DRON-3 type in Cr-anode radiation. The geometric stability of the samples was assessed by determining the residual deformation (ΔL) after testing the samples under the action of a static load equal to 0.8 of the proportionality limits (σpts) for 144 h.

Statistical processing of the obtained results was carried out using the standard method [6]. The assessment of the homogeneity of the distribution of residual stresses near the weld was determined by the value of the standard deviation (S).

The VIR method processing was carried out at different values of the initial static force (0.6 kN; 1.0 kN; 1.4 kN), which correspond to stresses in the welding zone equal to 0.28$\sigma_{пц}$; 0.45$\sigma_{пц}$; 0.6$\sigma_{пц}$, and at a different number of loading cycles.

The results of determining the values of residual stresses in welded samples are given in Table 1.

Table 1. The influence of VIR processing parameters on the magnitude of residual stresses in welded samples

No. p/p	VIR processing parameters		Average value of σ_{ost}, MPa	S
	Initial static load, kN	Number of loading cycles		
1	0.6	original	-220	29.8
		1	-215	15.1
		5	-205	42.0
		10	-230	25.7
2	1.0	original	-265	34.2
		1	-260	20.5
		5	-213	9.6
		10	-270	38.4
3	1.4	original	-260	36.1
		1	-240	35.3
		5	-233	25.9
		10	-250	38.6

As the obtained data show, the most appropriate is fivefold processing by the VIR method, since with a single processing there is practically no change in the value of residual stresses compared to the initial version, and an increase in the number of processing cycles to 10 leads to some growth in the values of residual stresses compared to fivefold processing and causes an increase in the heterogeneity of their distribution. It was established that a decrease in residual stresses near the weld seam along with their greatest homogeneity is ensured at an initial static load of 1.0 kN.

Therefore, further results are given for VIR processing in the optimal mode (initial static force 1.0 kN, number of loading cycles – 5).

Tables 2 and 3 present the results of the study of residual stresses and residual deformation, characterizing the influence of various stabilizing treatments on them.

Analysis of the obtained results (see Table 2) shows that tempering at a temperature of 650 ± 10ºC leads to a decrease in residual stresses by 2.2 times, significantly increasing the homogeneity of their distribution, which is confirmed by calculating the sample standard deviation. The decrease in residual stresses during such tempering is accompanied by some residual deformation of the samples (see Table 3).

Vibration treatment using forced undamped oscillations in resonance modes also reduces the level of residual stresses in comparison with the initial version by 1.85-2.25 times. At the same time, judging by the values of S, vibration treatment for 30 sec allows to reduce the heterogeneity of the distribution of residual stresses. Increasing the time of VRK treatment to 900 sec leads to a significant increase in the heterogeneity of the distribution of σres.

It should be emphasized that long-term vibration treatment in a resonant mode at high stress amplitudes (0.5…0.7σпц) leads to the formation of microcracks in welded seams and even to the destruction of samples. In addition, such treatment causes significant residual deformation (see Table 3).

Table 2. Effect of different stabilizing treatments on the level of residual stresses in welded samples

No. p/p	Processing option		Values of residual stresses $\sigma_{ост}$, MPa						
			Distance from the edge of the weld					σ_{ost}(average)	S
			Left side		Center of the seam	Right side			
			2mm	1mm	-	1mm	2mm		
1	vacation	original	-170	-200	+100	-250	-180	-200	35.6
		650±10⁰C, 1.5h	-80	-110	+40	-90	-78	-90	14.6
2	VRK-processing*	original	-220	-225	+180	-200	-210	-214	11.1
		30 sec	-120	-100	+50	-110	-95	-106	10.9
		900sec	-95	-65	+85	-145	-75	-95	35.6
3	VIR-processing **	original	-260	-220	+200	-280	-300	-265	34.2
		5 cycles	-220	-210	+90	-200	-220	-213	9.6

*–at a frequency of 100 Hz;

**–initial static load 1 kN Thus, when using the method of exciting, forced oscillations at a resonant frequency, the processing time should be carefully selected to avoid artificially reducing the durability of the part under subsequent operating conditions.

Table 3. Effect of various stabilizing treatments on the magnitude of residual deformation of welded samples

No. p/p	Processing option	The magnitude of residual deformation ΔL, mm
1	Welding without further processing (original)	0.51
2	Vacation at 650±10⁰C, 1.5 hours	0.21
3	Vibration treatment by VRK method* for 30 sec	0.19
4	Vibration treatment by VRK method* for 900 sec	1.3
5	Vibration treatment by VIR method (1 kN, 5 cycles)	0.2

*– at a frequency of 100 Hz

The VIR processing method is free from such disadvantages, allowing, on the one hand, to strictly regulate the stress level by setting the value of the initial static load, and on the other hand, to use a minimum number of loading cycles.

VIR processing under optimal conditions leads to a decrease in the values of residual stresses near the weld, while simultaneously ensuring the greatest uniformity of their distribution.

From the data presented in Table 3 it follows that tempering at 650 ± 10°C, vibration treatment by the VRK method for 30 sec and VIR treatment in the optimal mode lead to practically the same residual deformation, half that of the samples after welding. This allows us to conclude that vibration treatment is highly effective in stabilizing the geometry of parts.

Conclusions

The conducted comparison of the efficiency of the studied stabilizing treatments showed that these methods allow to reduce residual macrostresses by 1.3...2.3 times, and (under optimal conditions) provide a decrease in the standard deviation of the stress value, characterizing the heterogeneity of their distribution near the weld, by 1.1...3.5 times. Almost the same efficiency of the studied methods for stabilizing the dimensions of samples under the effect of long-term static loading in the elastic region is shown. It is established that residual macrostresses are not the only determining criterion of vibration treatment, since the magnitude of the reduction and their absolute values are not unambiguously related to the stability of the dimensions of the sample or part. Varying the main parameters of VIR treatment revealed that the most effective mode is the one in which the sum of the initial dynamic and residual stresses exceeds the proportionality limit σпц by approximately 10% under the selected loading scheme. Moreover, to create a more uniform stress state, the number of processing cycles should be equal to 5. Consequently, vibration processing is an effective way to reduce the level of residual stresses in welded parts and stabilize their geometry, and of the two methods studied, the method of vibration processing by pulsed unloading is preferable as it is more controllable, less energy-intensive and does not require noise protection.

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EFFECT OF COOLING HERMETİC COMPRESSOR ON NOISE REDUCTION IN A DOMESTIC REFRIGERATOR

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ABSTRACT

In the household refrigerator compressor is the main source of noise and heat, and the isolation of sound from the compressor in the air increases the thermal stress of his work and a significant reduction of energy efficiency of the refrigerator. Installing sound insulation panel in front of the compressor housing to reduce the noise level. The cooling system compressor oil is widely used in foreign and domestic refrigerators, implements "tropical" operating temperature Options refrigeration units tested at optimal doses of freon and constant permeability of the capillary tube. Soundproofing compressor housing leads to some increase in temperature of the motor windings and optimization of dose refrigerant with soundproofed compressor housing will increase the value of these indicators. The results confirm the possibility of reducing the noise of the refrigerator, through the use of an optimized heat removal from the cylinder head of the compressor.

Introduction

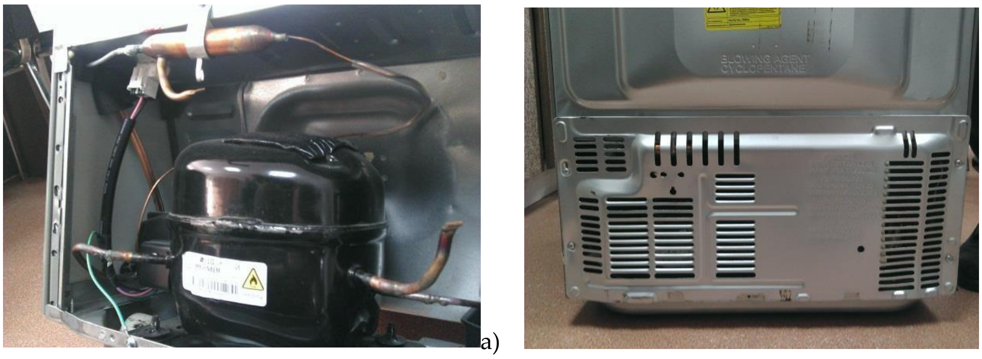
Vibroacoustic characteristics are an important indicator of the competitiveness of a household refrigerator as a representative of household appliances [2], and the hermetic refrigerant compressor has a decisive influence on their generation. Research into a hermetic compressor, which is a multi-level and multi-component airborne noise emitter, was implemented in the form of a set of technical solutions that make it possible to reduce the noise level of the compressor itself to 38-36 dBA [1]. Acoustic technical solutions for reducing noise levels are divided into: sound insulation, sound absorption and sound dampening.

Compressor technical level

A known sound-reflecting technical solution is described in the operating instructions for the ZIL-63 refrigerator [2]. In the lower part of the refrigerator cabinet there is a recess with a sealed compressor 1 placed in it and a metal shield 3 that partially covers the machine compartment, and the inner side of the shield is covered with sound-absorbing material.

Installing a soundproofing shield in front of the compressor casing allows you to reduce the noise level by 1.5 ÷ 2.5 dBa. A graph for reducing the noise level across the spectrum with sound-absorbing shields is given in the book by I.V. Bolgov, A.I. Naberezhnykh. and others.

Installing a soundproofing shield in front of the compressor casing significantly increases the temperature level of compressor operation, which is clearly demonstrated by studying the temperature field with a thermal imager in Fig.1.



a) b)

Figure 1. Study of LG refrigerator model GA-B409SLQ

a) image of a refrigerator without a shield; b) image of the soundproofing shield in front of the compressor casing

When a hermetic refrigeration compressor operates in a normalized temperature regime [4,5], the temperature of the working winding of the electric motor reaches 120 °C, the temperature of the suction vapor in the cylinder is 145 ÷ 155 °C, and the temperature of the compressed vapor is 170 ÷ 190 °C. It is obvious that the use of these compressors in refrigerators with sound and thermal insulation of the compressor, as well as in a tropical version without an additional cooling system, is not acceptable since the temperature of the electric motor windings exceeds 130 °C.

The following methods of cooling hermetic compressors have found their application in household refrigerators:

• suckedin refrigerant vapors;

• heat removal from the surface of the casing by external blowing with a fan;

• heat removal from the casing by cooling the oil with liquid refrigerants from the precondenser or a heat pipe.

Cooling of the hermetic compressor housing is achieved by forced circulation in the engine compartment, which is sucked in by a fan through the right side of the front grille (Fig.1 b) and thrown out of the compartment into the room through the left side of the grille.

The oil cooling system in the compressor has become widespread in foreign and domestic refrigerators, realizing a "tropical" temperature operating mode. In them, cooling is achieved by evaporation of liquid refrigerant coming from the precondenser into a coil, which is located in the oil bath of the compressor casing.

Calorimetric studies carried out by the Mazeikiai Compressor Plant and the Russian State University of Tourism and Service (formerly MIT) [5] show that compressors with mass production of the "S-KO – OS" series with cooling of the oil bath with liquid refrigerant from the precondenser, which can increase the cooling capacity by 3 ÷ 4.5%, the efficiency coefficient

(COP) by 3 ÷ 4%, and the power consumption remains practically constant. The temperature of the electric motor winding decreases by 15 ÷ 20 °C depending on the area of the precapacitor. These indicators are not highly effective and require further improvement and search for a constructive solution [3].

The technical problem is to develop and study the efficiency of a compressor cooling system, which makes it possible to enhance the sound insulation of the compressor.

A system has been developed for cooling the compressor head with liquid refrigerant from the precondenser [6], shown in Fig.3. A study of this system shows that the process of cooling the compressor elements from the precondenser is carried out with liquid refrigerant in the so-called calorimetric mode [4] in nominal parameters and provides: an increase in cooling capacity and efficiency coefficient, respectively, by 9 ÷ 16% and 15 ÷ 20%; reduce the temperature of the electric motor winding by 21 ÷ 25%; remove the thermal energy of the compression work through the pre-condenser into the environment at a temperature of 60°C.

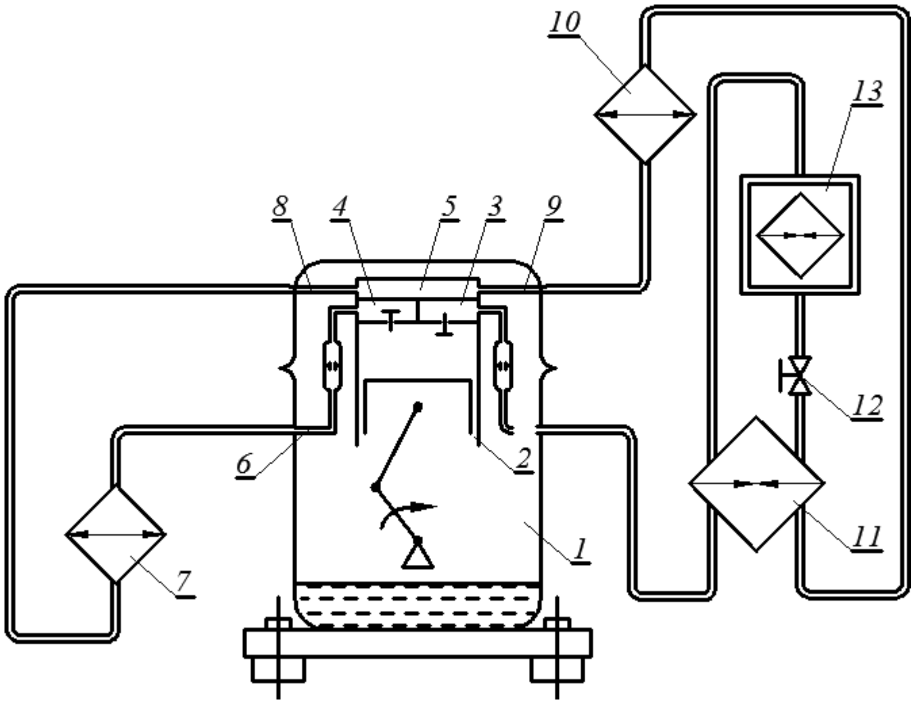


Figure 2. Cooling system of the compressor head with liquid refrigerant:
1 – GKhK; 2 – cylinder; 3 – suction chamber; 4 – discharge chamber; 5 – cylinder head; 6 – coil for removing refrigerant vapors from the compressor (discharge coil); 7 – precondenser; 8 – coil for introducing liquid refrigerant into the cylinder head; 9 – coil for removing refrigerant vapors from the cylinder; 10 – condenser of the refrigeration unit; 11 – regenerative heat exchanger; 12 – capillary tube; 13 – evaporator

The object of the study is a hermetic compressor S-KO120N5 (Fig.3) produced by the Baranovichi Machine Tool Plant (Belarus), with a cooling capacity of 123 W, included in the Atlant refrigeration unit with a pre-condenser, from which liquid refrigerant is supplied through a flexible coil into a sealed cavity mounted on the head cylinder block, where the coolant boils under condensation pressure, intensively cooling the compressor parts (Fig.3).

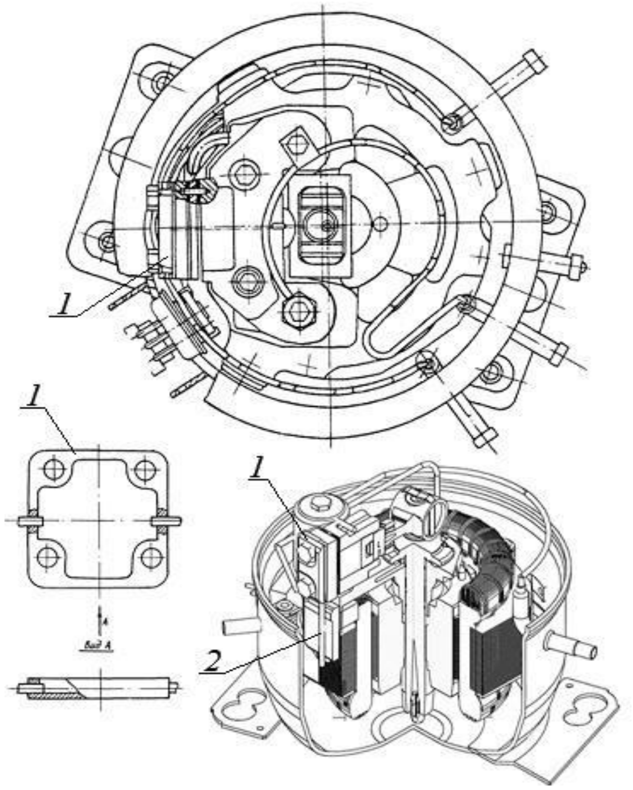


Figure 3. Hermetic piston-type compressor for household refrigeration equipment with cylinder head cooling
1 – cylinder head; 2 - coil for removing refrigerant vapors from the cylinder

The cylinder cooling head is structurally an element with a milled part to limit the lift of the discharge valve. An installed partition separates the suction and discharge cavities. Coolant for cooling the cylinder head is supplied through flexible steel coils.

Compressors with a cooled cylinder head and a soundproofed outer surface of the compressor casing were subjected to standardized [3,6] tests on a calorimetric stand.

Two modifications of the refrigeration unit were tested in the calorimetric cycle, presented in Fig.4.

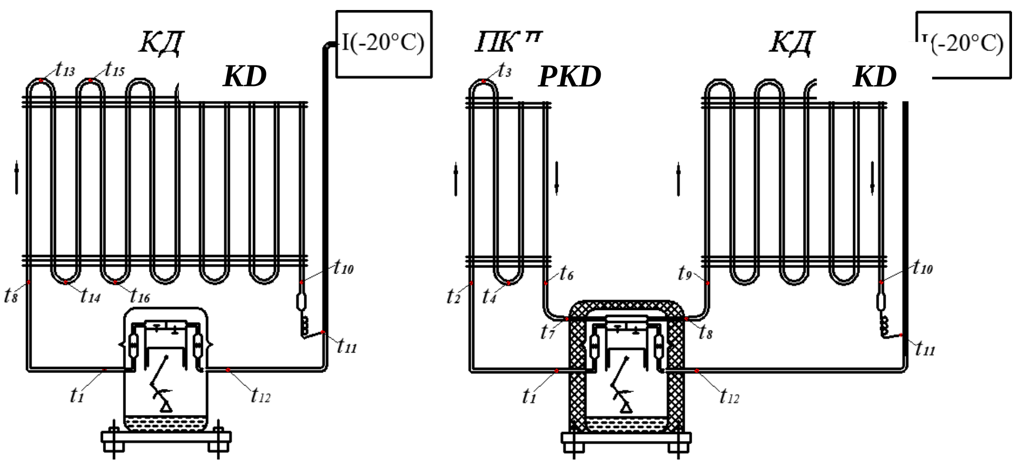


Figure 4. Modifications of the studied refrigeration unit circuits

a) option 1 serial refrigeration unit without a cooling system (condenser area *F*cd = 0.776 m2); b) option 2, the refrigeration unit includes a compressor with a cooling head (pre-condenser with an area of *F*cd = 0.26 m2 and a condenser with an area of *F*cd = 0.52 m2). I – evaporator, KD - condenser, PKD - pre-condenser.

The sound insulation of the compressor casing in the refrigeration unit was made in the form of a technological cap, Fig.3, made of polyurethane foam with a wall thickness of 20 ÷ 30 mm, which covered the compressor mounted on a polyurethane foam stand 50 mm thick. In the cap for the pipes connecting the compressor to the condenser and precondenser, a window measuring 190 × 170 mm was cut out. The outer diameter of the cap is 280 mm, height 260 mm.

The refrigeration unit made according to option 2 was tested twice: with and without soundproofing of the compressor casing at an ambient temperature of 32 °C and 43 °C. The results are presented in Fig.4a.

Test results

The main results of calorimetric testing of a compressor with sound insulation and without sound insulation of the compressor casing (Fig.4) are presented in Fig.5. Analysis of the results of calorimetry of a compressor with sound insulation and without sound insulation of the compressor casing allows us to draw the following conclusions:

a) the use of a cooling system with liquid refrigerant from the pre-condenser of the cylinder head ensures that the compressor casing is isolated from the environment with sound-insulating material, practically without disturbing its temperature regime and loss of cooling capacity.

b) the introduction of sound insulation will reduce the noise level by an amount determined by the thickness and effectiveness of the soundproofing material.

The results of testing refrigeration units (options 1 and 2) in the calorimetric cycle are presented in Fig.5a.

In Fig.5 shows a graph of changes in the temperature field of the pre-condenser and condenser of the refrigeration unit in calorimetering mode with a device for cooling the cylinder block with sound insulation and without sound insulation of the compressor casing at an ambient temperature of 32 °C and 43 °C.

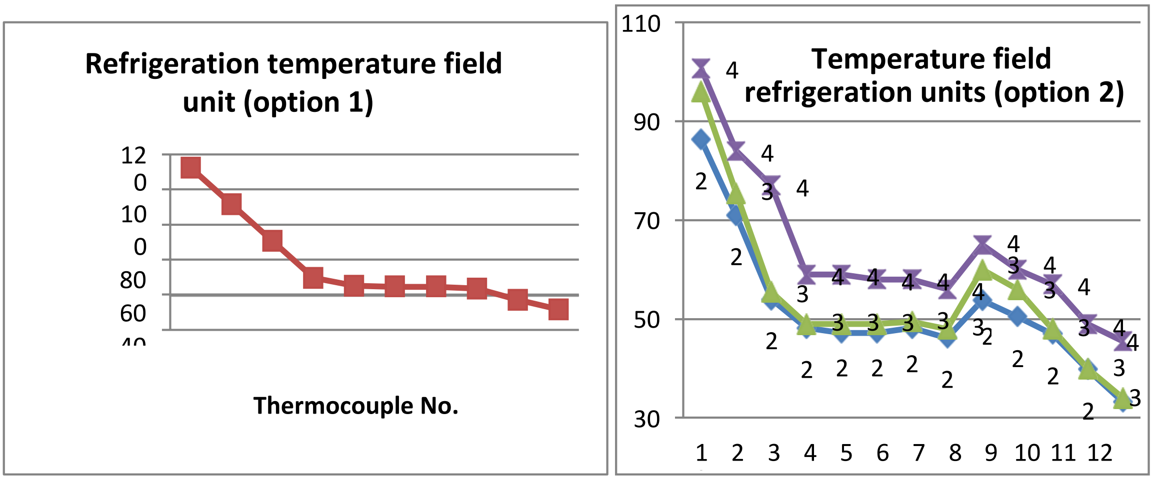


Figure 5. Results of the study of the temperature field of the refrigeration unit

(1-1) - the nature of the change in the temperature of the refrigerant in the condenser (option 1) at an ambient temperature of 32 °C without sound insulation of the compressor casing.

(2-2) and (3-3) – the nature of the change in the refrigerant temperature in the precondenser, cooling head and condenser (option 2) at an ambient temperature of 32 °C without sound insulation of the compressor casing (2-2) and with sound insulation of the compressor casing (3-3).

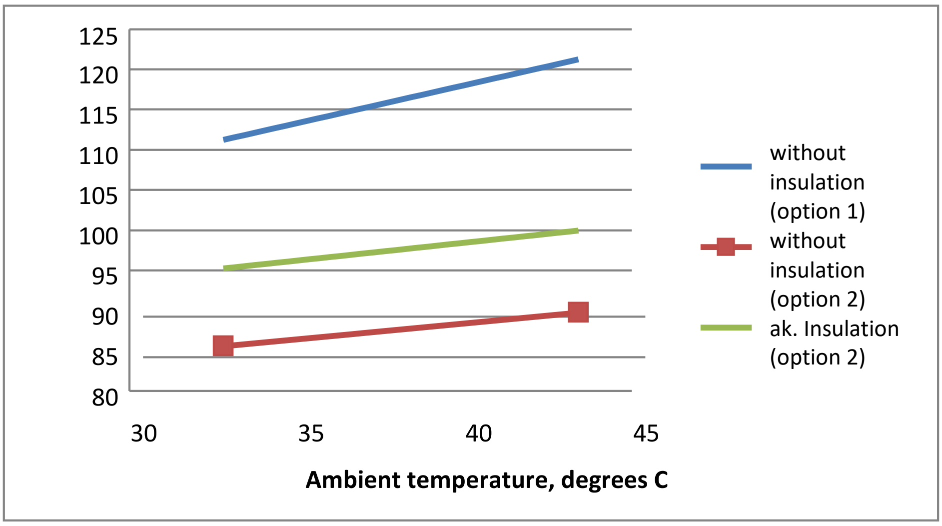


Figure 6. Dependence of ambient temperature on compressor winding temperature

Variants of refrigeration units were tested at optimal doses of refrigerant and a constant flow rate of the capillary tube of 6.2 ± 0.2 l/min. The optimal dose of refrigerant was selected for each option based on the ambient temperature of 32 °C and 43 °C without compressor casing insulation. When installing sound insulation of the casing and compressor, the dose of refrigerant was not adjusted, but remained the same, as when testing a refrigeration unit without sound insulation of the compressor casing.

Analysis of the research results shows that at temperatures of 32 °C and 43 °C, both with and without sound insulation of the compressor casing, there is an increase Q_0, Ke, decrease in the temperature of the working winding with an increase in the area of the precapacitor.

Sound insulation of the compressor casing leads to a slight decrease in Q_0, Ke, and an increase in the temperature of the operating winding of the electric motor. It is likely that optimizing the refrigerant dose during soundproofing of the compressor casing will increase the value of these indicators.

From the presented dependencies it follows that the sound insulation of the compressor casing is acceptable when operating the refrigeration unit at temperatures not exceeding 32 °C. For temperatures of 43 °C, operation of refrigeration units is possible (recommended pre-condenser area Fpkd = 0.64 m2).

In Fig.7 shows comparative results of the adjusted sound power level of MXM-268 refrigerators in the 1/3 octave frequency range with a cylinder head cooling system and sound insulation and without sound insulation of the compressor casing.

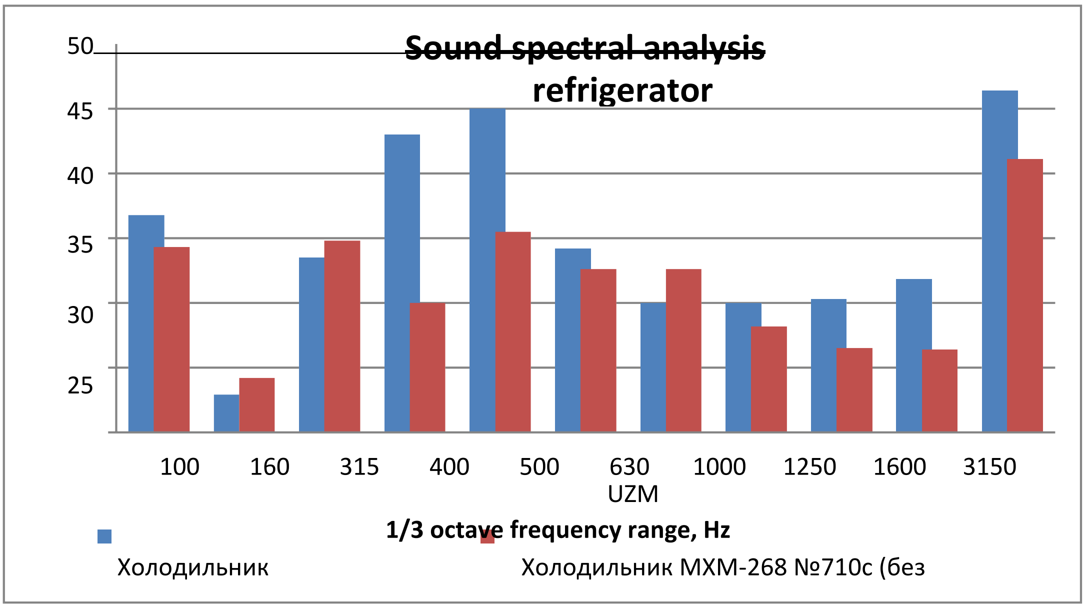


Figure 7. Comparative analysis of the adjusted sound power level of refrigerators in the 1/3 octave frequency range

Conclusion

1. Calorimetric tests of a compressor with a cooling system with liquid refrigerant from the cylinder head pre-condenser showed that the values of cooling capacity, power consumption and temperature of the electric motor windings make it possible to soundproof the compressor casing with insulating material.
2. The cooling capacity and refrigeration coefficient of a compressor with cylinder head cooling and a soundproof casing is higher than that of a compressor without a cooling system by 1 and 10%, respectively, and the operating winding temperature and sound power level are lower by 17% and 11%, respectively.
3. The temperature of the compressor operating winding with cylinder head cooling and soundproof casing does not exceed 100 °C at Tamb., WHO. = 32 °C and 105 °C.
4. The adjusted sound power level of a household refrigerator LG model GA-B409SLQA with a soundproof shield in front of the compressor (Fig.1.b), determined according to GOST 30163.0-95, was 47.56 dBA. Based on the presented results, it can be assumed that the use of heat removal from the compressor cylinder head [5], where recommendations for process control can be used [6], will increase the sound insulating and sound reflecting ability of the plate and thereby reduce the noise level of the refrigerator and increase its energy efficiency.

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JUSTIFICATION OF INCREASING THE DURABILITY AND PRODUCTIVITY OF PRECISION PARTS USING LASER METHOD

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ABSTRACT

The work summarizes and critically analyzes the existing criteria for optimizing the methods of increasing and restoring the surface hardness of precision parts of machines and devices, shows ways to improve and clarify them, and also attempts to select optimal methods for increasing and restoring the surface hardness of precision parts of machines and devices and develop a generalized model. Various methods for increasing and restoring the surface hardness of parts of machines and devices with increased surface hardness pose the most urgent problem of choosing the optimal option for specific production conditions from among various technological solutions. This necessitates a further study of methods for increasing and restoring surface hardness and is based on three criteria: applicability, durability and technical and economic efficiency. The solution to the problem of substantiating the optimal method for increasing and restoring the surface hardness of machines and devices should be carried out using properly selected optimization criteria, using controllable factors that have the greatest impact on the costs of the technological process.

1. Introduction

The activities of many enterprises show that enterprises are constantly looking for ways to remain competitive in market conditions. One of them is the continuous improvement of the level of organization of increasing the surface hardness of machines and devices using laser technology. To date, scientific research institutions have not developed and applied in production methods for increasing and restoring the surface hardness of precision parts of machines and devices. The importance of each of them should be determined by the goals and capabilities of the enterprise [1 - 4].

Various methods for increasing and restoring the surface hardness of parts of machines and devices with increased surface hardness pose the most urgent problem of choosing the optimal option for specific production conditions from a variety of technological solutions. It is necessary to ensure that the costs of increasing and restoring the surface hardness of precision parts are minimal. For this, it is necessary to take into account the maximum number of factors affecting the technology and organization of enterprises [5- 10].

The most acceptable forms of organization of increasing and restoring the surface hardness of parts: by defects, by route and by route-group. Depending on the program and type of production and repair work, it is necessary to choose one of the organizational forms (taking into account the type of production and repair enterprise) [11].

Defect technology is used in cases where the program for increasing and restoring the surface hardness of parts is small. In this case, the technological process is developed separately for each defect. Parts for increasing and restoring the surface hardness are processed only for parts of the same name, taking into account the combinations of defects in them.

Route technology is designed for complex defects that are eliminated in a certain sequence (route). In the design process, it is necessary to generalize the defects based on scientific and practical sources.

After the routes are determined, their number is determined (two or three), a rational method of eliminating defects is selected for each route, and a technological process scheme is developed for eliminating each defect. It is taken into account that when using defect technology, defects on the main surfaces are first eliminated, and then defects caused by thermal processing, deformation and other reasons are eliminated by taking into account mechanical processing [12].

Route-group technology involves the division of defective parts into classes and groups and the development of a single route technological process for increasing and restoring the surface hardness of groups of parts on one equipment using one tool.

Route and route-group technologies are used, as a rule, in manufacturing enterprises and repair plants.

In addition to meeting operational requirements, increasing the surface hardness of precessional parts of machines and devices (especially in market conditions) should be economically beneficial. In this regard, it is more expedient to use complex approaches and the interaction of multiple factors: constructive, technological, production, operational, etc. [13].

2. Materials and methods

Since in modern conditions the production of equipment has decreased by 20...30 times and the supply of spare parts has sharply decreased, the solution of the problem of increasing the performance of precessional parts of machines and devices is urgent. Therefore, the selection and justification of the method of increasing and restoring the surface hardness of precessional parts of machines and devices are the main tasks [14].

The analysis of scientific literature and research works showed that currently the assessment of methods of increasing and restoring the surface hardness of precessional parts of machines and devices is carried out according to the most generalized (technical and economic) criterion: $C_{SMA},_{Bi}$

$$G_i = \frac{C_{SMA,Bi}}{K_{Di}} \min; (1)$$

$$G_i = \frac{C_{SMA,Bi} + E_H \cdot K_{YDi}}{K_{Di}} \min, (2)$$

where,

$C_{SMA,Bi}$ – The unit cost of the defect elimination method using the ith method i, money/m^2;

K_{Di} - The durability coefficient of the part whose surface hardness has been increased and restored by the i-th method;

E_H - normative efficiency coefficient;

K_{YDi} - Special investment when eliminating the defect using the i-th method, money/m^2.

Choosing a rational method for increasing and restoring the surface hardness of parts. The methodology for selecting a rational method for increasing and restoring the surface hardness of parts is based on consideration of three criteria: applicability, durability, and technical and economic efficiency.

Application criterion or technological criterion

K_m = f (M_d; F_d; D_d; Y_d; H_d; $\sum_{i=1}^{m} T_i$) (3)

where

M_d – is the material of the part;

F_d; D_d – The shape and diameter of the part with increased and restored surface hardness;

Y_d – wearing of the part;

H_d – The loading value and nature of the part where the surface hardness is increased and restored;

$\sum_{i=1}^{m} T_i$ – a set of technological characteristics of a method that determines its rational scope of application.

The application criterion allows you to determine the methods by which the surface hardness of parts can be increased and restored and is not expressed in numbers..

The durability criterion is numerically determined by the durability coefficient K_d, which is proportional to the service life of parts under operating conditions:

$K_d = K_Y K_B K_İ$ (4)

where $K_Y, K_{B,} K_İ$ are the wear resistance, durability and adhesion coefficients of the coating, respectively. The most rational method can be selected based on the criterion of durability during operation.

$K_Y, K_{B,}$ coefficients are determined by wear and fatigue tests. The most commonly used method for determining $K_İ$ is the nail pull-out method.

The rational surface hardness increase and restoration method is a technological process that ensures the wear resistance and hardness of the part with increased surface hardness and restored surface hardness, providing a service life of at least 0.8 times that of the new part on the friction surface

3. Discussions

The optimal method for increasing and restoring the surface hardness of precessional parts of machines and devices is considered to be the method with the lowest value of the G_i criterion. Using formulas (1) and (2), it is advisable to determine the optimality criteria for individual parts. As a rule, manufacturing enterprises incur certain costs associated with increasing and restoring the surface hardness of a group (range) of parts, and in this regard, the optimal methods for one part may not always be optimal for a group of parts [5 – 9, 15].

Analysis of expressions (1) and (2) also shows that when evaluating methods for increasing and restoring surface hardness, it is assumed that the indicators $C_{SMA,B}$ and K_D are in equilibrium. However, taking into account the current trends in the production of machines and devices and the economy as a whole, the importance of these indicators is not equal. Determining the numerical value of the technical and economic criterion refers to calculating the costs of increasing and restoring the surface hardness of precessional parts of machines and devices and determining the durability coefficients.

The final decision on the feasibility of using the selected method is made in accordance with the technical and economic criterion, which is the correlation of the costs of increasing the surface hardness and restoring the part after eliminating defects with its durability [16].

The condition for the technical and economic efficiency of the surface hardness and restoration method is as follows:

$C_X \leq K_d \ C_H$ (5)

where C_X Costs of increasing and restoring the surface hardness of the part; C_H - is the cost of the new part.

The effectiveness of the surface hardening and restoration method is assessed taking into account the cost of environmental costs due to the characteristics of the physical and mechanical properties of the surfaces with increased and restored surface hardness. After increasing and restoring the surface hardness, the consumption of lubricants may decrease or increase, the time spent on maintenance may decrease or increase, and the service life of the connected part may decrease or increase. This criterion can be expressed by the following dependence [3, 4]:

$\sum C_{Э,в} \ \leq \sum C_Э$ (6)

where $\sum C_{Э,в}$ is the total operating costs incurred in the production or restoration service of a part with increased or restored surface hardness; $\sum C_Э$ - is the amount of operating costs incurred in servicing newly manufactured parts.

The economic feasibility of using this method to increase and restore the surface hardness of parts is assessed in the following sequence.

First, the cost of increasing and restoring the surface hardness of the part is determined:

$\sigma_{в,\partial} = \frac{\sum C_{в,\partial}}{L\ a}$ (7)

where $\sigma_{в,\partial}$ – The cost indicator for increasing and restoring the surface hardness of a part in a unit of value characterizing the operation of the machine (1 km of mileage between repairs, 1 hour between repair cycles); $\sum C_{в,\partial}$ – Costs of increasing and restoring the surface hardness of parts; L – Machine operating time between production and repair, km/h; a – The ratio of surface hardness to the performance characteristics of increased and restored parts, parts manufactured using factory technology [17].

Then, the cost indicator is determined in case of replacing the unusable part with a new one:

$\sigma_{H,\partial} = \frac{\sum C_{H,\partial}}{L}$ (8)

where $\sigma_{H,\partial}$ - Cost indicator when replacing a wearing part with a new one, per unit of value characterizing the overhaul period (1 km, 1 hour, etc.); $\sum C_{H,\partial}$ - is the value of the new part. In simplified form, the coefficient a is equal to one.

The ratio of cost indicators $\sigma_{в,\partial}$ to $\sigma_{H,\partial}$ characterizes the economic efficiency of increasing and restoring the surface hardness of wearing parts using the selected method:

$\beta = \sigma_{в,\partial} / \sigma_{H,\partial} \leq 1$ (9)

where β is an indicator of the economic efficiency of increasing and restoring the surface hardness of the wearing part.

In the case of β ≤ 1, the method of increasing and restoring the surface hardness of parts is the most rational. It meets the requirements of the durability criterion, that is, it ensures high wear resistance and hardness of the part for at least 0.8 of the service life of the new machine, and also meets the requirements of the technical and economic criterion.

When β = 1, the method of increasing and restoring the surface hardness of parts meets the requirements of the durability criterion and does not lead to an increase in costs during repair and operation in the event of replacing a wearing part with a new one.

When choosing methods for increasing and restoring the surface hardness of precision parts of machines and devices, the cost is expressed by the following formula, depending on the annual program:

$C_{SMA,B} = C_{\mathrm{Dy}} \bullet \mathrm{N} + C_{\mathrm{S}}$ (10)

where,

$C_{SMA,B}$ – cost price;

C_{Dy} – part whose value varies depending on the program;

C_{S} – fixed part that does not depend on the program.

The advantage of this expression is that the author for the first time used an economically justified approach in the form of a technical and economic criterion to assess not only the costs of increasing and restoring the surface hardness of parts, but also the method of restoring precision parts of machines and devices. This correlates not only the costs of increasing and restoring the surface hardness of parts, but also the costs of preparing and machining precision parts after increasing and restoring the surface hardness [3, 4, 6 – 9, 11].

Unfortunately, when determining the costs of increasing and restoring the surface hardness of worn precision parts (plungers and dies) of machines and devices in this methodology, mainly current costs are visible, that is, the methodology is suitable for increasing and restoring the surface hardness of wearing parts only in specialized repair facilities of machines and devices [3, 4, 18].

Currently, when solving technical maintenance and service issues of machines and devices, the technical and economic criterion - permissible limit values of resource parameters (limitation of resource costs, etc.) using controlled factors is increasingly used, since this approach is even more important in a market economy.

Therefore, in order to justify the method of increasing and restoring optimal surface hardness, it is necessary to take into account the maximum number of factors affecting the total unit cost and the order of interaction of these factors with each other.

Thus, the solution to the problem of justifying the optimal method for increasing and restoring surface hardness of machines and devices - using controlled factors that have the greatest impact on the costs of the technological process, should be carried out using correctly selected optimization criteria

4. Conclusions

1. Currently, when increasing and restoring the surface hardness of precision parts, serious requirements are imposed on production and repair enterprises by the operator: economic, technological processes, working conditions, environmental safety and energy costs.

2. When optimizing the method of increasing and restoring the surface hardness of precision parts of machines and devices using laser technology, the characteristics of the factors affecting the process and the technological process are poorly correlated. In this regard, there is a need to create a generalized mathematical model for optimizing the technological process of increasing and restoring the surface hardness of precision parts of machines and devices using laser technology, taking into account reliability parameters, environmental safety and economic indicators.

3. When developing a mathematical model, it is more appropriate to evaluate the efficiency of the unit cost of the technological process and to select the optimal method for increasing and restoring the surface hardness of precision parts of engines, machines and apparatuses - to justify the objective function for laser technology.

4. Especially in cases of shortage of spare parts, wear of expensive parts, as well as increasing the wear resistance of the friction pair, a method of increasing and restoring surface hardness may be recommended.

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STATIC ANALYSIS OF CYLINDRICAL SHELL USING THE FINITE ELEMENT METHOD

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ABSTRACT

In the article, two cylindrical shells with different dimensions are statically analyzed using the finite element method. Cylindrical shells with a height to outer diameter ratio (h/d=2) of two and four (h/d=4), and a wall thickness of 1 mm were selected as the research objects. Carbon steel grade 1023 was selected for the cylindrical shells. The cylindrical shells were rigidly fixed on one side (Fixed Geometry fastening type was selected), and free on the other side. A compressive force of 1000N was applied to the inner surfaces of the cylindrical shells. Static analysis was performed using the simulation application in the Solidworks program. As a result of the analysis, the distributed values of the Von Mises stress, normal stress along the x, y and z axes were obtained. At the same time, the distributed values of the total displacement along the x, y and z axes were obtained. The maximum values of von Mises stress, normal stress (along the given axes) and displacement were observed in the sample with a small diameter (d=25mm) and a large height (h=200mm) (h/d=4). A difference of 98% was obtained between the maximum values of stress and displacement in both samples.

2. Introduction

Cylindrical shells are structural elements commonly used in many engineering structures (tanks, pipes, silos, bridges, etc.). These structural elements offer advantages such as lightness and high strength thanks to their thin-walled and curvilinear geometry. However, in order to achieve these advantages, the design and analysis processes of the structures must be carried out correctly. In particular, static analysis plays a critical role in understanding the behavior of shell structures under loading. Recent studies have shown that significant progress has been made in the analysis of cylindrical shells. For example, classical studies by Hutchinson and Koiter have revealed the basic concepts of shell stability [1]. In addition, higher-order shell theories developed by Reddy have enabled more precise analysis, especially of thin-walled structures [2].

In [3] and [4], modeling work using the finite element method was studied. In [5], detailed information was given on the theory of stability of plates and cylindrical shells. Issues of practical importance were solved. In [6], the problem of dynamic stability of cylindrical shells with different resistance to tension and compression was solved. This literature review highlights the scope of methods used in static analysis of cylindrical shells and their importance in engineering applications.

This study focuses on the static analysis of cylindrical shells and examines in detail the fundamental parameters and analysis methods affecting the behavior of these structural elements. Analytical and numerical approaches such as classical shell theory and finite element method are discussed, and the advantages and limitations of these methods are discussed. The study also examines the effects of loading conditions and boundary conditions on the shell behavior and draws attention to the critical factors that should be considered in the design process of these structures. In addition, the effect of changes in material properties and geometric parameters on the analysis results is evaluated.

This study aims to reveal the basic factors to be taken into consideration in structural design by examining the theoretical and numerical methods used in determining the static behavior of cylindrical shells.

3. Materials and Methods

2.1. Design and material selection of cylindrical shells

Three samples were selected as the object of the study. The dimensions of the samples are as follows:

Cylindrical shell 1 (№1): Height h=100mm; outer diameter d=50mm, wall thickness b=1mm;

Cylindrical shell 1 (№1): Height h=200mm; outer diameter d=25mm, wall thickness b=1mm;

3D models of cylindrical covers were drawn in the Solidworks program. For this, a 2D drawing was first drawn using the shell application in the program and converted into a 3D model. (Fig. 1).

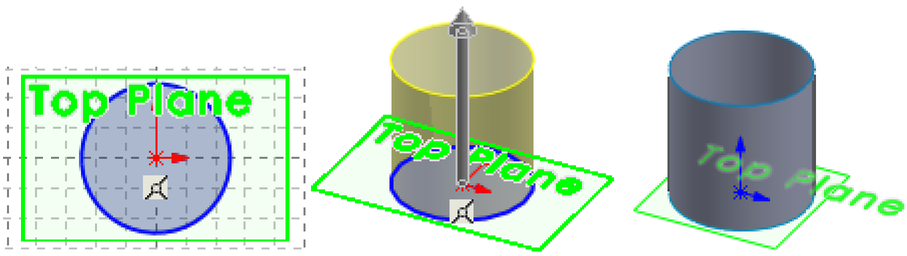


Fig. 1. Steps for drawing a cylindrical shell in Solidworks

Cylindrical shells are components that are often used in structural engineering, the automotive industry, the aerospace industry, or the chemical industry. These shells can be made from different materials, but some of the main materials commonly used are:

1. Steel: Steel is widely used in the production of cylindrical shells due to its durability and strength. It is preferred in applications such as automotive, construction and pipelines.

2. Aluminum: Aluminum is generally used in the aerospace industry and the automotive sector due to its lightness and corrosion resistance. Aluminum alloys are especially preferred for cylindrical structures.

3. Composite Materials: Composite materials, usually cylindrical shells made of carbon fiber or fiberglass, are preferred in sectors with high strength and lightness requirements. It is common in aircraft and high-performance automobiles.

4. Stainless Steel: Stainless steel is used especially in environments requiring corrosion resistance and high temperature resistance. It is frequently used in the chemical industry and energy sector.

5. Concrete: Concrete is used in the construction sector, especially for large cylindrical structures (water tanks, silos, pipelines). Concrete cylindrical structures are generally preferred as structures that carry large loads on the ground.

Each material is selected according to the requirements of a specific application and is usually designed based on engineering calculations. In this article, we select steel grade 1023 from the Soldworks library. The mechanical properties of the selected material are as follows:

Modulus of elasticity: 204999N/mm2;

Poisson's ratio: 0.29;

Tensile strength 425N/mm^2

Shear modulus 79999 N/mm^2

Density: 7858kg/m3;

Yield strength: 282.985N/mm2.

2.2. Static analysis of cylindrical shells

Developing computer technology has made it possible to widely use numerical methods in the analysis of cylindrical shells. One of the most preferred methods is the finite element method (FEM). FEM provides precise solutions for structures with complex geometry and boundary conditions.

In finite element modeling, the shell structure is usually represented by shell elements. These elements are solved by the principle of minimizing deformation energy. The advantages of FEM include:

• Easily model various loading conditions.

• Applicability to complex geometries.

• Precise and detailed analysis.

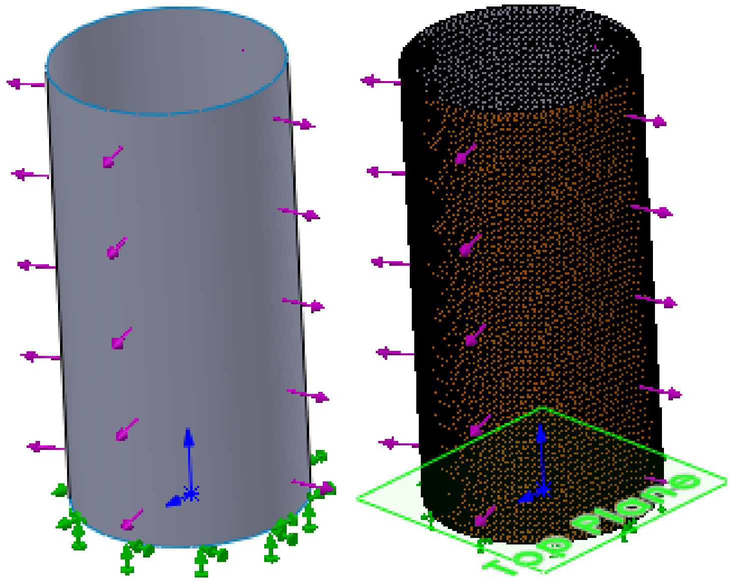
However, FEM requires high computational costs and the solution accuracy depends on the element size and mesh structure.

To start the static analysis of cylindrical shells, the loading conditions and boundary conditions must first be determined. In the static analysis of cylindrical shells, the loading conditions and boundary conditions play a critical role. The most common loading conditions include:

- Internal pressure: Commonly encountered in storage tanks and pipes.
- Axial load: Observed in silos and column-like structures.
- External pressure: Occurs in vacuum conditions or submarine structures.

Boundary conditions vary depending on whether the structure is fixed, simply supported or freely supported. Correct definition of these conditions in the analysis is of great importance in terms of solution sensitivity.

In the considered work, a static analysis of cylindrical shells is performed for axial loading conditions. In the static analysis, 2 samples with the dimensions specified in section 2.1 were selected. We assume the load amount to be 1000N. The plate is rigidly fixed on one side and free on the other. The cylindrical shells are subjected to a compressive force of 1000N from the inside. When performing an analysis using the finite element method, the mesh sizes must also be determined. The mesh sizes are an important factor affecting the accuracy of the analysis results. In this regard, we select a fine mesh with dimensions of 1.50mm and 0.075mm. Steel grade 1023 was selected as the material from the Solidworks library. The application of the aforementioned boundary conditions on a 3D model is shown in Figure 2.



a) b)

Fig. 2. Application of boundary conditions and mesh properties:

a) boundary conditions; b) application of mesh parameters

Within the given boundary conditions, as a result of static analysis, we obtain the distributed values of Von Mises stress, normal stress and displacement along the surface of the cylindrical shell (in the x, y and z axes). The distributed values of Von Mises stress and normal stress for Examples 1 and 2 are given in the figures below.

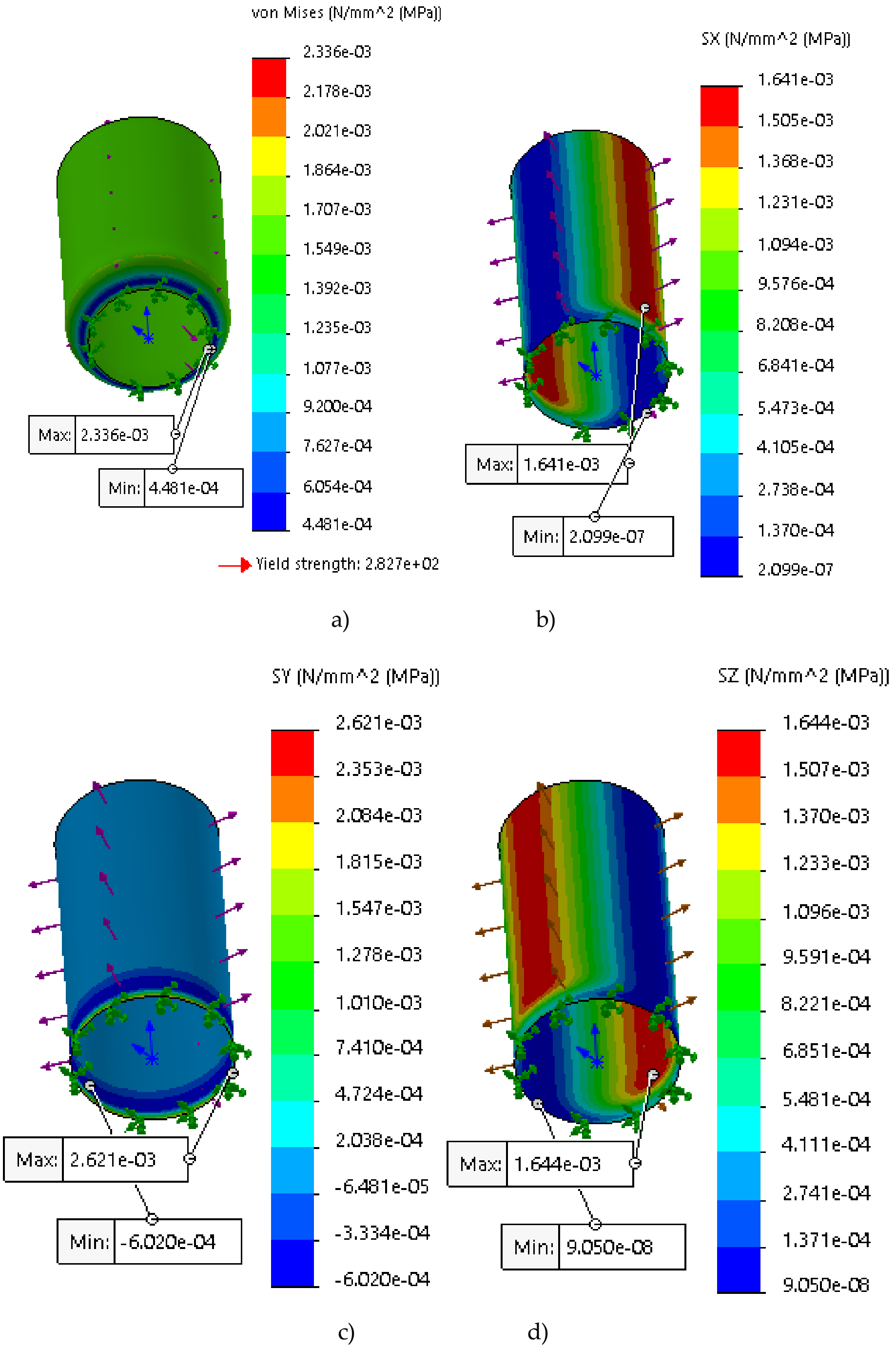


Fig. 3. Stress distribution values along the surface of the cylindrical shell for example 1:

a) Von Mises stress distribution values;

b) Normal stress values along the x axis;

c) Normal stress values along the y axis;

d) Normal stress values along the z axis.

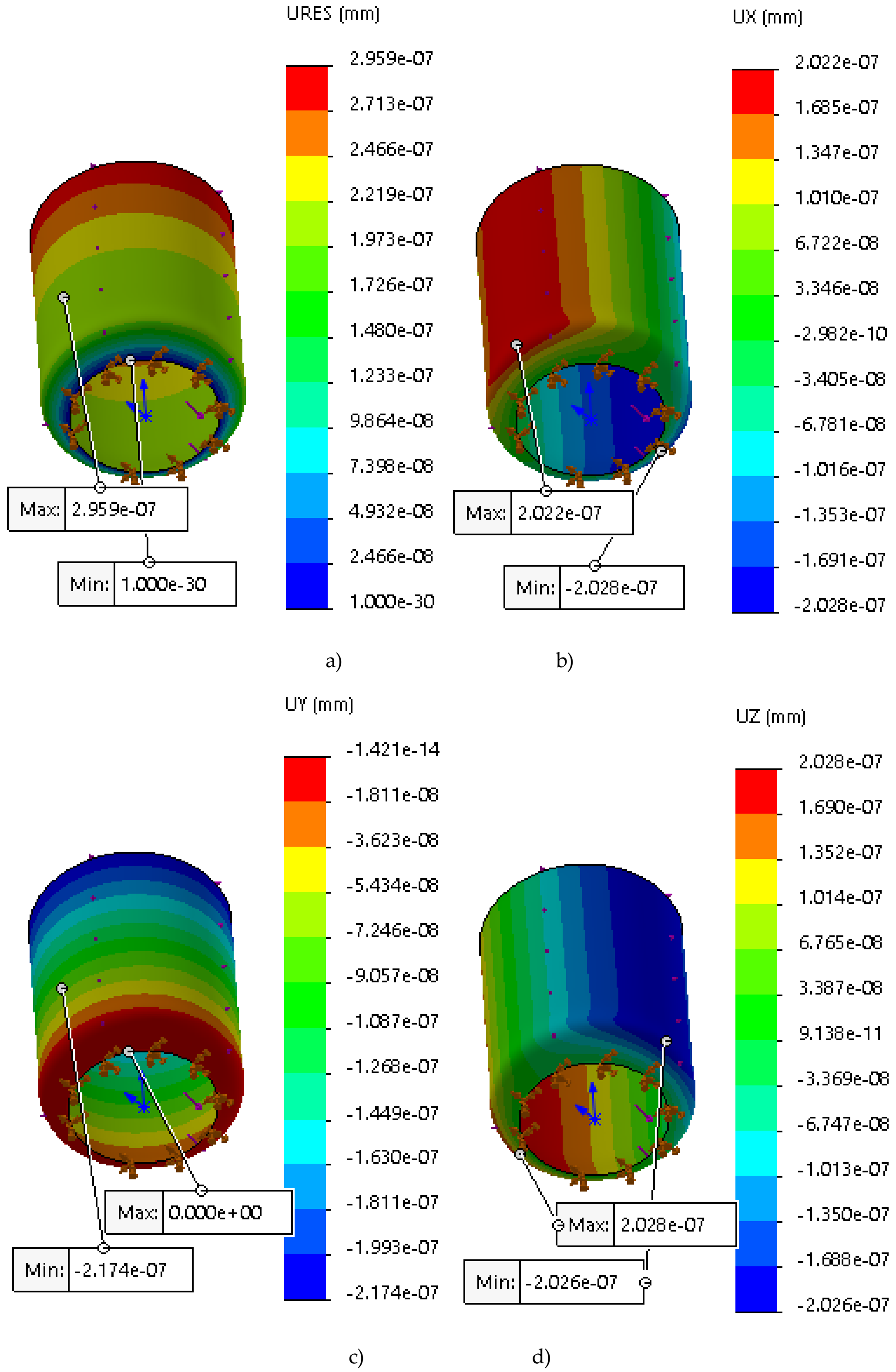


Fig. 4. Displacement values distributed along the surface of the cylindrical shell for example 1:

a) Displacement values distributed along the total displacement;

b) Displacement values along the x axis;

c) Displacement values distributed along the y axis;

d) Displacement values distributed along the z axis.

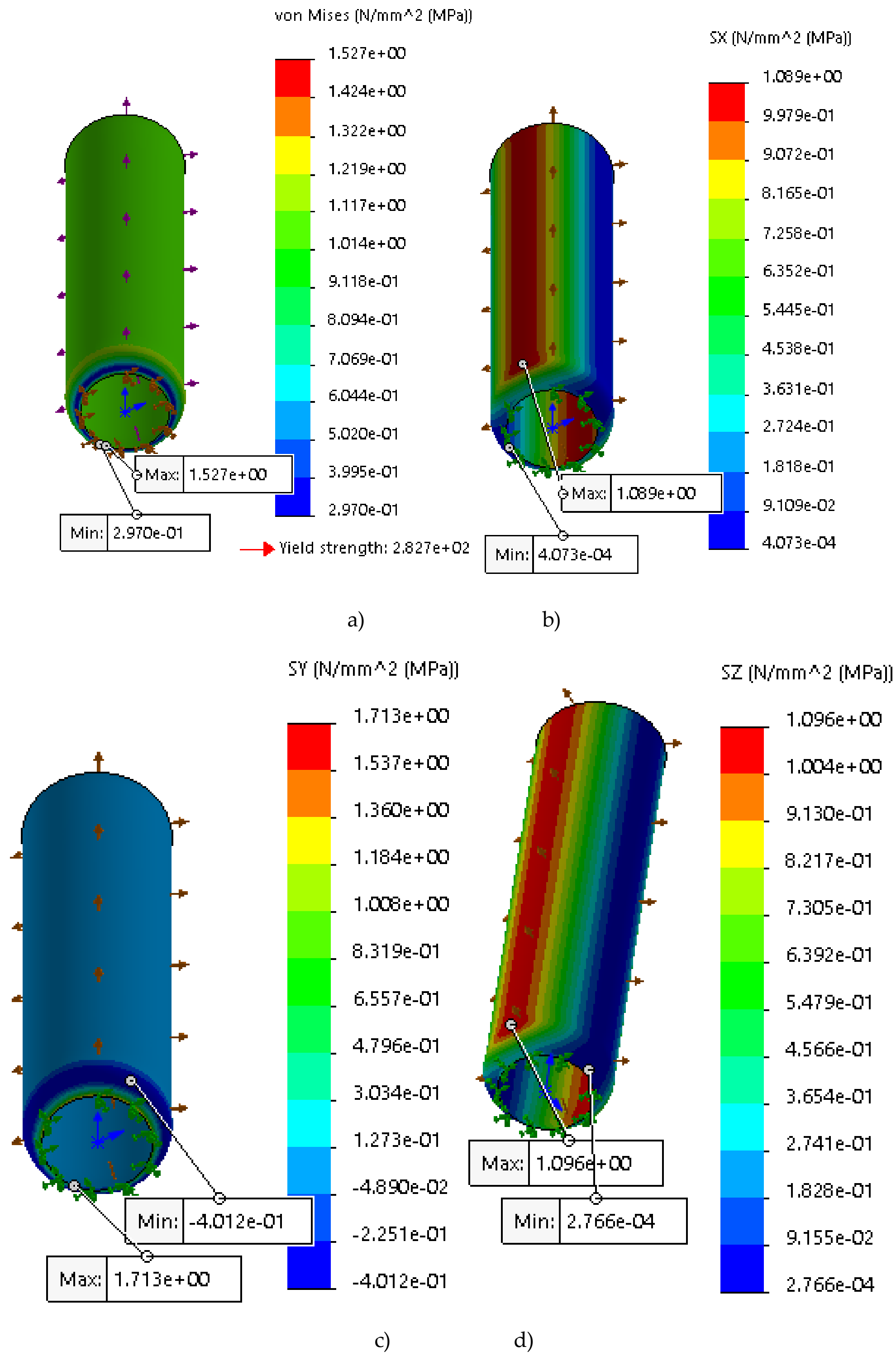


Fig. 5. Stress distribution values along the surface of the cylindrical shell for example 2:

a) Von Mises stress distribution values;

b) normal stress values along the x axis;

c) normal stress values along the y axis;

d) normal stress values along the z axis.

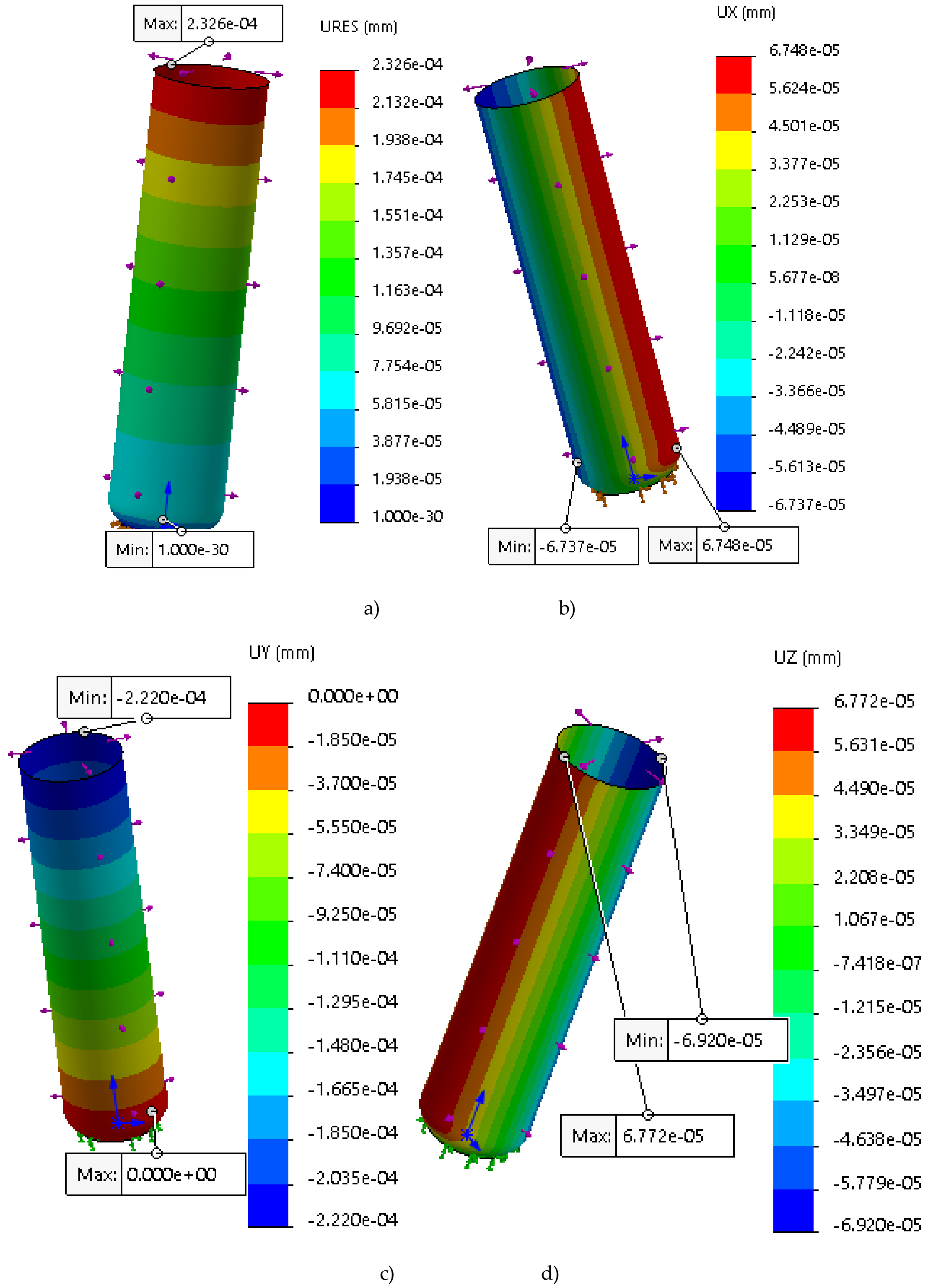


Fig. 6. Displacement values distributed along the surface of the cylindrical cover for Example 2:

a) Displacement values distributed along the total displacement;

b) Displacement values along the x axis;

c) Displacement values distributed along the y axis;

d) Displacement values distributed along the z axis.

4. Results of static analysis

In Example 1, the maximum value of the Von Mises stress was 0.002336N/mm2, and the minimum value was 0.00048N/mm2. The maximum value of this stress was observed in the fastening zone of the cylindrical shell, and the minimum value was observed in the zone relatively close to the fastening zone. The maximum value of the normal stress along the x axis was 0.00164N/mm2, and the minimum value was 2x10-7 N/mm2, and the y and z axes were 0.00262N/mm2, 6x10-4N/mm2, and 0.00164N/mm2, 9x10-8 N/mm2, respectively. The maximum value of the stress along the x and z axes was distributed symmetrically along the surface of the cylindrical shell in the axial direction. The minimum value of the stress was observed in other zones in the axial direction (Fig. 3).

The maximum value of the total displacement in sample 1 was 2.9x10-7mm, and the minimum value was 1x10-30 mm. The maximum value of the displacement in the x-axis direction was 2.02x10-7mm, and the minimum value was -2x10-7mm, and the values along the y and z axes were -1.4x10-14mm, -2.174x10-7 mm and 2x10-7mm, -2.02x10-7mm, respectively. The maximum value of the total displacement was observed on the top surface of the sample, and the minimum value was observed at points close to the specimen's mounting zone. The maximum value of the displacement along the x and z axes was observed along the cylinder's surface in the axial directions (Fig. 4).

In sample 2, the maximum value of the Von Mises stress was 1.527N/mm2, and the minimum value was 0.297N/mm2. The maximum value of this stress was observed in the fastening zone of the cylindrical shell, and the minimum value was observed in the zone relatively close to the fastening zone. The maximum value of the normal stress along the x axis was 1.089N/mm2, and the minimum value was 4x10-4 N/mm2, and the values along the y and z axes were 1.713N/mm2, 0.04N/mm2, and 1.096N/mm2, 2.7x10-4 N/mm2, respectively. The maximum value of the stress along the x and z axes was distributed symmetrically along the surface of the cylindrical shell in the axial direction. The minimum value of the stress was observed in other zones in the axial direction (Fig. 5).

The maximum value of the total displacement in sample 1 was 2.32x10-4 mm, and the minimum value was 1x10-30 mm. The maximum value of the displacement in the x-axis direction was 6.7x10-5 mm, and the minimum value was -6.7x10-5 mm, and the values along the y and z axes were -2.22x10-4 mm, 6.77x10-5 mm, and -6.9x10-5 mm, respectively. The maximum value of the total displacement was observed on the top surface of the sample, and the minimum value was observed at points close to the fastening zone of the sample. The maximum values of the displacement along the x and z axes were observed along the surface of the cylindrical shell in the axial direction. The displacement values in the tension and compression regions along the surface of the sample were almost equal (Fig. 6).

5. Conclusion and discussions

Static analysis of cylindrical shells requires a combination of both theoretical and numerical methods. Classical shell theory provides a simple and effective approach, while numerical methods such as FEM provide a suitable option for more complex cases.

In the article, a static analysis was performed on a cylindrical shell with two different dimensions (h/d=2, h/d=4). Soldworks software was used to perform the static analysis. The following results were obtained.

1. The largest value of the von Mises stress was obtained in sample 2. The difference in value was 95%. The smallest value was observed in sample 1. In both samples, the largest value of the von Mises stress occurred at points close to the fastening zones of the cylindrical shell.

2. The largest values of normal stress along the x, y and z axes were observed in sample 2, and the smallest values were observed in sample 1. There is a difference of 98% between the largest values of stress along the x and z axes, and 99% between the largest values of stress along the y axis. In both samples, the normal stress along the x and z axes is distributed in the tensile zones along the surface of the cylindrical shell.

3. The maximum value of the total displacement occurred in sample 2, and the minimum value occurred in sample 1. There is a 99% difference between the largest values of displacement for both samples. The largest values of displacement along the x, y, and z axes were again observed in sample 2. There is a 98% difference between the values of displacement along the x and z axes. In both samples, the values of displacement in the tension and compression zones along the x and z axes are equal.

As can be seen, based on the stress and displacement values, the most effective cylindrical shell is considered to be example 1. That is, as the diameter of the cylindrical cover decreases and its height increases, the stress and displacement values increase.

This study aims to contribute to the development of more effective methods in the design and analysis processes of cylindrical shells. Future studies could examine the integration of innovations in materials technologies and AI-based analysis methods into this field.

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STUDY OF CHANGES IN THE STRUCTURE OF STEELS DURING THERMOMECHANICAL TREATMENT

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ABSTRACT

Thermomechanical processing is a combination of the operations of deformation by heating and cooling (in different sequences), as a result of which the formation of the final structure of a metal alloy, and consequently its properties, occurs under conditions of increased density and the corresponding distribution of structural imperfections created by plastic deformation. Therefore, firstly, research in the field of thermomechanical processing is reduced to studying the effect of plastic deformation on transformations in heat-treated alloys and on the structure in properties after these transformations. Secondly, thermomechanical processing is advisable in all cases where heat treatment of metal alloys is effective. Phase transitions during heat treatment and plastic flow occur as a result of the restructuring of the same atoms, connected not only by general regular structures, but also by certain, also to a certain extent regular, deviations from these structures, the main ones of which are dislocations.

Introduction

Thermomechanical treatment, which has been developed in recent years, allows achieving strength values for technical alloys that are greater than those achieved by alloying and conventional heat treatment. With comparable strength, thermomechanical treatment determines a higher level of plasticity and viscosity than alloying and thermomechanical treatment. Thermomechanical treatment determines a unique combination of increased strength and increased resistance to destruction, which creates better structural strength for real products.

One of the features of the thermomechanical method of hardening is that it bridges the gap between pressure metal treatment and heat treatment. In this case, both of these factors affecting the structure and properties of metal alloys are combined, creating continuity in the technological chain of product manufacturing [1].

Depending on the nature of the alloy, certain schemes of thermomechanical treatment are used, in particular, the variety of options of which is determined in particular by the variety of possible transformations. Identification of the most promising areas of use of thermomechanical treatment is possible with a relatively wide.

Research, when using most modern means of studying the structure and testing properties

As has been said, the use of thermomechanical processing is effective in cases where heat treatment is advisable. This is determined by the fact that the processes of phase and structural transformations occurring during heat treatment are significantly influenced by structural imperfections, the density of which increases as a result of plastic deformation. At the same time, as a result of some processes of phase and structural transformations, a new number of imperfections is formed. Thus, the kinetics and mechanism of phase and structural transformations during thermomechanical and heat treatment depend on the type and density of structural imperfections, and in turn, these transformations affect the number and distribution of imperfections [2].

With regard to the known (traditional) methods of heat treatment that create a complex of high mechanical properties - martensite quenching, isothermal transformation, dispersion hardening - the use of increased density and the corresponding distribution of structural imperfections in the process of structure formation has already brought significant results.

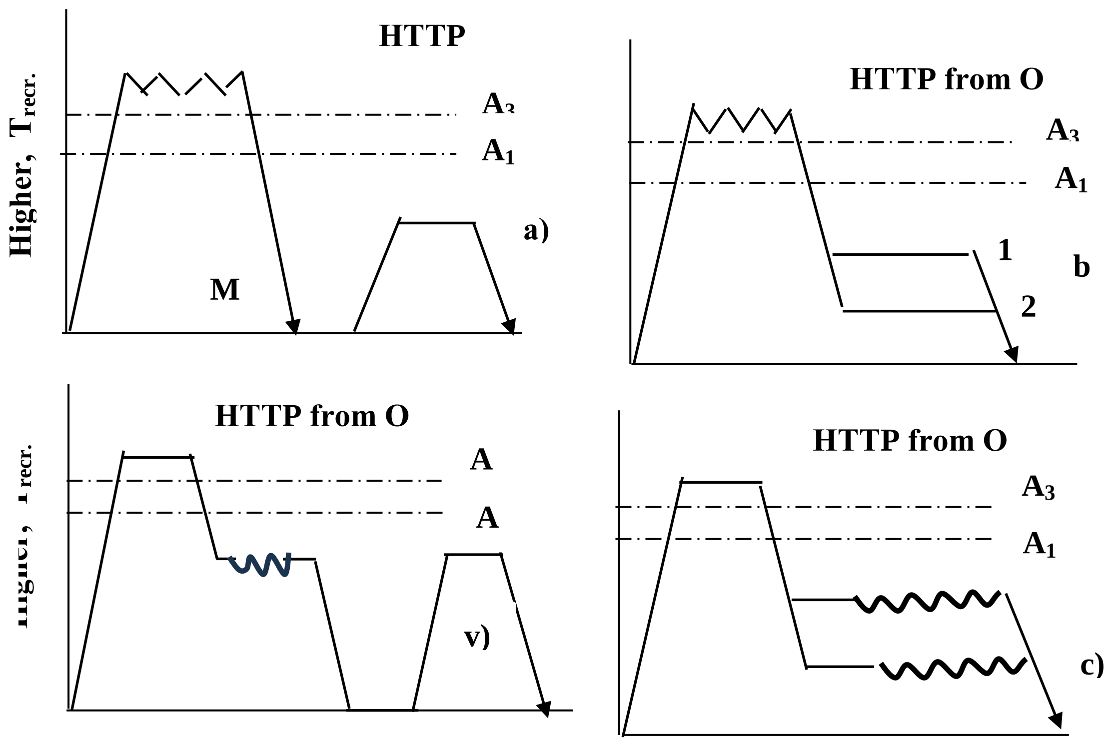

Fig. 1. Scheme of thermomechanical processing processes (meaning the main processing operations that form the final structure of the alloy, which does not change its type in the future).

a – **HTTP** with martensitic hardening and tempering; **b** - **HTTP** surface layers of parts; **v** - high-temperature thermomechanical isothermal treatment with decomposition in the pearlite (1) or bainitic (2) region; **c -** low-temperature thermomechanical treatment (ausforming);

This was manifested, in particular, in the creation of various schemes of thermomechanical treatment, determined by the variety of possible transformations. It seems appropriate to compile a classification of various methods of thermomechanical treatment, choosing in accordance with the stated principles as a classification feature the sequence of the deformation and heat treatment operation (see the diagram in Fig. 1) state if the planes of easy shear are oriented in the direction of maximum shear stress. High temperature thermomechanical treatment (HTTT).

Naturally, each type of thermomechanical In the process of thermomechanical treatment of steels hardened for martensite, all known methods of strengthening are used and enhanced: an increase in the density and, accordingly, the interaction of dislocations as a result of deformation of austenite and phase work hardening during martensitic transformation: a change in the composition of martensite due to the influence of deformation primarily on the redistribution of carbon and on the morphology of martensite crystals: the creation of more dispersed and more dispersed and more uniformly distributed particles of strengthening (carbide) phases at low tempering of martensite, while the martensite crystals themselves are crushed after TMP. Apparently, the complex of high mechanical properties achieved as a result of thermomechanical treatment is determined by a change in many elements of the structure and substructure [3,4].

The mechanical properties of some steels after high-temperature thermomechanical processing are shown in Table 1

Table 1. Mechanical properties of some steels after high-temperature thermomechanical treatment

Thermomechanical processing	**$\sigma_в$, кг/мм²**	**$\sigma_т$, кг/мм²**	**δ, %**	**Ψ, кг/мм²**	**$a_н$ kг·m/sm²**	**HRC**
Compressed air cooling in high temperature thermomechanical processing						
400	128 (113/44,5)	98 (92/103)	11.6 (10,3/13,5)	44,5 (41/47)	6,3 (5,7/6,8)	36
500	112,5 (107/1162)	86,5 (84,5/88)	12,4 (10,8/14)	55 (53,5/57)	9,7 (9,0/10,2)	29,5
600	100 (98,5/103)	82 (80,5/83)	13,6 (12,4/14,5)	58 (56,5/62)	10,4 (9,9/11,2)	28,5
Note: Upper and lower property limits are given in brackets in the numerator and denominator, respectively						

Processing is characterized by the formation of a specific structure and fine structure of steel.

The increasing density of dislocations during the A→M transformation in combination with the low solubility of carbon in the α-lattice leads to an increase in the concentration heterogeneity of martensite.

X-ray structural studies using a mathematical method to analyze the shape of the diffraction line allowed to determine the change in the composition of martensite to carbon after ordinary quenching and HTTP. It is established that as a result of HTTP, the amount of low-carbon martensite increases and the degree of tetragonality of high-carbon martensite increases at the stage of its two-phase decomposition. Compared to conventional quenching, HTTP leads to a greater degree of two-phase decay of martensite and thus to a greater relaxation ability of the martensitic structure. The resulting increased volume of low-carbon martensite in the steel is more evenly and more dispersedly distributed in the structure, which is a consequence of the influence of the most polygonized dislocation substructure of the hot-deformed austenite. These substructural features, in turn, determine a unique combination of properties after HTTP, when the resistance to brittle and viscous destruction increases simultaneously with the increase in resistance to plastic deformation. The high thermal stability of the created substructure with small-angle, fixed segregation sub-boundaries determines the possibility of regulating the mechanical properties during subsequent heat treatment operations [5].

The main mechanisms of the braking of dislocations in steel and alloys:

a) formation of clusters (segregations) of carbon atoms and degenerate elements (or vacancies) around dislocations in solid solutions;

b) increase in the density of dislocations, which leads to an increase in the interaction of moving dislocations, when the tension field around some dislocations prevents the movement of others;

в) the formation of barriers for moving dislocations in the form of interface surfaces (different types of boundaries) in crystals or particles of the second hardening phase, i.e. creation within the raft of volumes with different crystallography of sliding dislocations;

д) the creation of ordered (by composition or by crystallographic orientations) atomic structures, when moving through which dislocations spend part of their energy on performing processes of disordering - disordering, which leads to their braking. One of the structures formed after high-temperature thermomechanical processing is presented in Figure 2.

It would seem that, having such a wide arsenal of mechanisms for restraining dislocations, it is possible to achieve the creation of such a structural state in metallic alloys, in which the mobility of dislocations at the highest loads will decrease sharply, and the resistance to plastic deformation will correspondingly increase significantly.

Fig. 2. Structures formed after high-temperature thermomechanical treatment

The danger of such clusters is determined by the fact that they create a region in the metal with a very high concentration of stresses, reaching theoretical strength. In this case, it is possible to either relax the dangerous nick tensions by "breaking through" the barrier and relay dislocations to adjacent volumes, or the formation of the embryo of a brittle crack (detachment). The latter is also the mechanism of relaxation of dangerous peak tension, but by the exit, dislocation into the cavity of the crack formed [6].

At the same time, the alloy will have high values of pdastichnosti and viscosity of destruction. As convincingly shown by studies of recent years, such a structure in metal alloys, which are in a high-strength state, when semi-permeable barriers are formed, is created during unused thermomechanical processing.

The first scheme differs in relative simplicity for practical implementation.

Its disadvantage is the danger of strong recrystallization development due to high-temperature deformation carried out at the quenching temperature. In products of a significant section, the necessary changes in the structure are saved with effort. However, the use of special methods of deformation (for example, with a low speed) allows to eliminate this drawback to a certain degree. The advantage of processing according to this scheme is that the fragmented substructure created as a result has high mechanical and thermal resistance.

The second scheme essentially provides for polygon processing. Due to the relative complexity of this scheme, special techniques are required when implementing it on concrete details: in addition, the increase in service properties is observed only at relatively low temperatures [7].

The fragmented structure does not disappear, then high mechanical properties are preserved, so a short softening vacation, during which recrystallization is excluded, will lead to the disintegration of martensite (and will make possible mechanical processing, for example, by cutting) , but will not cause a significant decrease in the density of imperfections and destruction of the dislocation structure (due to the absence of migration of high-angle boundaries), characteristic of the development recrystallization. The subsequent high-speed heating under quenching with short-term exposures will lead to the transition of the α-phase with an increased density of dislocations to the γ-phase, which will also have a high density of them (the mechanism of inheritance of dislocations will be implemented here, as well as during the transition of metal from g.t.c. to[8-11]. After the final quenching, martensite is formed, which preserves (to one degree or another) the additional density of dislocations, and most importantly, to one degree or another, preserves the fragmentation that was immediately after TMO, which determines the "restoration" of the high mechanical properties that were obtained as a result "Private" TMO (effect "following").

Thus, the high-temperature deformation of austenite, accompanied by polygonization processes, leads to the formation of a stable fragmented substructure, and the martensite formed during quenching inherits the specified structural changes. In it, a thin submicroscopic delamination is created on carbon, which determines its increased resistance to deformation and plasticity (increased plasticity allows to realize high strength of low-tempered martensite, therefore the maximum hardening effect of HTTP during relatively tough tests, (for example, on tension) is observed at low tempering temperatures. However, when moving to an even tougher test method (at low temperatures, impact with a crack, etc.), an effect is found HTTP and after high tempering) in comparison with the martensite obtained by conventional quenching.

There are many experimental proofs of the preservation of the stable substructure created by HTTP during high-temperature tempering and subsequent heating under quenching and phase recrystallization. Changes in mechanical properties are determined not only by the dislocation structure, but also by changes in the degree of its interaction with carbon at various stages of processing. When the annealing temperature increases after HTTP, carbon atoms leave the martensite lattice, forming carbides, and the connection with dislocations weakens. When, upon subsequent heating to the austenitic state and rapid cooling, a sufficient amount of carbon is fixed in the solid solution, its interaction with the dislocation structure appears again, causing a corresponding change in mechanical properties

The effect of inheritance also manifests itself after cold hardening with subsequent heat treatment that does not cause recrystallization, which is achieved with rapid and short-term heating. This was the main technological scheme of preliminary thermomechanical treatment (PTMT), the feasibility of which has been confirmed by many studies.

The above methods for creating a high-strength state are carried out on steel of optimal chemical composition with a regulated method of its production, ensuring high metallurgical quality. Rational alloying of high-strength steel is of independent importance. Data on the effect of thermomechanical hardening on the structure and properties of various types of metallurgical products and specific steels are presented in the sections of this volume. The theoretical foundations of thermomechanical treatment are presented in the section.

3. Conclusions and Discussions

1. Positive changes are noticeably felt in parts subjected to thermomechanical treatment
2. During deformation, new separation boundaries are formed in austenite grains, which depends on the degree of deformation and temperature after thermomechanical treatment
3. The decrease in the carbon content (compared to the thermal treatment processing mode) increases the degree of deformation during high-temperature thermomechanical treatment
4. The mechanical properties of the plates after low and high-temperature thermomechanical treatment have increased in a complex way.
5. As a result of the high-temperature thermomechanical treatment process, not only the mechanical properties of the steel increase, but also a significant change in the shape and size of the grain is observed

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ANALYSIS OF THE FUTURE PERSPECTIVES OF AIR FLOW ENERGY İNSTALLATİONS

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ABSTRACT
The article provides information on the creation and study of energy systems based on the power of air flows with maximum use of advantages in processes associated with nature. An analysis of available data in this area is carried out, the advantages and disadvantages of energy installations are indicated. Based on the analysis and theoretical considerations, the indicators of the model for improving the energy system using the power of the air flow, including economic efficiency, are analyzed. Changes in influencing parameters to improve the efficiency of using the design of the energy installation are described, a comparative analysis is given.

Introduction

The ability to use the power of nature to benefit is an indicator of the development of human consciousness. In particular, this can be attributed to the use of wind energy by people for their own needs. At the dawn of science, mankind did not have the slightest knowledge of the physics and movement of air masses across the plane of the earth's surface, but in time, man learned to use the power of the wind as a traction force for navigation on water. In addition, the natural continuation of the development of scientific thought was the appearance of wind turbines or windmills.

Wind power plays an important role in the global transition to sustainable energy sources. As a form of energy, it uses the kinetic energy of air flow to generate electricity and offers several advantages that make it an important component of renewable energy. However, like all energy sources, air flow energy has its problems and disadvantages. Airflow energy is considered to be an industrial and environmentally loyal (ecologically clean) source of electricity because it does not produce any greenhouse gases or air pollutants during its operation. For example, the variability and intermittency of wind power require effective integration with energy storage systems or other energy sources to ensure a reliable and continuous energy supply [1].

Its many advantages, from environmental loyalty to low operating costs, make it a key player on the path to a cleaner energy future, because:

• it is renewable and sustainable

• it is ecologically clean

• its operating costs are low

• its water demand is low

• it has energy efficiency and the ability to use land efficiently

However, there are also disadvantages of wind energy. Among them, in particular, the following can be mentioned:

• Intermittency (because the wind speed is unstable)

• high initial investment cost

• visual and sound impact effectiveness

• the risk of affecting the environment

As we have seen, even if wind energy is considered to be environmentally loyal and renewable, it has its problems and drawbacks. Advances in planning and technology are needed to address these issues and effectively integrate wind energy into our supply grid systems [2].

The next global advance in the process of weather and wind control was at the end of the 19th century, when the first wind power installation was built. The problem that led to the search for an alternative source of energy is the desire to save financial resources, because every year an increase in the prices of other fuel resources was observed.

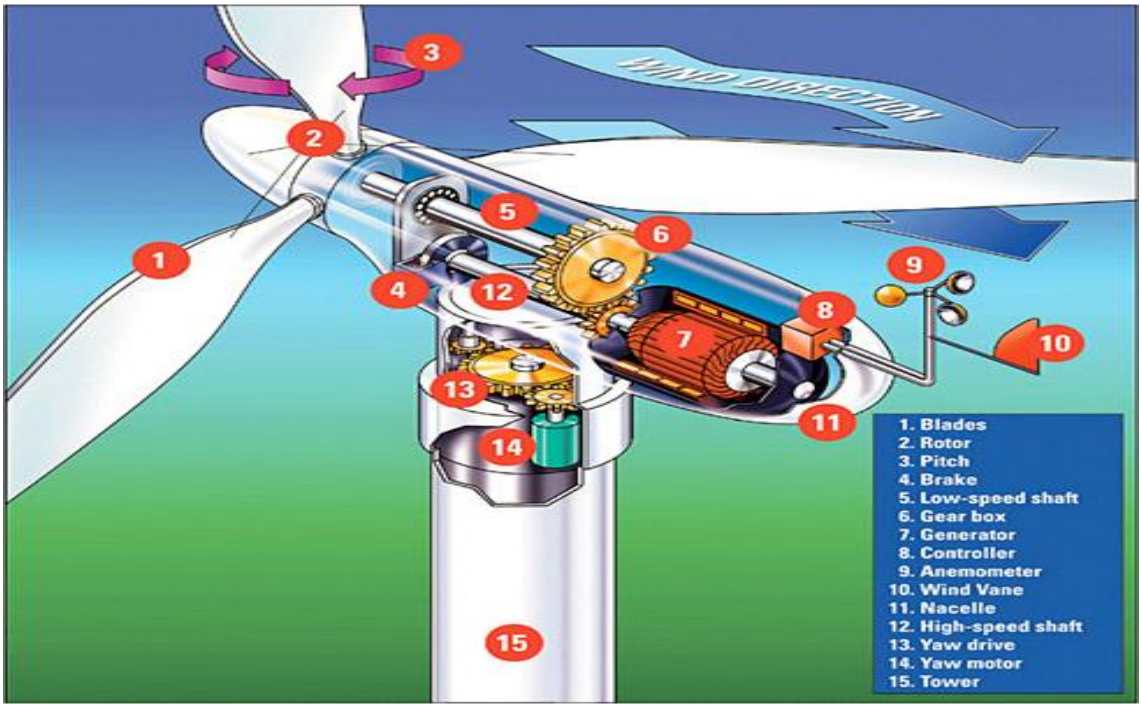

Fig. 1 Wind generator construction scheme.

It is necessary to approach the issues related to the topic in a complex way. There should be a single criterion for determining the power of air flow (streams), so that it can be applied universally (with mutual conversion to other units) in each case.

Often, special terminology is used when reporting on forecasts of wind formation as a result of the movement of air masses (flows), for example: *strong wind, moderate wind, thunderstorm, hurricane, etc.*

It should be noted that the wind speed is measured by the 12-point Beaufort scale, and the corresponding score is determined by quickly comparing the information about the predicted increase of the wind (for example, a speed of 20.8-24.4 m/s corresponds to a "storm" wind of 9 points).

In fact, one parameter is taken into account here, and that is the wind speed (another criterion - the score is conventionally connected to the speed interval and called appropriate). At the same time, the average wind speed on the Beaufort scale is shown at a standard height of 10 meters above an open, flat surface (and wind generators are installed at heights up to and above 50m) to estimate the effect on terrestrial (land) objects or waves in the open sea.

However, as we mentioned, it is practically impossible to determine the effectiveness of air flows based on the existing scale. Here, in addition to the speed of the air flow, its density and pressure on the surface must be taken into account, otherwise the final benefit of air flow energy installations will be very low. For this, a single criterion should be defined, which includes the three parameters mentioned above.

Therefore, first of all, a generalized universal criterion should be determined by conducting theoretical analyzes of the researched field. Based on real (factual) information, that criterion should include such parameters as wind speed, wind direction, impact force on the surface area of the relevant element, air flow regimes, and duration of wind action (generalizing correlation coefficient).

In the second stage, after the relevant information (forecast) received via satellite is processed in a special program in real time (online), based on the received data, comparing it with the above-mentioned criterion, determining the selection of the operation (work) mode of the wind power installation.

Then (in the third stage), achieving a slight increase in the useful work coefficient (UWC) by making minimal structural changes (additions) to ensure the maximum efficiency of the wind power installation in the selected operating mode.

Methods

By analyzing the procedures carried out in connection with the researched topic, certain conclusions can be reached to improve the useful service life (or the useful work coefficient - UWC) in the process of obtaining energy acquisition of air-flow machines and mechanisms.

Thus, today, the use of air flow generators is a very common way to produce electricity, and a modern wind turbine is known in every research area (Figure 1). The leaders in terms of the number of used wind power plants are the United States of America and China, but already other countries also understand that the advantage of wind power plants is the possibility of obtaining cheap energy, and this energy sector is developing seriously [3].

The instantaneous power of the flow (wind) turbine can be calculated as follows (W):

$$P = A \cdot (\rho s v^3 / 2) \quad (1)$$

where, A – coefficient of wind energy use;

ρ – air density, $\rho = 1.18$ kg/m^3;

S – air flow cross-sectional area, m^2;

υ – wind speed, m/s.

In the calculations, $A = 0.30 \div 0.45$ is assumed for wind energy installations. The annual production capacity of wind turbines (kW) is determined by the following formula (kW):

$$P_{annual} = 8{,}76P \quad (2)$$

However, the value of the annual energy produced is equal to:

$$D_e = P_{annual} \cdot b \quad (3)$$

where b is the price of electricity (determined per kW) [4].

The self-correction period of wind energy (wind-flow power installation) is calculated as follows:

$$T = D_i / D_e \quad (4)$$

where D_i is the value of the wind-flow power installation.

As a result of the analysis of existing studies related to the mentioned issue, it can be concluded that most of the wind turbines in operation in the world cannot produce more than 4% of their declared power when the annual average wind speed is 3 m/s. This, in turn, leads to the estimated payback period of wind turbines, based on the considered conditions (taking into account the coefficient b, which represents the price of electricity), approximately from 50 years to 150 years [5,6].

Investigations and calculations also show that even if the average annual wind speed is doubled, i.e. 6 m/s, it will be able to produce an average of 20% of its declared power, and the payback period in this case will be 10 to 50 years depending on the conditions considered.

As can be seen from the formula (1), among the parameters that affect the increase in the power of the wind-flow power installation (taking into account the limitation of the quantity ***S***) is the ***A*** coefficient.

Despite the possibility of predicting the wind speed and direction during the year, it is very difficult to build a scenario in advance for the processes occurring in nature, and the probability of accuracy is low. Therefore, increasing the coefficient of use of wind energy is an important issue.

Approaching the solution of the issue in a complex way, it can be concluded that the improvement of the technical indicators of wind turbines (as well as the UWC) plays one of the main roles.

At the same time, it should be noted that the construction changes currently proposed by researchers are aimed at making them more complicated and expensive. As a result, the work carried out increases the quality indicators of the devices, but also increases the costs of the production and maintenance of the equipment, making the already quite long-term self-correction coefficient completely unacceptable [7,8].

In this case, it would be more appropriate to make "slight" changes to the air flow energy installations, for example - from improved multipliers, to the introduction of additives (confusers) that increase the concentration of the air flow.

As a result, it could be concluded that

- the formation of a single coefficient by combining the main parameters characterizing the air currents affecting wind power plants in a row will make it possible to select any operation in this field or perform calculations in the correct format in the future (scientific and practical significance is also shown in the implementation stages of the project);

- it is possible, taking into account advanced technologies for transmitting data via satellite, determining the main parameters (wind speed, direction, etc.) based on software analysis with predicted weather changes (meteorological conditions) and preselecting the optimal operating mode for the entire complex of wind-flow power installation in a certain area;

- wind energy contributes to a greener (environmental) and sustainable energy future. Its renewable, low-cost and environmentally loyal (ecologically clean) properties make it an attractive option for reducing greenhouse gas exports and increasing energy independence. However, issues such as business interruption, visual light-shadow and noise issues and impact on wildlife need to be addressed. The role of wind power in the wider energy complex will depend on effective mitigation of these challenges, technological progress and public acceptance. Wind energy remains a valuable solution to the development of a cleaner, more stable world.

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ON THE CALCULATION OF BRAKING TORQUES IN DRUM SHOE BRAKE MECHANISMS

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ABSTRACT

Friction brakes generate braking torque through the friction force between rotating (disk, drum) and non-rotating (pad, band) components. Drum-type brakes are primarily used in heavy-duty vehicles. To initiate braking, the shoe is pressed against the drum, creating contact pressure between the pressed surfaces. Tangential stresses arising from drum rotation produce the braking torque.

The braking torque in the brake mechanism depends on the contact pressure and the coefficient of friction between the compressed surfaces. With a large compressive force, the deformations of the drum and shoe differ, altering the drum's round cross-section and causing variations in braking force at different moments. Therefore, accurately determining contact pressure is crucial for drum brakes.

This article focuses on calculating braking torque by determining the deformations of the mold as close to real-world conditions as possible.

Introduction

In existing methods of calculating brake mechanisms, four laws of pressure forces are accepted a priori: constant $g = const$, sinusoidal $g = g_{max} sin\theta$, cosine $g = g_{max} cos\theta$ and square sinusoidal $g = g_{max} sin^2\theta$ by the length of the block. There is currently no single method for calculating brake parameters.

The pressure is distributed along the length of the shoe according to a sinusoidal law if the friction drum and brake shoes are absolutely rigid, the friction lining is ideally adjusted to the drum, and the deformation of the friction lining obeys Hooke's law. At high compression force, the deformations of the drum and shoe differ, changing the circular cross-section of the drum and causing changes in the braking force at different moments. The sinusoidal distribution law is typical for service braking; during braking with greater intensity, due to the increase in deformation of the shoes and the drum, which acquires an oval shape, it is distorted and approaches uniform [1].

The opposite of this is the cosine law, when the concentration of the specific load is observed in the middle part of the brake shoe, and an equal decrease in load is observed towards the edges of the advancing and escaping parts of the brake shoe. According to the statements presented in the work, the cosine law reflects the ovality of the rotating brake drum, which negatively affects the wear of the middle part of the shoe.

Determination of contact pressure

In fact, the true distribution of contact pressure is unknown in advance and must be determined from the solution of the contact problem for the lining - drum system. The material of the drum and the friction lining of the brake shoe is modeled by an elastic medium, the mechanical characteristics of which will be E_1, μ_1 and E_2, μ_2, respectively.

For any law of pressure distribution, the braking moment is determined through the resultant force of all elementary forces applied at a point whose coordinates are determined by the reduced radius and angle.

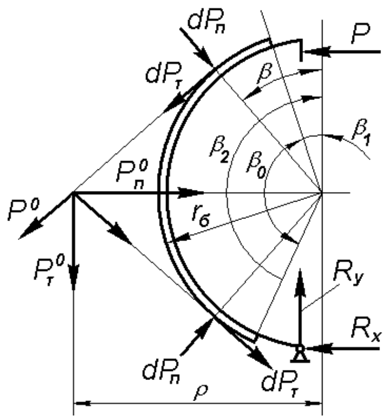

Figure 1. Diagram of forces acting on the shoe of a drum brake mechanism: where p – is the pressure on the linings; dF – is the elementary area of the lining; b is the width of the lining; r_b –is the radius of the drum; β – is the angular coordinate of the elementary area

The shoe is pressed against the brake drum under the action of force P. When the drum rotates in the direction indicated by the arrow, interaction forces arise between the drum and the shoe lining: the elementary normal force dP_n and elementary tangential force dP_τ.

To determine the total braking torque created by the shoe, it is necessary to know how the pressure changes along the length of the lining. As seen in the diagram, the resultant friction force (conditional) P_n^0 acts at a radius ρ, which depends on the angle β_0 = 90 – 120°. In calculations of braking torque, the resultant friction force is usually reduced to the radius of the brake drum, allowing the use of simplified formulas. However, this approach provides approximate values, which are insufficient for designing brake mechanisms. Therefore, we strive to determine the contact pressure more accurately by considering deformations and displacements during the drum-shoe contact in the braking process.

Let us assume that the friction lining is pressed into the inner surface of the drum over a certain section. A normal force (pressing force) is applied by an eccentric to each unit of length of the lining. It is assumed that the contact area extends across the entire width of the friction lining and does not change over time during braking. In the contact zone, in addition to normal forces (pressures) $g(\theta,\ t)$, tangential stress $\tau(\theta, t)$ also acts

$$\begin{aligned} g(\theta,\ t) &= -\sigma_r\left(\theta,t,r_\delta\right); \\ \tau(\theta,\ t) &= \tau_{r\theta}\left(\theta,t,r_\delta\right), \end{aligned} \qquad (1)$$

associated with the contact pressure $g(\theta, t)$ according to Coulomb's law

$$g(\theta,\ t)\ =\ f\cdot g(\theta,\ t), \qquad (2)$$

where f is the coefficient of friction of the pair, "drum – lining". We assume that under the action of specific forces, $g(\theta,\ t)$ both $\tau(\theta,\ t)$ the drum and the friction lining are in a state of plane deformation.

Let us establish the relationships between the components of the displacements that occur in the contact area during vehicle braking [5-8]. We place the origin of the coordinates at the point of initial contact between the friction lining and the drum. Under the action of the pressing force, the lining will experience a displacement (settlement) of δ_2, and the drum will experience a displacement *of* δ_1. Point A, located on the surface of the lining, and point B, which comes into contact with it and is located on the inner surface of the drum, experience displacements of ν_1 and ν_2 in the radial direction, respectively, as a result of elastic deformation. Since the coordinates of points *A* and *B* become identical after they come into contact, this allows us to write the condition relating the displacements of the drum and the lining in the following form

$$\nu_1+\nu_2=\delta(\theta) \quad \left(\theta_0\le|\theta|\le\theta_1\right),$$

Here $\delta(\theta)$ *is* the settlement of the points of the surface of the drum and the lining, determined by the shape of the stamp base and the magnitude of the force F acting on it, θ_0, θ_1 – angles of the beginning and end of the pad coverage.

Let us now move on to finding the radial displacements v_1 and v_2.

Tangential forces $\tau(\theta,\ t)$ contribute to heat generation in the contact zone, with the total amount of heat per unit time being proportional to the friction power, and the amount of heat generated at a point in the contact area with coordinate θ is as follows

$$Q\left(\theta,\ t\right)=V\cdot\tau\left(\theta,\ t\right)=V\cdot f\cdot g\left(\theta,\ t\right), \qquad (3)$$

where $V=\omega r_b$ is the initial speed of movement of the drum surface points during braking; ω and r_b are the angular velocity and radius of the drum.

The amount of heat $Q(\theta, t)$ will be spent on the heat flow into the brake drum Q_* and a similar heat flow Q^* on increasing the temperature of the friction lining, i.e.

$$Q\left(\theta,\ t\right)=Q_*+Q^*. \qquad (4)$$

In [1-4, 9-10] it is recommended to use the average effective heat flow distribution coefficient in the calculation, which for one of the elements of the friction pair will be

$$\alpha_{TP2}=\left\{1+C_0\left\{\frac{F_{02}}{F_{01}}-\frac{F_{01}-F_{02}}{3F_{01}^2A}\left[\frac{1}{3}\ln\frac{\left(1/3-A\right)\left(1/3-F_{01}+A\right)}{\left(1/3+A\right)\left(1/3-F_{01}-A\right)}-A\ln\frac{3}{2}F_{01}\right]\right\}\right\}^{-1},$$

here

$$A=\sqrt{F_{01}^2+\frac{1}{9}};\qquad C_0=\frac{\Psi_{V2}b_2\lambda_1}{\Psi_{V1}b_1\lambda_2}.$$

For the other element of the pair

$\alpha_{TP1} = 1 - \alpha_{TP2}.$

Here $F_{01} = a_i t_T / b_i^2$ *is* the Fourier number; b_i is the thickness of the friction pair element; α_i is the thermal diffusivity coefficient; t_T – duration of a single braking in sec.; Ψ_{V1} – coefficient taking into account the effective volume participating in heat absorption; λ_i – thermal conductivity coefficient.

Based on the above, for the radial displacement v_1 – we will have [9, 10]

$v_1 = v_{1y} + v_{1T} + v_{1mp}.$

Here the term v_{1y} represents the elastic displacements of the points of the contact surface of the drum, v_{1T} - thermoelastic displacements caused by the temperature difference in the drum, term v_{1mp} represent movements caused by the removal of micro-protrusions from the inner surface of the drum.

To determine v_{1y} – it is necessary to solve the following auxiliary problem of elasticity theory on the inner surface of the drum at $\theta_0 \le |\theta| \le \theta_1$,

$$\begin{aligned} &\sigma_r = -g(\theta); \quad \tau_{r0} = 0 \quad at \quad r = r_b; \\ &\sigma_r = 0; \quad \tau_{r0} = 0 \quad at \quad r = R. \end{aligned} \tag{5}$$

Here *g(θ) is* an as yet unknown distribution function of contact stresses; *R* is outer radius of the drum.

To find v_{1r} it is necessary to solve the thermoelastic problem for a drum on the boundary of which there are no forces, and a heat source acts on the contact area.

3. Solution of the elastic problem

As is known [8], the stress state in an elastic body in the case of a plane problem in polar coordinates is determined by three stress components: σ_r, σ_θ, $\tau_{r\theta}$ which satisfy two equilibrium equations:

$$\begin{aligned} &\frac{\partial \sigma_r}{\partial r} + \frac{1}{r}\cdot\frac{\partial \tau_{r\theta}}{\partial \theta} + \frac{\sigma_r - \upsilon_\theta}{r} = 0; \\ &\frac{1}{r}\cdot\frac{\partial \sigma_\theta}{\partial \theta} + \frac{\partial \tau_{r\theta}}{\partial r} + \frac{2\tau_{r\theta}}{\tau} = 0; \\ &\sigma_r + \sigma_\theta = 4\,\mathrm{Re}\,\Phi(z), \quad z = x + iy; \\ &\sigma_\theta - \sigma_r + 2i\tau_{r\theta} = 2\left[\bar{z}\Phi'(z) + \Psi(z)\right]e^{2i\theta}; \\ &\sigma_r - i\tau_{r\theta} = \Phi(z) + \bar{\Phi}(z) - e^{2i\theta}\left[\bar{z}\Phi'(z) + \Psi(z)\right]; \\ &2G(v_r + iv_0) = e^{-i\theta}\left[\chi\varphi(z) - z\bar{\Phi}(z) - \psi(z)\right], \\ &G = \frac{E}{2(1+\mu)}. \end{aligned} \tag{6}$$

Here Φ(z) and Ψ (z) are arbitrary analytic functions of the complex variable $z = x + iy$ in the region occupied by the elastic medium; E is the Young's modulus of the material; μ is the Poisson's ratio of the material; G is the shear modulus.

Using formulas (6), we write the boundary conditions of the elasticity theory problem in the following form

$$\Phi(z)+\bar{\Phi}(z)-e^{2i\theta}\left[\bar{z}\Phi'(z)+\Psi(z)\right]=0 \quad at \quad r=R;$$
$$\Phi(z)+\bar{\Phi}(z)-e^{2i\theta}\left[\bar{z}\Phi'(z)+\Psi(z)\right]=\begin{cases}-g(\theta) \;\; at \;\; r=r_\delta, \;\; \theta_0\le|\theta|\le\theta_1 \\ 0 \;\; at \;\; r=r_\delta, \;\; |\theta|\le\theta_0 \;\; and \;\; \theta_1\le|\theta|\le\pi\end{cases} \tag{7}$$

Expanding the unknown function $g(\theta)$ on the inner contour $r = r_\delta$ in a Fourier series, we will have

$$g(\theta)=A_0+\sum_{n=1}^{\infty}A_n\cos n\theta+\sum_{n=1}^{\infty}B_n\sin n\theta. \tag{8}$$

Here

$$A_n=\frac{1}{\pi}\int_{\theta_0}^{\theta_1}g(\theta)\cos n\theta d\theta;\;\; B_n=\frac{1}{\pi}\int_{\theta_0}^{\theta_1}g(\theta)\sin n\theta d\theta.$$

The expansion (2.3.5) can be rewritten as follows

$$g(\theta)=\sum_{k=-\infty}^{\infty}A'_k e^{ik\theta};$$
$$A'_0=\frac{A_0}{2};\quad A'_k=\frac{A_k-iB_k}{2};\quad A'_{-k}=\frac{A_k+iB_k}{2}.$$

We seek the solution to the boundary value problem in the form [70]

$$\Phi(z)=A\ln z+\sum_{k=-\infty}^{\infty}a_k z^k;$$
$$\Psi(z)=\sum_{k=-\infty}^{\infty}a'_k z^k.$$

Omitting the cumbersome calculations, we present the solution of problem (5) of elasticity theory

$$\sigma_r=\frac{A_0}{\beta^2-1}\left(\frac{\beta^2}{\rho^2}-1\right)-\frac{\beta A_1}{\beta^4-1}\left(\frac{\rho}{\beta}-\frac{\beta^3}{\rho^3}\right)\cos\theta-$$
$$-\frac{\beta B_1}{\beta^4-1}\left(\frac{\rho}{\beta}-\frac{\beta^3}{\rho^3}\right)\sin\theta+\sum_{n=2}^{\infty}\frac{1}{2N}\left\{n\left[(n-1)-n\beta^2+\beta^{-2n}\right]\rho^{n-2}\right. \tag{9}$$
$$+n\left[(n+1)-n\beta^2-\beta^{2n}\right]\rho^{-(n+2)}+(n-2)\left[(n+1)-n\beta^{-2}-\beta^{-2n}\right]\rho^n+$$
$$\left.+(n+2)\left[(n-1)-n\beta^{-2}+\beta^{2n}\right]\rho^{-n}\right\}\times(A_n\cos n\theta+B_n\sin n\theta);$$

$$
\begin{aligned}
\sigma_\theta = {} & \frac{A_0}{\beta^2-1}\left(1-\frac{\beta^2}{\rho^2}\right)-\frac{\beta}{\beta^4-1}\left(\frac{3\rho}{\beta}+\frac{\beta^3}{\rho^3}\right)\left(A_1\cos\theta+B_1\sin\theta\right)+ \\
& +\sum_{n=2}^{\infty}\frac{1}{2N}\left\{n\left[-(n-1)+n\beta^2-\beta^{-2n}\right]\rho^{n-2}+\right. \\
& n\left[-(n+1)+n\beta^2+\beta^{2n}\right]\rho^{-(n+2)}+(n+2)\left[-(n+1)+n\beta^{-2}+\beta^{-2n}\right]\rho^{n}+ \\
& \left.+(n-2)\left[-(n-1)+n\beta^{-2}-\beta^{-2n}\right]\rho^{-n}\right\}\times \\
& \times\left(A_n\cos n\theta+B_n\sin n\theta\right);
\end{aligned}
\tag{10}
$$

$$
\begin{aligned}
\tau_{r\theta} = {} & \frac{\beta}{\beta^4-1}\left(\frac{\rho}{\beta}-\frac{\beta^3}{\rho^3}\right)\left(A_1\sin\theta+B_1\cos\theta\right)+ \\
& +\sum_{n=2}^{\infty}\frac{1}{2N}\left\{n\left[-(n-1)+n\beta^2-\beta^{-2n}\right]\rho^{n-2}+\right. \\
& n\left[(n+1)-n\beta^2-\beta^{2n}\right]\rho^{-(n+2)}+n\left[-(n+1)+n\beta^{-2}+\beta^{-2n}\right]\rho^{n}+ \\
& \left.+n\left[(n-1)-n\beta^{-2}+\beta^{2n}\right]\rho^{-n}\right\}\times\left(A_n\sin n\theta-B_n\cos n\theta\right);
\end{aligned}
\tag{11}
$$

$$
\begin{aligned}
v_r = {} & -\frac{A_0}{\left(\beta^2-1\right)E}\left[(1-\mu)+(1+\mu)\frac{\beta^2}{\rho^2}\right]r_\delta\rho+ \\
& +\frac{1}{E}\left[\frac{m+1}{m}\cdot\frac{\beta^2}{\rho^2}\cdot c_2+\frac{m-3}{m}\frac{\rho^2}{\beta^2}\cdot c_2\right]\beta r_\delta\cos\theta+ \\
& +\frac{1}{E}\left[\frac{m+1}{m}\cdot\frac{\beta^2}{\rho^2}\cdot c_2'+\frac{m-3}{m}\frac{\rho^2}{\beta^2}\cdot c_2'\right]\beta r_\delta\sin\theta+ \\
& +\sum_{n=2}^{\infty}\frac{1}{E}\left[-n\left(1+\frac{1}{m}\right)c_1''\left(\frac{\rho}{\beta}\right)^{n-1}+n\left(1+\frac{1}{m}\right)c_2''\left(\frac{\rho}{\beta}\right)^{-(n+1)}-\right. \\
& \left.-\left(n-2+\frac{n+2}{m}\right)c_3''\left(\frac{\rho}{\beta}\right)^{n+1}+\left(n+2+\frac{n-2}{m}\right)c_4''\left(\frac{\rho}{\beta}\right)^{-(n+1)}\right]\beta r_\delta\times \\
& \times\left(A_n\cos n\theta+B_n\sin n\theta\right);
\end{aligned}
\tag{12}
$$

$$
\begin{aligned}
v_\theta = {} & \frac{1}{E}\left[\frac{m+1}{m}\cdot\frac{\beta^2}{\rho^2}\cdot c_2+\frac{5m+1}{m}\frac{\rho^2}{\beta^2}\cdot c_2\right]\beta r_\delta\sin\theta- \\
& -\frac{1}{E}\left[\frac{m+1}{m}\cdot\frac{\beta^2}{\rho^2}\cdot c_2'+\frac{5m+1}{m}\frac{\rho^2}{\beta^2}\cdot c_2'\right]\beta r_\delta\cos\theta+ \\
& +\sum_{n=2}^{\infty}\frac{1}{E}\left[n\left(1+\frac{1}{m}\right)c_1''\left(\frac{\rho}{\beta}\right)^{n-1}+n\left(1+\frac{1}{m}\right)c_2''\left(\frac{\rho}{\beta}\right)^{-(n+1)}+\right. \\
& \left.+\left(n+4+\frac{n}{m}\right)c_3''\left(\frac{\rho}{\beta}\right)^{n+1}+\left(n-4+\frac{n}{m}\right)c_4''\left(\frac{\rho}{\beta}\right)^{-(n-1)}\right]\beta r_\delta\times \\
& \times\left(A_n\sin n\theta-B_n\cos n\theta\right).
\end{aligned}
\tag{13}
$$

Here

$$\beta = \frac{R}{r_\delta}, \quad \rho = \frac{r}{r_\delta}, \quad m = \frac{1}{\mu};$$

$$c_2 = -\frac{A_1\beta}{2\left(\beta^4-1\right)}, \quad c_2' = -\frac{B_1\beta}{2\left(\beta^4-1\right)};$$

$$c_1'' = \frac{-(n-1)+n\beta^2-\beta^{-2n}}{2(n-1)N}\beta^{-(n-2)};$$

$$c_2'' = \frac{-(n+1)+n\beta^2+\beta^{2n}}{2(n+1)N}\beta^{-(n+2)}; \tag{14}$$

$$c_3'' = \frac{-(n+1)+n\beta^{-2}+\beta^{-2n}}{2(n+1)N};$$

$$c_4'' = \frac{-(n-1)+n\beta^{-2}-\beta^{2n}}{2(n-1)N}\beta^{-n};$$

$$N = 2\left(n^2-1\right)-n^2\left(\beta^{-2}+\beta^2\right)+\left(\beta^{-2n}+\beta^{2n}\right).$$

CONCLUSION

To improve the accuracy of calculations for brake shoe-drum brake mechanisms, a method is proposed for determining the deformations of contacting surfaces and the contact stresses that arise in the shoe-drum pair during braking. Based on this approach, the problem of elasticity in the deformation of parts is addressed, and a method for the numerical calculation of contact stress is developed. Calculating contact stresses using modern computer technologies is straightforward.

In the future, to enhance the accuracy of calculations, it is proposed to consider thermoelastic deformations, displacements caused by the compression of micro-steps, and determine the total displacement of the contact points between the lining and the drum during braking. Determining the actual displacement will enable the development of a new approach to calculating drum-type brake mechanisms.

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THINK ABOUT OF PROPERTIES (P, Ρ, T) Of "KHACHMAZ" GEOTHERMAL WATER OF KHACHMAZ REGION

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ABSTRACT

The test research facility, in which the properties of warm water "Khachmaz" (p,ρ,T) in Khachmaz area, Examined was Azerbaijan which has air conditioning and a constant temperature of T=293.15 K.Data from various sources were compared with the re sults obtained for the watery arrangement of water, toluene, and NaCl (m=2.9666 1 mol·kg-1). The gotten comes about are displayed graphically within the figures. In article, the reliance of the thickness of Khachmaz warm water of the Khachmaz locale of Azerbaijan on the temperature of ρ/(kg · m-3) was measured within the high-precision temperature extend T = (278.15-468.15) in a tubular densimeter 5000M Anton-Paar DSA. Utilizing exploratory values at chosen temperatures, expository connections of warm water were set up. The gotten values are depicted by numerical conditions.

Presentation

Global initiatives are underway to decrease atmospheric carbon dioxide emissions. Significant measures are being implemented nationwide to address this issue. The Ministry of Industry and Energy was designated as the coordinator for executing the "State Program on the use of alternative and renewable energy sources in the Republic of Azerbaijan," as per the Presidential Decree dated October 21, 2004 1. This program outlines key strategies for harnessing the country's most efficient energy sources, including wind power, geothermal waters, solar energy, mountain rivers, water channels, and hydropower [1].

The republic's thermal water distribution zone, encompassing the entire Mesozoic layer, features both regional fractures and tectonically disturbed rock zones with complex, steeply falling crack systems. Water velocity is highest in primary drainage channels, where heat transfer to surrounding rocks is minimal, resulting in maximum water temperature. This phenomenon is also observed in highly fractured zones, as evidenced by geothermal source outputs in erosion depressions at the intersection of major suture zones [2].

The discharge zone's hydrogeological characteristics for thermal waters are directly influenced by rock fracturing and the nature of water-bearing formations. The prevalence and abundance of gryphons in natural outcrops indicate well-developed rifts.

This region's geothermal energy resources are particularly valuable. The area's diverse chemical composition, therapeutic properties, and favorable geographical location provide an excellent foundation for medical applications and extensive use across various sectors of the national economy [2,3].

1. The issue explanation and arrangement

The "Khachmaz" geothermal energy resource in this Azerbaijan's region underwent density measurements at atmospheric pressure using the DMA 5000M device, achieving 0.01% accuracy (superior to high-pressure measurements). This instrument enables precise measurements up to 363.15 K. The chemical composition of the Khachmaz thermal waters was analyzed using an IRIS Intrepid II Optical Emission Geomotograph with inductively coupled plasma atomic emission spectrometer 3,4. Results indicate that sodium (Na) is the predominant chemical element, comprising approximately 72.41 to 90.12% of all chemical substances in the "Khachmaz" thermal water of the Khachmaz region in Azerbaijan [3].

3. Dialog of the investigate work and it`s comes about:

The figures below demonstrate that the discrepancy between the newly acquired density measurements and previously published data falls within the estimated margin of error for this apparatus. Water that underwent double distillation was procured from various laboratory sources. Sodium chloride, methanol, and additional chemicals were acquired from the German company Merck. The outcomes consistently exhibited minimal variations and remained in close proximity to one another. This consistency serves as evidence of the high precision achieved by the newly developed experimental equipment. [3,4].

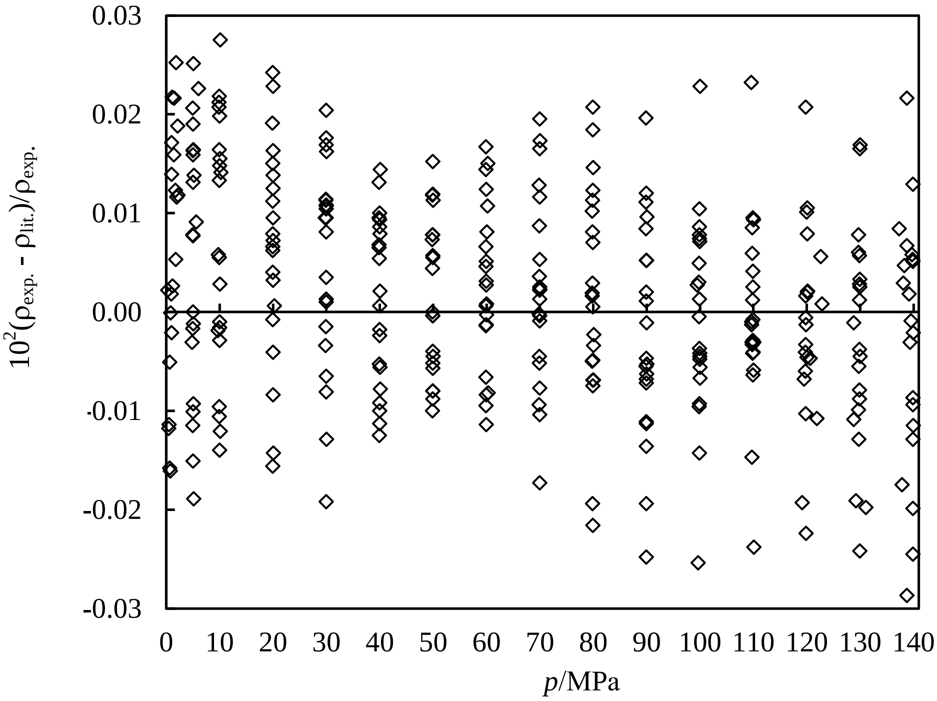

Fig1. Relationship between observed water density and pressure at temperatures ranging from 278.15 K to 468.15 K, and its deviation from IAPWS 95 3 reference data.

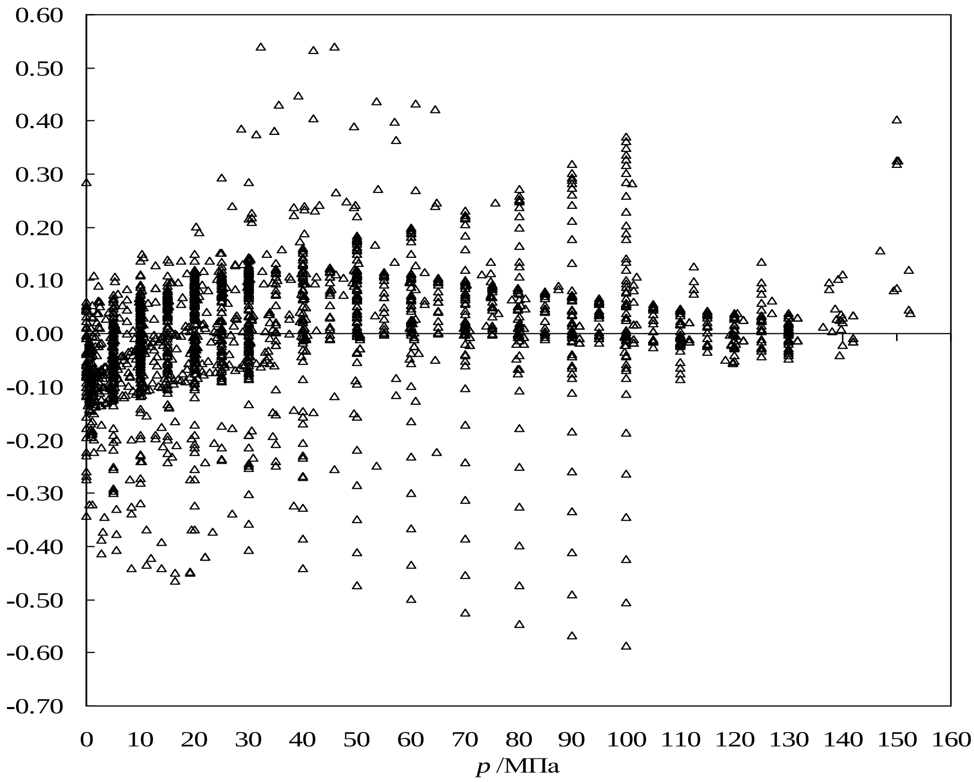

Fig. 2. Dependence of the measured density of toluene at temperature T = (278.15-468.15) K and its difference with data from various literature (data up to 2000) on pressure.

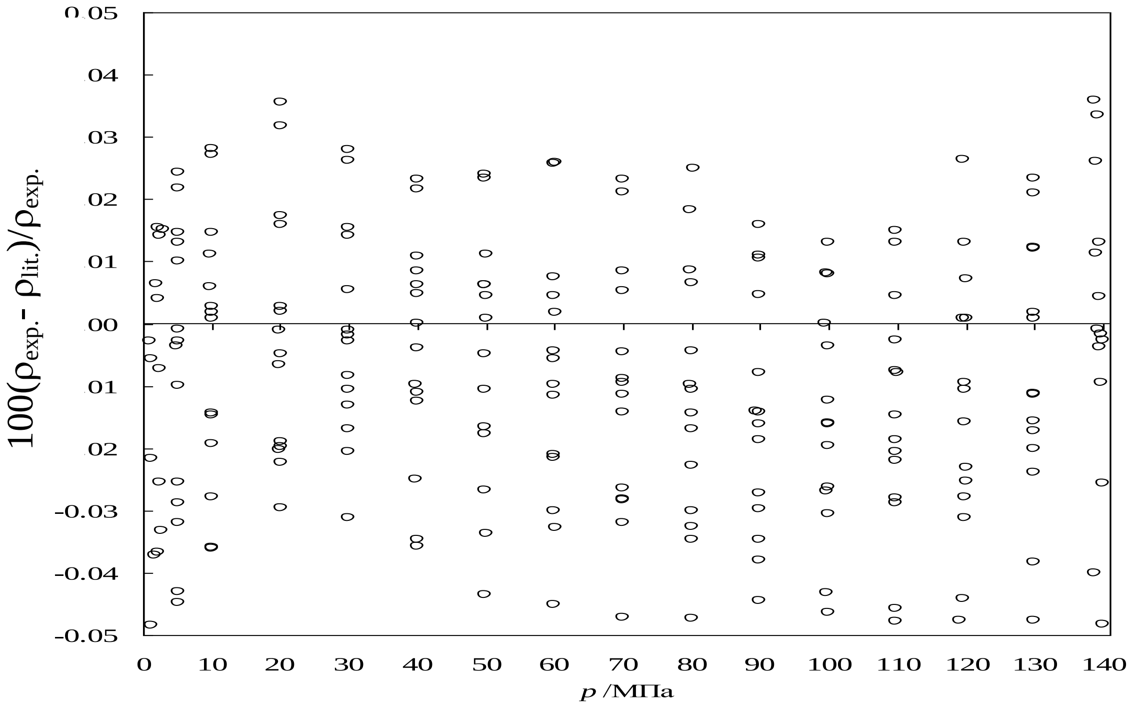

Fig. 3. Dependence of the measured density of an aqueous solution of NaCl (M=2.96661 mol·kg1) at temperature T= (278.15-468.15) K on pressure and differences with data from various literature (data up to 2000).

When calculating each agreement state's dependences (p, ϱ, and T), it was looked for to form as moo weight values as conceivable in arrange to get profoundly precise thickness values utilizing graphical extrapolation at barometrical weight, which were compared with the values of thickness measured on the DMA 4500 gadget. The information gotten by these diverse strategies are in great understanding inside ±0.02%. Each isotherm test was performed with weight interims of around 5 MPa. Investigates for all examined objects were conducted at temperatures beginning from T=(278.15÷373.15) K and weights up to p=40 MPa. The gotten test parameters (p, ϱ, T) are given in table1. [3, 4].

Table 1. Test values of the thickness of warm water "Khachmaz" in Khachmaz area, Azerbaijan at different weights and temperatures

$\frac{p}{MPa}$	$\frac{\rho}{kg \cdot m^3}$	$\frac{T}{K}$	$\frac{p}{MPa}$	$\frac{\rho}{kg \cdot m^3}$	$\frac{T}{K}$
0.624	1004.04	278.15	1.160	989.32	328.02
5.004	1006.21	278.15	5.024	990.99	328.04
10.023	1008.65	278.16	10.079	993.00	328.17
15.012	1011.05	278.15	15.576	995.38	328.18
20.035	1013.43	278.14	19.985	997.22	328.19
25.036	1015.76	278.15	25.527	999.61	328.17
30.054	1018.07	278.15	30.023	1001.50	328.14
35.124	1020.37	278.14	35.513	1003.58	328.12
40.021	1022.56	278.15	39.978	1005.37	328.06
0.539	1002.59	288.14	0.846	981.22	343.15
5.006	1004.66	288.16	5.097	983.06	343.16
9.855	1006.87	288.17	9.967	985.17	343.14
15.151	1009.25	288.17	15.525	987.55	343.15
20.064	1011.43	288.17	20.000	989.45	343.15
25.121	1013.64	288.16	25.586	991.82	343.14
30.103	1015.79	288.16	30.045	993.68	343.16
35.111	1017.92	288.16	35.514	995.98	343.15
40.145	1020.04	288.15	40.050	997.87	343.15
1.025	1000.15	298.27	0.846	974.29	354.24
5.079	1002.02	298.22	5.097	976.23	354.25
9.818	1004.22	298.22	9.967	978.33	354.27
15.593	1006.61	298.17	15.525	980.63	354.27
20.018	1008.46	298.13	20.000	982.58	354.27
25.104	1010.69	298.13	25.586	984.92	354.27
30.155	1012.85	298.12	30.045	986.83	354.28
35.089	1014.82	298.13	35.514	989.17	354.27
40.040	1016.88	298.13	40.050	991.07	354.27
0.898	995.52	313.08	1.626	962.02	372.90
4.995	997.25	313.10	5.059	963.59	372.90
9.972	999.20	313.15	10.042	965.73	372.96
15.563	1001.65	313.17	15.525	968.08	372.97
20.008	1003.42	313.20	20.014	970.00	372.99
25.534	1005.80	313.18	25.596	972.31	373.00
30.057	1007.65	313.19	30.001	974.44	372.90
35.586	1009.82	313.17	35.576	976.79	372.91
39.970	1011.52	313.15	40.013	978.53	372.92

Isotherms are plotted in p-ϱ arranges within the weight extend of 0,1-40 MPa (Figure 4).

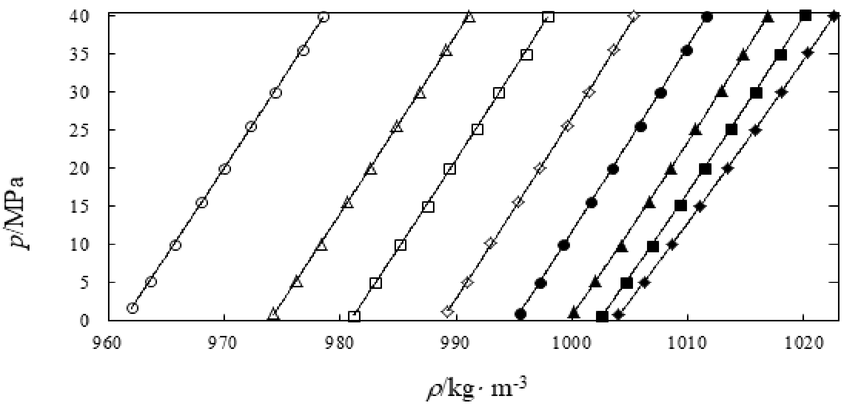

Fig. 4. Reliance of weight (p) on thickness (ϱ) of warm water "Khachmaz" in the same locale, Azerbaijan, calculated agreeing to equations1-2: ♦, 278,15 K; ■, 288,16 K; ▲, 298,17 K; •, 313,18 K; ◊, 328,18 K;□, 343,15 K; Δ, 354,27 K; ○, 372,96 K.

The DMA of the thermal water under study was measured with a 5000M density meter. The mentioned device makes it possible to carry out measurements at temperatures up to T = 363.15 K. The results obtained are recorded using the following formulas [5]:

$$p = A\rho^2 + B\rho^8 + C\rho^{12} \quad (1)$$

The use of the Akhundov-Imanov formula of Azerbaijani scientists reduces the measurement error to $\Delta\varrho/\varrho=\pm(0.002\div0.004)\%$. The coefficients A(T), B(T) and C(T) depend polynomially on temperature:

$$A(T) = \sum_{i=1}^{3} a_i T^i,\ B(T) = \sum_{i=0}^{2} b_i T^i,\ C(T) = \sum_{i=0}^{2} c_i T^i \quad (2)$$

The values of the coefficients a_{ij}, b_{ij} və c_{ij} in equation (2) are given in table 2.

Table 2.

a_1= -3.9409326	b_0= 8245.9457814	c_0= -6498.34719856
a_2= 0.028469201	b_1= -55.5978423	c_1= 44.49657812795
a_3= -0. 3472496358·10^{-4}	b_2= 0.043887215947	c_2= -0.06724687932

Condition (1) enables it possible to type in down the experimental values of the dependency on warm water "Khachmaz" (p, ϱ, T) with a normal blunder of 0.006%, taking into account the n umbers of the integrals A(T), B(T) and C(T).

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3. File format should be compatible with **Microsoft Word** and must be sent to the electronic mail (**journal@beu.edu.az**) of the Journal. The submitted article should follow the following format:

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4. The title of the article should be in each of the three languages of the abstract and should be centred on the page and in bold capitals before each summary.
5. **The abstract** should be written in **9 point** type size, between **100** and **150** words. The abstract should be written in the language of the text and in two more languages given above. The abstracts of the article written in each of the three languages should correspond to one another. The keywords should be written in two more languages besides the language of the article and should be at least three words.
6. .**UDC** and **PACS** index should be used in the article.
7. The article must consist of the followings:

- Introduction
- Research method and research
- Discussion of research method and its results
- In case the reference is in Russian it must be given in the Latin alphabet with the original language shown in brackets.

8. **Figures, pictures, graphics and tables** must be of publishing quality and inside the text. Figures, pictures and graphics should be captioned underneath, tables should be captioned above.
9. **References** should be given in square brackets in the text and listed according to the order inside the text at the end of the article. In order to cite the same reference twice or more, the appropriate pages should be given while keeping the numerical order. For example: [7, p.15].

Information about each of the given references should be full, clear and accurate. The bibliographic description of the reference should be cited according to its type (monograph, textbook, scientific research paper and etc.) While citing to scientific research articles, materials of symposiums, conferences and other popular scientific events, the name of the article, lecture or paper should be given.

Samples:

a) ***Article:*** Demukhamedova S.D., Aliyeva İ.N., Godjayev N.M.. *Spatial and electronic structure af monomerrik and dimeric conapeetes of carnosine üith zinc*, Journal of structural Chemistry, Vol.51, No.5, p.824-832, 2010

b) ***Book:*** Christie ohn Geankoplis. *Transport Processes and Separation Process Principles*. Fourth Edition, Prentice Hall, p.386-398, 2002

c) ***Conference paper:*** Sadychov F.S., Aydın C., Ahmedov A.İ.. Appligation of Information – Commu-nication Technologies in Science and education. II International Conference."*Higher Twist Effects In Photon- Proton Collisions", Bakı, 01-03 Noyabr, 2007, ss 384-391*
References should be in 9-point type size.

10. The margins sizes of the page: - Top 2.8 cm. bottom 2.8 cm. left 2.5 cm, right 2.5 cm. The article main text should be written in Palatino Linotype 11 point type size single-spaced. Paragraph spacing should be 6 point.
11. The maximum number of pages for an article should not exceed 15 pages
12. The decision to publish a given article is made through the following procedures:

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- The article is sent back to the author to make amendments upon the recommendations of referees.
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YAZI VƏ NƏŞR QAYDALARI

1. "Journal of Baku Engineering University- Mexanika və sənaye mühəndisliyi" - əvvəllər nəşr olunmamış orijinal əsərləri və müəllifin tədqiqat sahəsi üzrə yazılmış icmal məqalələri qəbul edir.
2. Məqalələr İngilis dilində qəbul edilir.
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- Elektron ünvanı,
- Xülasə və açar sözlər.

4. **Məqalədə başlıq hər xülasədən əvvəl** ortada, qara və böyük hərflə xülasələrin yazıldığı hər üç dildə olmalıdır.
5. **Xülasə** 100-150 söz aralığında olmaqla, 9 punto yazı tipi böyüklüyündə, məqalənin yazıldığı dildə və bundan əlavə yuxarıda göstərilən iki dildə olmalıdır. Məqalənin hər üç dildə yazılmış xülasəsi bir-birinin eyni olmalıdır. Açar sözlər uyğun xülasələrin sonunda onun yazıldığı dildə verilməklə ən azı üç sözdən ibarət olmalıdır.
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- Tədqiqat işinin müzakirəsi və onun nəticələri,
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8. **Şəkil**, **rəsm**, **grafik** və **cədvəllər** çapda düzgün, aydın çıxacaq vəziyyətdə və mətn içərisində olmalıdır. Şəkil, rəsm və qrafiklərin yazıları onların altında yazılmalıdır. Cədvəllərdə başlıq cədvəlin üstündə yazılmalıdır.
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Ədəbiyyat siyahısında verilən hər bir istinad haqqında məlumat tam və dəqiq olmalıdır. İstinad olunan mənbənin biblioqrafik təsviri onun növündən (monoqrafiya, dərslik, elmi məqalə və s.) asılı olaraq verilməlidir. Elmi məqalələrə, simpozium, konfrans, və digər nüfuzlu elmi tədbirlərin materiallarına və ya tezislərinə istinad edərkən məqalənin, məruzənin və ya tezisin adı göstərilməlidir.

Nümunələr:

a) ***Məqalə:*** Demukhamedova S.D., Aliyeva İ.N., Godjayev N.M.. *Spatial and electronic structure af monomeric and dimeric complexes of carnosine with zinc,* Journal of structural Chemistry, Vol.51, No.5, p.824-832, 2010

b) ***Kitab:*** Christie ohn Geankoplis. *Transport Processes and Separation Process Principles*. Fourth Edition, Prentice Hall, 2002

c) ***Konfrans:*** Sadychov F.S., Aydın C., Ahmedov A.İ.. Appligation of Information-Communication Technologies in Science and education. II International Conference. "*Higher Twist Effects In Photon- Proton Collisions", Bakı, 01-03 Noyabr, 2007, ss 384-391*

Mənbələr 9 punto yazı tipi böyüklüyündə olmalıdır.

10. **Səhifə ölçüləri:** üstdən 2.8 sm, altdan 2.8 sm, soldan 2.5 sm və sağdan 2.5 sm olmalıdır. Mətn 11 punto yazı tipi böyüklüyündə, **Palatino Linotype** yazı tipi ilə və tək simvol aralığında yazılmalıdır. Paraqraflar arasında 6 punto yazı tipi aralığında məsafə olmalıdır.
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YAZIM KURALLARI

1. "Journal of Baku Engineering University- Makine ve endüstri mühendisliği" önceler yayımlanmamış orijinal çalışmaları ve yazarın kendi araştırma alanın-da yazılmış derleme makaleleri kabul etmektedir.
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Örnekler:

a) ***Makale:*** Demukhamedova S.D., Aliyeva İ.N., Godjayev N.M.. *Spatial and Electronic Structure of Monomerik and Dimeric Conapeetes of Carnosine Üith Zinc*, Journal of Structural Chemistry, Vol.51, No.5, p.824-832, 2010

b) ***Kitap:*** Christie ohn Geankoplis. *Transport Processes and Separation Process Principles*. Fourth Edition, Prentice Hall, p.386-398, 2002

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Kaynakların büyüklüğü 9 punto olmalıdır.

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- Аннотация и ключевые слова

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7. Представленные статьи должны содержать:

- Введение
- Метод исследования
- Обсуждение результатов исследования и выводов.
- Если ссылаются на работу на русском языке, тогда оригинальный язык указывается в скобках, а ссылка дается только на латинском алфавите.

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   Например:

a) ***Статья:*** Demukhamedova S.D., Aliyeva I.N., Godjayev N.M. *Spatial and electronic structure of monomeric and dimeric complexes of carnosine with zinc,* Journal of Structural Chemistry, Vol.51, No.5, p.824-832, 2010

b) ***Книга:*** Christie on Geankoplis. *Transport Processes and Separation Process Principles.* Fourth Edition, Prentice Hall, 2002

c) ***Конференция:*** Sadychov F.S, Fydin C,Ahmedov A.I. Appligation of Information-Communication Nechnologies in Science and education. II International Conference. *"Higher Twist Effects In Photon-Proton Collision",* Bakı,01-03 Noyabr, 2007, ss.384-391

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