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Drobenko, B.D., Budz, S.F., and Astashkin, V.I.

Ya.S. Pidstryhach Institute of Applied Problems of Mechanics and Mathematics of NASU, Lviv

EVALUATION OF SUITABILITY OF POWER ENGINEERING EQUIPMENT ELEMENTS FOR THEIR USE OVER THE FLEET LIFE

An approach to the determination of stress-strain state of boiler's operating elements has been proposed. The possibility of their further exploitation has been discussed. The state of the boiler drums at Unit 6 of the Dobrotvir power plant and at Unit 5 of the Burshtyn power plant of DTEK ZakhidEnergo, PJSC, with exhausted fleet life has been studied. Recommendations on the improvement of techniques for their repair have been developed; a conclusion on the possible extension of their operational lifetime has been made.

Key words: mathematical and numerical simulation, strength and operational lifetime.

The reliability of power equipment and extension of its operational lifetime is one of the most relevant problems of the domestic energy sector. Because of limited funding the assessment of the residual life and the suitability of such equipment for being used over the guaranteed operational life is of paramount importance, insofar as a significant portion of such equipment has exhausted or nearly exhausted its fleet life [1], with a new boiler having a power capacity of 200 MW costing about USD 250 million [2]. If there are numerous operational damage it is necessary to develop efficient technologies for repairing the power equipment elements in order to lengthen their life for forecast period.

The operational life of elements of power engineering equipment is assessed by checking their status by level of accumulated metal damage (see [3]), which is the basis for quantitative evaluation of additional resources, opportunities, and conditions for further exploitation. The level of accumulated damage greatly depends on the maximum stress and the amplitude of their changes under different regimes of industrial equipment exploitation. In this regard, there is a practical need to build and to develop methods for the refined calculation of the stress state of the elements of power engineering equipment under the operational conditions, which adequately take into account the actual geometric shape of structural elements changeable in the course of operation, the thermal sensitivity of material properties within the whole range of temperature change, and the character of elastic-plastic deformation. Such techniques can be based on the general non-linear thermomechanic relationships using state-of-the-art numerical methods [4].

The research of deformation of structural elements of NPP existing boilers and the possibilities of their further exploitation over the fleet life were conducted at the Pidstryhach Institute for Applied Problems of Mechanics and Mathematics (IPPMM) of NASU. Below, the results of research are given. The study is based on the ratios of 3D non-isothermal thermoelastic plasticity [5] using the finite element method (to approximate

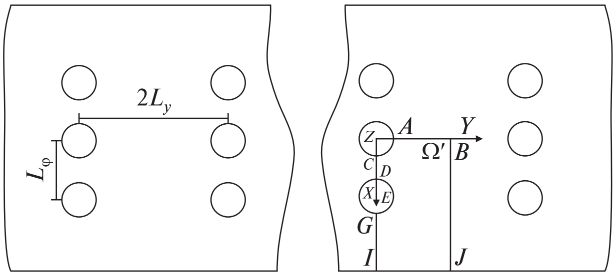

Fig. 1. Computational domain of TP-10 drum: hollow cylinder (L = 14 m, R_1 = 0.8 m; R_2 = 0.889 m) with series, each having three holes (m = 3; r = 0.0525 m) with distances $2L_y$ = = 0.71 m, $L_\varphi = 4\pi R_2/45$ m

the temperature and mechanical fields by spatial variables) and many one-parametric difference algorithms [6] (to approximate by time). The computer simulation was conducted using the software as described in the monograph [4]. The results have been implemented at Dobrotvir and Burshtyn NPP of DTEK *Zakhidenergo*, PJSC, and *ZAES*.

In Ukraine, the majority of TPP units is equipped with drum-type boilers. Therefore, boiler drums, one of the most loaded boiler's elements, are of particular importance for ensuring the reliability of the units under consideration. Given the fact that a third of drums has exhausted their fleet life, with the rest thereof approaching to this limit [1], to prolong their life over fleet life is a problem of profound importance.

The boiler drum is a massive cylindrical body with a complex system of inlets and outlets and with spherical bottoms at the edges. It is important to adequately simulate the deformation of spatial body with such a complex geometry under the relevant thermal and power loads which allow for different modes of drum operation (scheduled commissioning, shutdown, emergency shutdown, hydraulic test, slow fluctuations of working environment temperature around the stationary operating temperature, i.e. thermal cycling), insofar as the maximum stress is one of the most important factors taken into account when assessing the residual life of the drum and the accumulated metal damage and when deciding on the suitability of the drum for its further use.

In connection with the above, a 3D hollow cylinder (length L, inner radius R_1 and outer radius R_2), the edges of which are closed with spherical bottoms, is taken as a drum design model. In general, in the cylinder body there are periodic series with m inlets and outlets (of radius r), the distances between them are $2L_y$ and L_φ in the axial and in the circumferential directions, respectively (see Fig. 1). It should be noted that in general, the geometric dimensions and the number of holes depend on the type of drum.

The pressure at the inner surface of the cylinder and at the surfaces of holes p is set depending on the working environment (water-steam mixture) at a temperature of T_p. The temperature and pressure may vary from time to time depending on the particular operational regime.

Firstly, let us build a design model for estimating the stress state of a TP-10 drum in the vicinity of series of three holes in the circumferential direction. Let us choose a random series of three holes and assign the hollow cylinder to the right-handed Cartesian coordinate system so that the Z axis forms an axis of rotation of the center hole and the Y axis is directed along the cylinder axis (see Fig. 1).

Having assumed that the hole series are equivalent and the operational stresses in the vicinity of the selected three holes affect the stress state in the vicinity of a nearby series of holes in the same manner as the stresses in the vicinity of neighboring series affect the stresses in the vicinity of the selected three holes, let us confine ourselves to the consideration of a fragment of V' region schematically showed in Fig. 2. Similarly, the computational domain for more holes is built using the symmetry conditions.

Then, the determination of the stress state of the drum under operational conditions is reduced to determining the spatial and time distribution of temperature and displacements, strains, and stresses from the unsteady heat transfer equation and the complete system of equations of non-isothermal thermoelastic plasticity [4] in V' area under appropriate initial and boundary conditions.

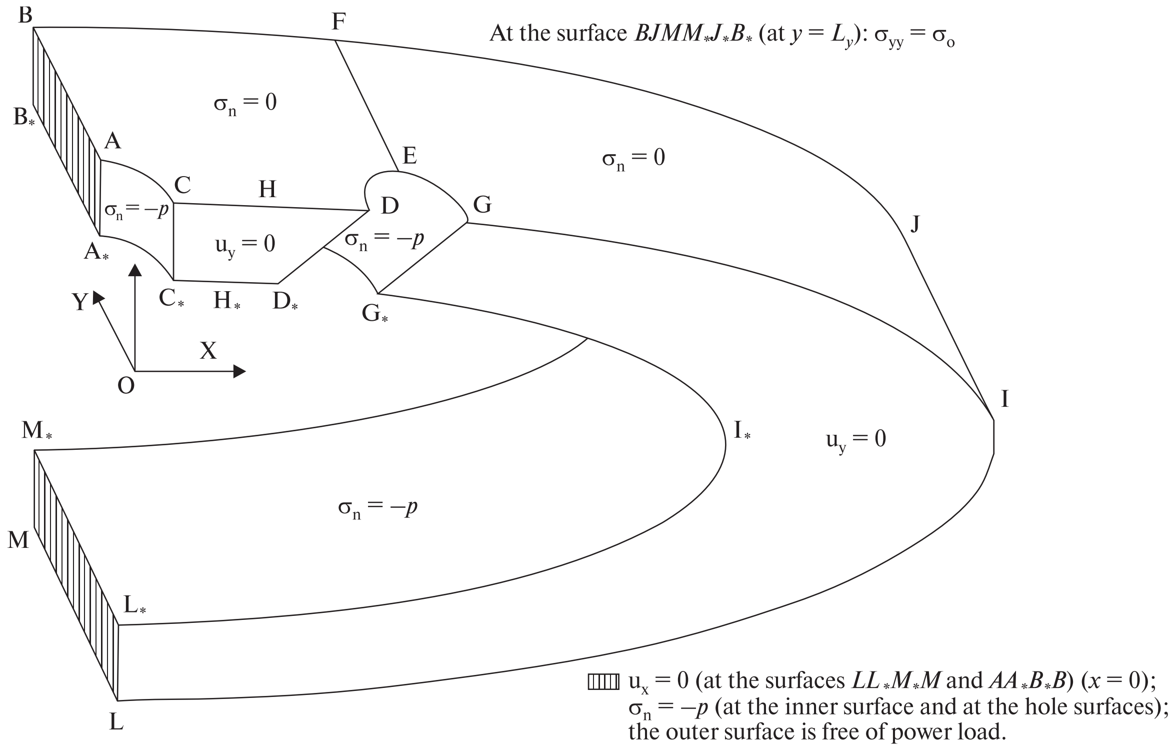

Fig. 2. The computational domain V' (subject to the symmetry conditions) and the boundary conditions

For the simulation of drum operation in a stationary mode on the following surfaces: $GILL_*I_*G_*$ and SDD_*C_* (where $y = 0$) and LL_*M_*M and AA_*B_*B (where $x = 0$) the symmetry conditions with respect to the movement are established as $u_y = 0$ and $u_x = 0$, respectively (see Fig. 2).

On the inner surface and on the surfaces of the holes there is pressure p which ranges from the nominal operation pressure to zero (depending on the proposed operational mode of the boiler); on the surface $BJMM_*J_*B_*$ there are stretching stresses $\sigma_{yy} = \sigma_0 = pR_1^2/(R_2^2 - R_1^2)$ caused by pressure effect on the bottoms. The outer surface is free from stresses.

The heat exchange in the considered area with working environment is simulated using the boundary conditions of convective heat transfer [4]. The temperature of water-steam mixture slowly fluctuates in the vicinity of operating temperature T_0 with amplitude ΔT and frequency ω as a result of mixing the boiler water and the feed water in the drum (thermal cycling phenomenon). On the outer surface $ACHDEGILMJFB$ and the following parts of surface S' of area V': $LL_*M_*M, AA_*B_*B, GILL_*I_*G_*$, SDD_*C_*, and $BFJMM_*J_*F_*B_*$ due to the symmetry conditions the thermal insulation conditions (heat flow is equal to zero) are established.

To simulate the scheduled commissioning and shutdown, the temperature T_p and the pressure p of water-steam mixture over time are set in accordance with the schedule of accelerated heating of boilers (see Fig. 3). While simulating the shutdown of drum boiler in emergency mode, it is assumed that the cylinder cools down from the working temperature T_0 due to convective heat exchange (from the inner surface and holes) with environment whose temperature T_e is lower by 100 °C (according to [3]) ($T_e = T_0 - 100$). The internal pressure drops down from the nominal value to zero at a rate that allows us to consider the problem of thermomechanics in quasi-static approximation [4].

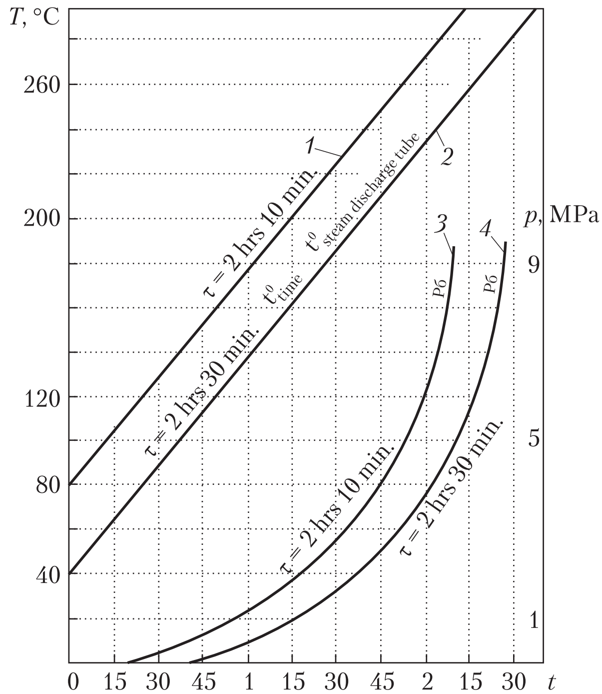

Fig. 3. Acceleration of TP-10 boiler heating

In the case of hydraulic test mode, the initial and boundary conditions coincide with the conditions of stationary operation (without thermal cycling). In this case, the internal pressure is higher by 25% as compared with the nominal operating one (consequently, the axial strain set on the surface $BJMM_*J_*B_*$ increases).

Thus, determining the stress-strain state in the vicinity of holes of the boiler high-pressure drum in operational conditions is reduced to the solution of the full system of nonlinear thermo-mechanical equations [4] using the finite element method. The fragments of the finite-element division of the holes vicinity are showed in Figs. 4 and 5 for the cases of three and four holes in the circumferential direction, respectively.

The study of convergence of numerical methods for solving the nonlinear thermo-mechanical problem using the finite element method proposed in [4] have showed that for obtaining exact solutions it is enough to have a four-layer-thick finite element division with eight elements at half the hole. Hence, the solutions obtained for twice denser divided area differ by less than 1%. The study of h-convergence (comparison of solutions obtained for the finite element divisions of various density with the use of the same finite elements) and p-convergence (comparison of solutions obtained using the elements of different orders of approximation on divisions of the same density) testify to a rather high accuracy of solutions obtained using biquadratic finite elements which are found to be optimal by mutually contradictory criteria of cost effectiveness and precision.

The stress-strain state of TP-10 drum boiler at the unit 6 of Dobrotvir TPP of *DTEK Zakhidenergo*, PJSC, which as of 01.07.2013 had worked for 295,149 hours (2,504 scheduled commissioning and shutdowns, 11 emergency stops) were studied. The calculations were performed for 22K steel at a pressure ranging from the nominal working one (11.5 MPa, in the stationary mode and 14.3 MPa, in the hydraulic test mode) to zero (depending on the proposed mode of boiler operation), for the following numerical parameters: $R_1 = 0.8$ m; $R_2 = 0.889$ m; $r = 0.0525$ m; $L_y = 0.335+/-0.425$ m; $L_\varphi = 0.124$ m; and $T_0 = 311$ °C.

Heat transfer coefficient $\beta = 2204$ W/m²/K. Under the stationary operating condition the temperature of working environment fluctuated with an amplitude $\Delta T = 5$ °C and a frequency $\omega = 4.672$ 1/hour around the working temperature T_0. In the case of emergency stop the temperature is $T_e = 211$ °C.

The computational experiment has showed that the maximum stress in the drum arises on the inner surface in the vicinity of the holes at the points A_* and E_*. Figure 6 shows the distribution of the stress intensity on the inner (*solid* lines) and the outer (*dashed* lines) drum surfaces along the lines of holes A_*C_* and AC (Fig. 6, *a*) and $D_*E_*G_*$ and DEG (Fig. 6, *b*) depending on the angular coordinate α′ introduced for convenience (the corresponding coordinate α′ = 0 at points A_*, A, E_*, E). It can be seen that the maximum stresses on the outer drum surface in similar locations are significantly lower.

The distribution σ_{xx}, σ_{yy}, and σ_{zz} components of the stress tensor (in the Cartesian coordinates)

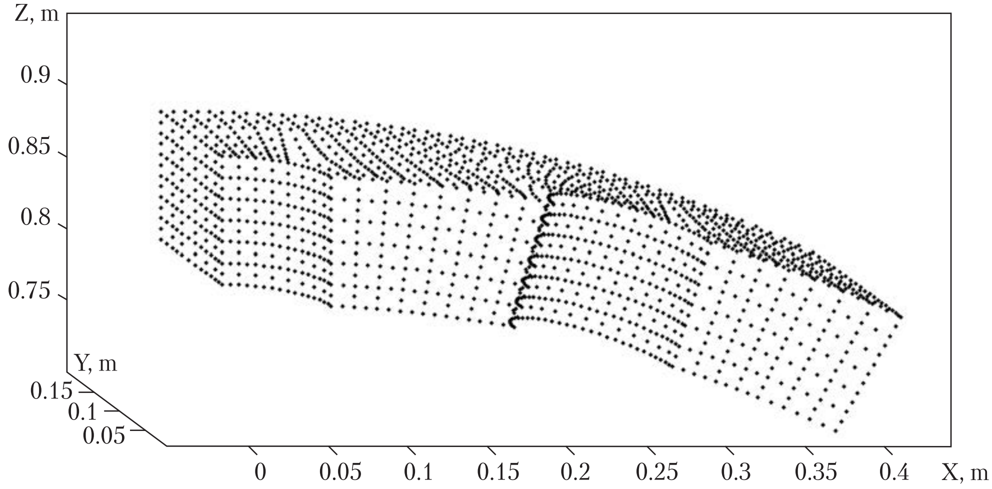

Fig. 4. Units of finite-element division into biquadratic curvilinear elements in the vicinity of the holes [4]

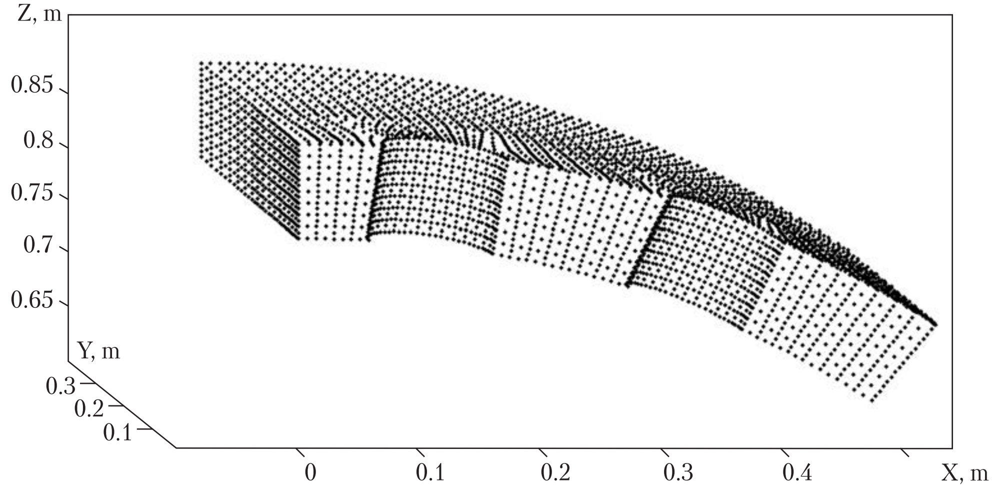

Fig. 5. Units of finite-element division into biquadratic curvilinear elements in the vicinity of the holes

on the inner (along the A_*C_* line) and the outer (along the *AC* line) drum surfaces along the lines of the median hole is showed in Figs. 7, *a* and 7, *b*, respectively.

Figure 8 shows the distribution of stress intensity along the lines A_*B_* and *AB* (Fig. 8, *a*) and E_*F_* and *EF* (Fig. 8, *b*). One can see that in the vicinity of holes there is a significant stress concentration. The size of neighborhood of the stress state disturbance is approximately equal to the hole radius *r*. Outside this neighborhood, in the drum, there is momentless stressful condition that occurs under the internal pressure within a long hollow cylinder of the same geometric dimensions without holes. The maximum stress in the vicinity of points A_* and E_* on the holes of the drum inner surface exceeds the limit of plasticity. Therefore, the description of the drum stress state in these areas based on the theory of elasticity is not quite adequate (hereinafter, the results of cal-

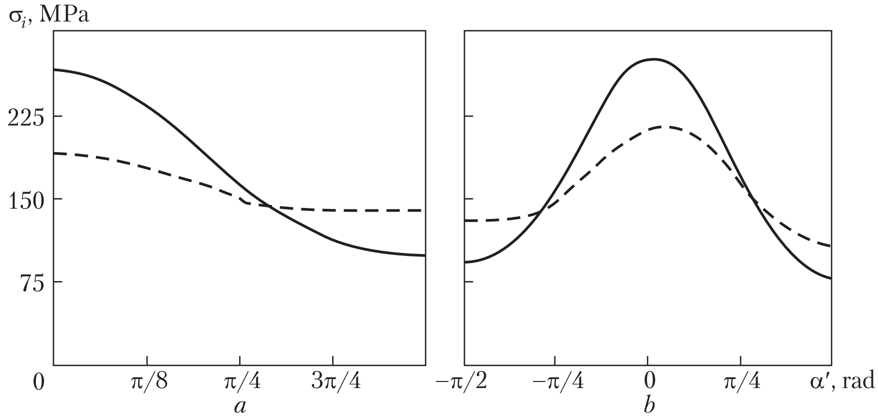

Fig. 6. The intensity of stresses σ_i on the outer and the inner drum surfaces along the lines of the median (a) and the end (b) holes

culations based on the theory of non-isothermal thermo-elastic plasticity are given).

It should be noted also that the end holes is slightly more stressed than the median one. The similar results were obtained for the vicinity of the four-hole series in the circumferential direction. In this case, the maximum stress is slightly lower (by 5 MPa).

In general, it can be stated that under the action of internal pressure the maximum stress slightly decreases as the number of holes in the circumferential direction increases; as a rule, they act on the outer holes from the drum inner surface. The most stressed places in the drum is small neighborhood of the holes (having a size of several centimeters). In these areas, damage and defects are expected to occur above all others. The maximum stress intensity was estimated using the spatial theory of elasticity. It was found to be equal to 284.6 MPa (in the neighborhood of point E_* on the inner surface, see Fig. 2) for the end holes in the presence of three-hole series and 279.6 MPa for the end holes in the presence of four-hole series.

In the hydraulic test mode (when the pressure increases by 25%) the maximum «conditionally elastic» strains for the cases under review are equal to 355.75 MPa (for the neighborhood of three holes in the circumferential direction) and to 349.5 MPa (for the neighborhood of four holes in the circumferential direction) which significantly exceeds the limit of plasticity of 22K steel (225 MPa) [3, 7, and 8]. Hence, one can conclude that describing the drum deformation processes using the elastic body model is inadequate. Some results of calculations based on the theory of thermo-elastic plasticity for the neighborhoods of three holes are showed in Fig. 9.

For the shells made of 22K steel with a plastic limit of 225 MPa in the hydraulic test mode the plastic strain range in the vicinity of the most stressed end holes covers the entire width of the drum (see Fig. 9, a), and extends over 25 mm in the longitudinal direction (with respect to the hole) on the inner surface.

After complete unloading in the places where the plastic strains occur the compressive residual stresses appear (the maximum stress is about 110 MPa). As a result of plastic strain the material strengthens in these areas and later (under the nominal load of p = 11.5 MPa in the stationary operation mode), it is strained only according to the elastic strain law.

When the temperature of working environment rises monotonously (for example, during the boiler heating) the temperature of the drum inner surface grows faster than the temperature of its outer surface. On the drum inner surface, there occur the compressive thermal stresses which, to some extent, compensate the tensile stresses cau-

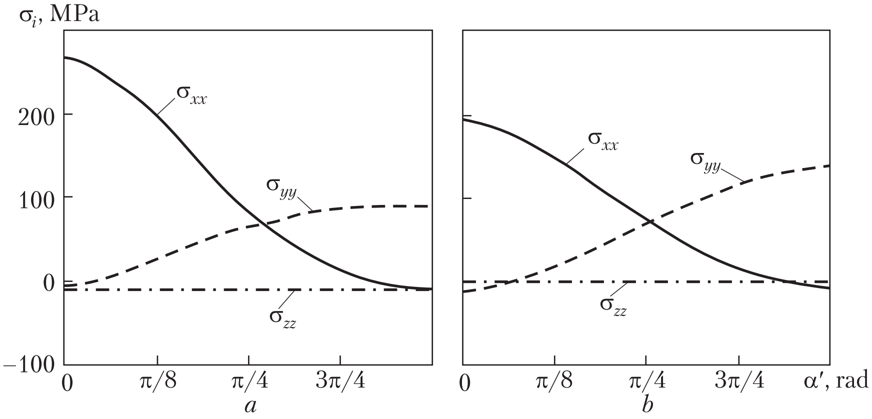

Fig. 7. Components of the stress tensor on the median hole of the drum on the inner and the outer surfaces, respectively

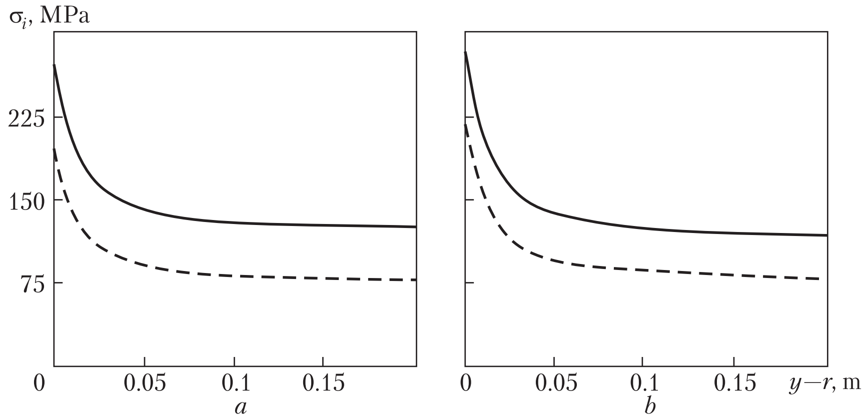

Fig. 8. The intensity of the stresses on the inner (solid lines) and outer (dashed lines) surfaces of the drum depending on the distance from the hole edges in the longitudinal direction

sed by pressure effects, so that the total stress is always lower than the stress resulting from the effects of pressure.

At the initial stage of heating of the boiler when the inner pressure of working environment is at the level of atmospheric pressure (see Fig. 3), there are only thermal compressive stresses. These stresses reach their absolute maximum value (about 80 MPa in the neighborhood of A_* point for the median hole on the drum inner surface) during the planned commissioning, when the temperature grows with a heating rate not exceeding 3 °C/min, at a temperature of water-steam mixture of 105 °C. Later, the internal pressure increases (see Fig. 3), which compensates the compressive thermal stresses, with the total stresses (from the temperature and the power load) being tensile strains eventually. In this regime, the total stresses reach their maximum values when temperature and pressure approach their nominal values: 311 °C and 11.5 MPa, respectively.

It should be noted that the allowance for unsteady thermal stress changes slightly the distribution of stresses. In particular, while heating the boiler, the maximum stresses in the drum occur on its outer surface in the vicinity of point E on the outer holes (see Fig. 2). Thus, the stress at this point (172 MPa) does not exceed the limit of plasticity, therefore, the drum deformation at the planned commissioning of the boiler is of pure elastic nature.

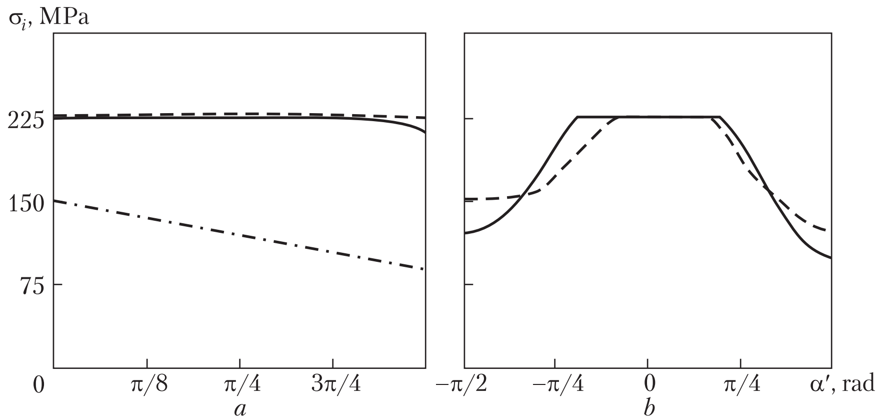

Fig. 9. The stress intensity by thickness in three specific locations (*solid line*: along A_*A; *dashed line*: along E_*E; *dash-dotted line*: along L_*L; see Fig. 2) and on the edges of outer holes on the inner (*solid lines*) and the outer (*dashed lines*) surfaces of the drum (see Fig. 9, *b*)

The intensity of the stress at point E_* on the inner surface, where the maximum compressive stress appear at the beginning of heating of the boiler, is 160.4 MPa. Analyzing the nature of changes in the drum stress state during the planned commissioning mode, it was concluded that at this point, the maximum stresses during the cycle of planned commissioning mode range from –80 MPa («minus» means compression) to 160.4 MPa. The amplitude of stress fluctuations during the cycle of planned commissioning at point E is smaller (–20 MPa to 172 MPa). Subsequently, the amplitudes of stress fluctuations during the cycle were used for determining the accumulated damage of the drum metal.

Almost the same amplitude of stress fluctuations during the cycle of planned commissioning can be obtained for heating the water-steam mixture at a rate of 1.6 °C/min (see. Fig. 3). At this heating rate a lower compensatory effect of thermal stresses is reported. As a result, the total maximum stress when approaching to the stationary operational mode at point E is equal to 194 MPa. The maximum amplitude of stress fluctuations at this rate is registered in the neighborhood of E_* point (from –48.4 MPa to 179.7 MPa).

So, one can see beneficial effects of thermal stress during the planned heating of the boiler (thermal load, to some extent, compensates the power load effect, with total stress while heating the boiler being lower than the corresponding stress from internal pressure effects). A completely different effect of temperature stress is observed during the planned shutdown of the boiler when the inner surface of the drum cools slightly faster than outer one. Therefore, in this mode, the cooling rate of water-steam mixture was limited to 2 °C/min.

At this cooling rate the thermal stresses are quite insignificant. The stress reaches its maximum on the 32nd minute of cooling. The maximum thermal tensile strain like the maximum stresses caused by power load arises on the inner surface, but in the other place (in the vicinity of points D_* and G_*). In addition, due to rapidly dropping pressure at the early stages of planned shutdown of the boiler, at low temperature stresses, the total maximum stresses do not increase unlike those in the case of mechanical loads.

The study of the drum stress-strain state in the stationary operation mode allowing for thermal cycling has showed that if the temperature of working environment inside the drum decreases monotonously (from 311 to 306 °C) a temperature jump gradually increases through thickness of the drum wall (inner surface temperature falls faster). The thermal tensile strain is added to the stress caused by the pressure of water-steam mixture on the inner surface of the drum. When the effect of thermal strain reaches its maximum (the temperature of working environment reaches 306 °C), the maximum stress intensity in the drum caused by the temperature and pressure effects reaches 255.28 MPa (in the neighborhood of E_*, on the inner surface).

Subsequently, the temperature of the water-steam mixture inside the drum starts to grow steadily (up to 316 °C). The distribution of stresses through thickness of the drum changes as a result of the interaction of heat transfer and heat convection processes. After a certain time, the temperature of drum inner surface gets higher than the temperature of the outer surface and compressive thermal stresses appear in the drum thereby partially compensating the stresses caused by pressure. When the impact of temperature growth reaches its peak (at a temperature of water-steam mixture of 316 °C), the total stress in the vicinity of the drum's most loaded point (which is the E_* point on the hole, see Fig. 2) reaches 218.52 MPa. The maximum difference of stresses per one step of this thermal cycle in the drum appears in the neighborhood of E_* point and is equal to 36.76 MPa.

In the emergency shutdown mode the maximum stress in the drum on the 79th second after the shutdown. The maximum temperature difference through the drum thickness at this time is observed in the places that are mostly distanced from the hole and amounts to 64.3 °C (310.9 °C, on the outer surface of the drum and 246.6 °C, on the inner surface). The temperature gradient through the thickness in the vicinity of the holes is much lower (about 25 °C). The maximum amplitude of stress fluctuations during the emergency shutdown cycle is 275.9 MPa.

Thus, the study of the stress-strain state of the drum in the operational modes has showed that in the main cycles of loading the intensity of stresses in the metallic casing does not exceed the yield strength, except for small areas in the vicinity of the holes in the drum volume. In these areas of stress concentration, in the course of exploitation, the metal «works» under conditions of low-endurance fatigue which can cause accumulation of damage and cracks. At the same time, in the areas of plastic deformation the metal gets harder. When unloading, the residual stresses of opposite sign appear in the metal and under the repeated load the deformation is of the elastic character (no secondary plastic deformation of the opposite sign appear). Therefore, the metal may be damaged only when the stresses reach their maximum for the first time i.e. when the metal gets used to the cyclic loading. Therefore, if there is no explicit damage of the drum its lifetime can be prolonged.

The obtained values of maximum stress amplitude fluctuations in the drum for the considered modes of operation make it possible to evaluate theoretically its service life. To do this, let us associate the maximum amplitude of stress fluctuations during one cycle of loading σ_a (planned commissioning-shutdown, emergency shutdown, hydraulic or thermal tests in the stationary mode of boiler operation) with the corresponding number of damage cycles N by the formula [3]

$$\sigma_a = \frac{aE_T}{n(4N)^{m_1}} + \frac{b}{n(4N)^{m_2} + \frac{1+r}{1-r}}, \qquad (1)$$

where $r = \sigma_{min}/\sigma_{max}$ is stress skewness coefficient (σ_{min} and σ_{max} are respectively the minimum and maximum stresses in the load cycle under review); E_T is modulus of elasticity at the temperature of operation mode under review; n is yield factor (which takes into account the level of material degradation); m_1, m_2, a, and b are characteristics of the material defined by the limit of metal tensile strength break σ_B^T and relative narrowing of metal Ψ_T at the working temperature.

Consequently, when determining the number of damage cycles N it is very important to have sufficiently accurate values of stresses, insofar as according to (1) this is just the maximum stress amplitude that determines the allowable number of cycles for a particular mode of operation, which is used for calculating the final evaluation of accumulated damage.

Let us determine the allowable number of cycles for each of the four operation modes under consideration using the formula (1). Then, the total value of accumulated damage for four non-stationary operation modes [3] is as follows:

$$A = 2\frac{n_{ss}}{N_{ss}} + 0.08\frac{n_{ts}}{N_{ts}} + \frac{n_{ht}}{N_{ht}} + \frac{40n_{ss} + \omega\tau}{N_{tc}}, \quad (2)$$

where n_{ss}, n_{ts}, and n_{ht} are the numbers of scheduled commissionings and shutdowns, emergency shutdowns, and hydraulic tests of the drum for the entire period of its operation; N_{ss}, N_{ts}, N_{ht}, and N_{tc} are allowable number of cycles N determined from the corresponding fatigue curves (1) for planned commissionings and shutdowns, emergency shutdowns, hydraulic tests, and temperature fluctuations during the stationary operating mode; τ is time of drum operation (in hours).

According to best practice of the industry [3] for determining the parameter of accumulated metal damage of the drum under the conditions that take into account the temperature fluctuations during the boiler operation in the stationary mode, the planned commissioning and shutdown, emergency shutdown of the boiler, and hydraulic tests it was established that the total parameter of accumulated metal damage of the drum A = 0.6. The planned shutdown and the planned commissioning have the most significant shares in the accumulated damage (0.324 and 0.215, respectively). The resulting theoretical estimate of drum accumulated damage can be interpreted as the remaining service life of the drum is about 40%, as of today.

Thus, on the basis of strength and cyclic fatigue calculations for the boiler drum at Dobrotvir TPP of *DTEK Zakhidenergo,* PJSC, given a satisfactory condition of its metal, the absence of unacceptable defects, the compliance of metal test results with the requirements of applicable regulations, and the drum trouble-free operation over the whole period of its exploitation it is concluded that the boiler drum of unit 6 can operate for further 50, 000 hours until its total in-service time reaches 350 000 hours, at the following nominal parameters: p = 11 MPa, T_w = 311 °C, subject to the requirements NPAOP 0.00-1.60-66 Regulations for Construction and Safe Operation of Steam and Hot Water Boilers.

Using the proposed approach the possibility and conditions of further operation have been studied for the TP-100 type boiler of unit 5 at the Burshtyn TPP of *DTEK Zakhidenergo*, PJSC, which has exhausted its fleet life. The study was conducted for 16 HNM steel under pressure that varies from the nominal working value (15.5 MPa in the case of stationary operating mode and 19.4 MPa in the case of hydraulic test mode) to zero (depending on the proposed mode of boiler operation) with the following numerical parameters: R_1 = 0.9 m, R_2 = 0.995 m, r = 0.0645 m, $2L_y$ = 1.12 m, L_φ = = 0.13 m; L = 22.4 m; m = 6; and T_0 = 340 °C.

During the stationary operating mode the temperature of working environment fluctuates with an amplitude ΔT = 20 °C and a frequency ω = = 6.6836 1/h around working temperature T_0. For the emergency shutdown T_S is equal to 240 °C.

The design model of TP-100-type boiler drum (in a similar way as for TP-10 type) is a 3D hollow cylinder with spherical bottoms at the edges. Inside the cylinder body there are periodic series of six inlets and outlets having radius r (a fragment of finite-element division of the neighborhood of the area around the holes is showed in Fig. 10).

As a result of computational experiment the maximum stresses have been estimated quantitatively and their scope per a cycle of stationary operating mode has been defined with thermal cycling, scheduled commissioning and shutdown, hydraulic test, and emergency shutdown taken into consideration.

Rational geometric shapes have been proposed for sampling the defective metal in the vicinity of areas with cracks in the body and on the holes of the drum and fitting [9, 10], which made it possible to reduce the level of stresses in them below the acceptable one. The fitting tubes in which the working stresses in the vicinity of the sample exceed the allowable ones have been recommended to be replaced. The results obtained underlie the

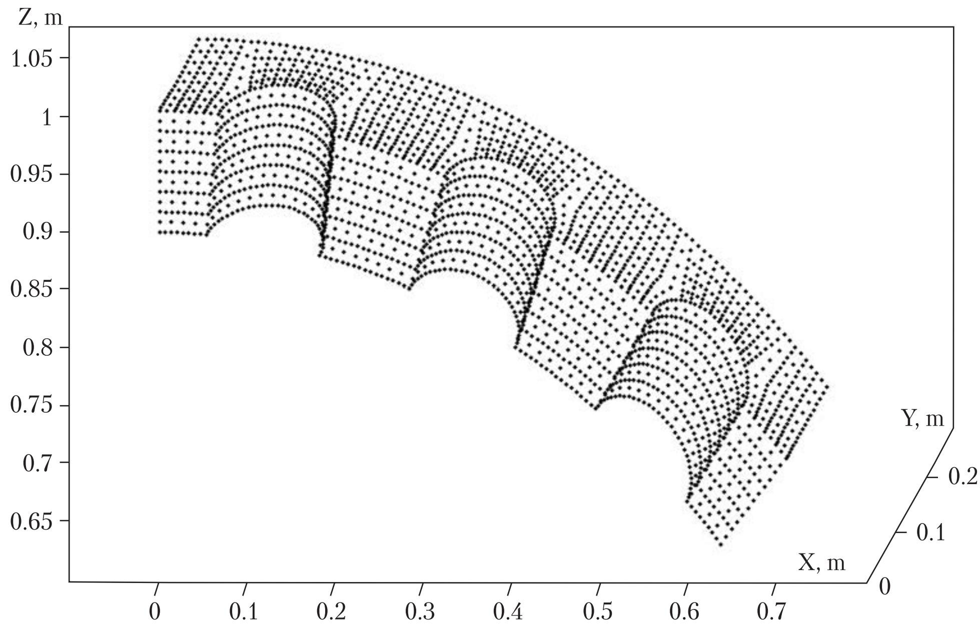

Fig. 10. Units of fragment of finite element division in the vicinity of holes

recommendations for repair of the drum to extend its service life.

The parameter of drum metal accumulated damage has been defined. Proceeding from this parameter the remaining service life of the drum is estimated as 38%. The thermal cycling in the stationary operating mode and the planned commissioning and shutdown are established to have the most significant contribution to the accumulated damage (38.8% and 22.5%, respectively). The theoretical evaluation of the reported accumulated damage for the drum which exhausted its fleet life substantiated the possibility of extending its service life by 50 000 hours in compliance with requirements [3] and has been implemented at the Burshtyn TPP.

It should be noted that it is very difficult to obtain the adequate stress values for the elements of power engineering equipment having complex geometric shapes with holes using the simplified computational models commonly practiced in the energy sector. Often, this approach gives incorrect results. In particular, taking into account the thermal cycling or emergency shutdown the stress amplitude in the stationary operating mode according to the standard industrial practice [3] is determined by the simple formula:

$$\sigma_a = 2E_T \alpha_T \Delta T, \tag{3}$$

where E_T is Young's modulus (at working temperature); α_T is linear coefficient of thermal expansion (at working temperature); ΔT is temperature gradient.

Proceeding from formula (3) having estimated the stress state of the drum under review in the stationary operating mode given the thermal cycling it is established that σ_a = 101.9 MPa (E_{340} = = 191 GPa; α_{340} = 13.341 · 10^{-6} 1/K; ΔT = 20 °C). The corresponding number of allowable cycles N_{ts} = 188,405 (according to (1), $r = -1$). The accumulated damage of the drum that has worked 283,117 hours (n_{ss} = 1209) is

$$A_{tc} = \frac{40n_{ss} + \omega\tau}{N_{tc}} = \frac{40 \cdot 1209 + 6.6836 \cdot 283117}{188405} = 10.3. \tag{4}$$

This means that the drum under review has exceeded 10 times the allowable service life while working in one operating mode (without any scheduled commissioning and shutdown, emergency shutdown, and hydraulic tests), while using the proposed refined methodology for estimating the stress state it is established that $A_{tc} = 0.388$.

Naturally, the advanced methods for quantitative estimation of the stressed elements of power engineering equipment are effective means to extend their service life.

The developed approach has been used to study the possibility of re-determining the service life of heat exchangers for automatic pond cooling at units 1 and 2 of *ZAES* plant.

Hence, the reliable software based on refined mathematical models and methods makes it possible to analyze the behavior of structural elements of power engineering equipment under operating conditions and to adequately estimate the possibility of extending its service life.

REFERENCES

1. Decisions of Scientific Conference «Diagnostics and Extension of Service Life of Power Engineering Equipment Elements at Trypillia TPP of 27.02.2013. Ministry of Energy and Coal Industry of Ukraine (in Ukrainian).
2. Akimov, A: Chornobyl NPP, Vuhlehirsk TPP...Who is the Next? *Dzerkalo Tyzhnia*, 15 (2013) (in Ukrainian).
3. Guidelines СОУ 40.1-21677681-02:2009. (2009). Procedure for Extending Service Life of High-Pressure Boiler Drums. Kyiv: «Industrial Investment Fund for Energy Sector Development», Power Engineering Companies Association (in Ukrainian).
4. Simulation and Optimization in Thermomechnics of Conductive Inhomogeneous Bodies. In Burak, Ya.Y. and Kushnir, R.M. (Eds.). Volume 4: Hachkevych, O.R. and Drobenko, B.D.: Thermomechnics of Magnetized Conductive Thermal Sensitive Bodies. Lviv: SPOLOM, 2010 (in Ukrainian).
5. Allen, D.H. and Heisler, W.E.: A Theory for Analysis of Thermoplastic Materials. *Computers & Structures*, 13, 129—135 (1981).
6. Zienkiewicz, O.C., Wood, W.L., and Nine N.W.: A Unified Set of Single Step Algorithm. Part 1: General Formulation and Applications. *Int. J. for Num. Meth. in Eng.*, 20**,** 1529—1552 (1984).
7. Key Guidelines for Inspection and Techniques of Repairing High-Pressure Boiler Drums Made of 16GNM, 16 GNMA and 22K Steel. (1978). Moscow: Publishing House of the Ministry of Electrification of USSR and the Ministry of Power Engineering (in Russian).
8. Stationary Boilers and Pipelines for Steam and Hot Water. Standards for Strength Calculation. OST 108.031.08-85 — OST 108.031.10-85. (1987). Moscow: Publishing House of the Ministry of Power Engineering (in Russian).
9. Nemchenko, V.I., Vasylyk, A.V., Zadorozhnyi, M.V. et al.: Stress Optimization of Geometric Parameters of Defect Samples for High-Pressure Boiler Drums. *Power Engineering and Electrification*, 6, 22—28 (2011) (in Ukrainian).
10. Nemchenko, V.I., Vasylyk, A.V., Zadorozhnyi, M.V. et al.: Refined Computation of Stressed State of Fitting Tubes with Local Defect Samples. *Power Engineering and Electrification*, 10, 17—22 (2010) (in Ukrainian).

Б.Д. Дробенко, С.Ф. Будз, В.И. Асташкин

Институт прикладных проблем механики
и математики им. Я.С. Пидстрыгача
НАН Украины, Львов

ОЦЕНКА ПРИГОДНОСТИ ЕЛЕМЕНТОВ ЭНЕРГЕТИЧЕСКОГО ОБОРУДОВАНИЯ К ЭКСПЛУАТАЦИИ СВЕРХ ПАРКОВОГО РЕСУРСА

Предложен подход к определению напряженно-деформированного состояния действующих элементов котлоагрегатов и исследование возможности их дальнейшей эксплуатации. Исследовано состояние барабанов котлоагрегатов блоков № 6 Добротворской и № 5 Бурштинской тепловых электростанций ПАТ ДТЕК «Захидэнерго», которые исчерпали свой парковый ресурс. Разработаны рекомендации относительно улучшения технологии их ремонта и сделан вывод о возможности продления сроков их эксплуатации.

Ключевые слова: математическое и численное моделирование, прочность и эксплуатационный ресурс.

Б.Д. Дробенко, С.Ф. Будз, В.І. Асташкін

Інститут прикладних проблем механіки
і математики ім. Я.С. Підстригача, Львів

ОЦІНКА ПРИДАТНОСТІ ЕЛЕМЕНТІВ ЕНЕРГЕТИЧНОГО ОБЛАДНАННЯ ДО ЕКСПЛУАТАЦІЇ ПОНАД ПАРКОВИЙ РЕСУРС

Запропоновано підхід до визначення напружено-деформованого стану діючих елементів котлоагрегатів та дослідження можливості їх подальшої експлуатації. Досліджено стан барабанів котлоагрегатів блоків № 6 Добротвірської та № 5 Бурштинської теплових електростанцій ПАТ ДТЕК «Західенерго», які вичерпали свій парковий ресурс. Розроблено рекомендації щодо покращення технології їх ремонту та зроблено висновок про можливість подовження термінів їх експлуатації.

Ключові слова: математичне й чисельне моделювання, міцність і експлуатаційний ресурс.

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