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THERMAL SHOCKS AND FATIGUE LIFE OF HELICALLY COILED ONCE-THROUGH STEAM GENERATOR TUBES IN TRANSITION BOILING ZONE

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Abstract. Several computational programs were established and used for "direct" computational modelling of strained metal condition in the transition zone of once-through steam generator under random high cyclic heat transfer intensity oscillation resulting from transient rivulet formation with superimposed low cyclic zone fluctuation due to hydraulic instability. The estimations of thermal stress oscillation and metal fatigue life were also provided for a particular design of a steam generator, depending on operational parameters, but are intended to be generally applicable. Finally, stress oscillation intensity and tube metal fatigue life were estimated depending on operational parameters in the transition zone such as heat flux, inside pressure, flow mass velocity, and heat of evaporation that were combined into dimensionless criteria. A conclusion was drawn that for a particular steam generator design, tube fatigue life is sufficient for practical application.

Key words: high cyclic fatigue, transition boiling, once-through steam generator, helically coiled tubes, thermal stress oscillation.

1. INTRODUCTION

This research is based on a once-through steam generator designed for heat utilization. The generator was assembled from helically coiled tubes with a small coil diameter. Helium under pressure of 4 MPa was used as a source heat carrier. The heat capacity of the generator was 50 MW. As compared to straight tube surfaces, the helically coiled tubes allow for the creation of a compact and economical steam generator heating surfaces because of more intensive heat transfer. Unfortunately, an increase in heat transfer may result in an increased magnitude of thermal stress in the transition boiling zone, where transition from nucleate to film boiling (or Departure from Nucleate Boiling (DNB)) and thermal shocks due to random rewetting of the steam-cooled tube surface by water rivulets take place, with subsequent shortening of tube life. The aim of this research is to analyze tube metal strained conditions in the transition boiling zone and to estimate cyclic fatigue life.

2. NUMERICAL SIMULATION PROCEDURE

Numerical investigation of this problem includes three main phases: first, location of the transition boiling zone must be fixed and the governing parameters of heat transfer determined; second, temperature oscillations must be modelled due to the fluctuation of random heat transfer coefficient as well as induced thermal and full stress oscillations; third, cyclic fatigue cumulation in the tube metal must be analyzed and fatigue life estimated. Finally, it seems important for engineering practice to estimate the dependence of temperature and stress oscillation intensity on heat transfer parameters in the transition boiling zone. Appropriate computer programs were developed according to the above-mentioned phases, and metal life was analyzed depending on the integrated dimensionless criteria describing heat transfer in the transition boiling zone.

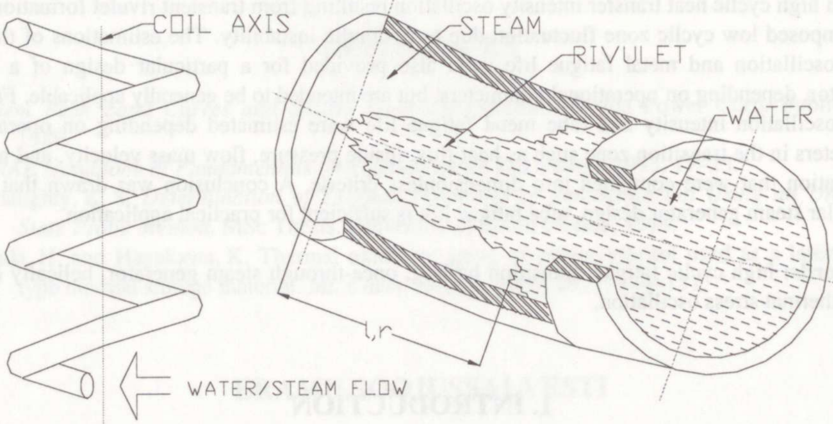

Fig. 1. Schematic presentation of the DNB zone in a helically coiled tube.

Figure 1 shows a schematic presentation of the DNB zone in a helically coiled tube. Tube perimeter prior to the DNB zone is entirely covered by a liquid film; the core of the tube contains a steam flow. The entire tube after the DNB zone is filled with steam flow. Temperature oscillations do not exist in the above-mentioned tube regions. Finally, the tube surface at the DNB zone is covered with rivulets at the extent equal to the rivulet length. High frequency temperature oscillations occur here due to random appearance of rivulets changing to dry patches. Because of the DNB zone shifting (due to hydraulic instability or changing of generator load), low frequency oscillations occur in that tube region. A sophisticated phenomenon results from superimposition of low and high frequency oscillations. Experimental research carried out on the analogue device showed high frequency, equal to 1 Hz and low frequency, equal to 1/60 Hz*.

The *first phase computer program* was established to fix the DNB zone location in a particular generator. The program was based on the algorithm of once-through steam generator design proposed in [1]. The iterative technique was used to determine the precise location of the DNB zone. The program provided for the heat transfer parameters for generating tube region at the regular DNB location (at full generator load, displacement due to hydraulic instability is absent).

The *second phase computer program* was established to obtain temperature and stress distribution in a tube at the regular DNB location under high and low fluctuations of heat transfer intensity. One-dimensional energy equation used for temperature field determination is written in the cylindrical coordinates as

$$\rho c \frac{\partial T}{\partial \tau} = \frac{1}{r}\frac{\partial}{\partial r}\left[kr\frac{\partial T}{\partial r}\right], \tag{1}$$

where the appropriate boundary conditions are given by

$$-k\frac{\partial T}{\partial r}\bigg|_{r=R_1} = h_1\left[T(R_1,\tau)-T_1\right], \tag{2a}$$

$$-k\frac{\partial T}{\partial r}\bigg|_{r=R_2} = h_2\left[T(R_2,\tau)-T_2\right]. \tag{2b}$$

Here ρ is the metal density of the tube; c is the specific heat of tube metal; T is current metal temperature; τ is current time; r is tube radial coordinate; k is the thermal conductivity of the metal; R is tube radius; h is the heat transfer coefficient; subscript 1 refers to the outside surface of the tube; subscript 2 refers to the inside surface of the tube.

A Crank–Nicolson, semi-implicit, transient, finite difference algorithm was used to solve the resultant energy equation. The temperature fields were then combined with the thermoelasticity equations

* Unpublished data from the Moscow Institute of Nuclear Power Engineering Research and Development (here and further).

$$\sigma_{\theta,T} = \frac{\alpha E}{(1-\nu)r^2}\left[\frac{r^2+R_2^2}{R_1^2-R_2^2}\int_{R_2}^{R_1} rT(r)dr + \int_{R_2}^{r} rT(r)dr - r^2T(r)\right], \quad (3a)$$

$$\sigma_{r,T} = \frac{\alpha E}{(1-\nu)r^2}\left[\frac{r^2-R_2^2}{R_1^2-R_2^2}\int_{R_2}^{R_1} rT(r)dr - \int_{r}^{R_1} rT(r)dr\right], \quad (3b)$$

$$\sigma_{z,T} = \frac{\alpha E}{(1-\nu)}\left[\frac{2}{R_1^2-R_2^2}\int_{R_2}^{R_1} rT(r)dr - T(r)\right], \quad (3c)$$

where σ is stress; α is the coefficient of thermal expansion; E is the modulus of elasticity; ν is Poisson's ratio; subscript θ refers to the circumferential coordinate; subscript r refers to the radial coordinate; subscript z refers to the axial coordinate; subscript T refers to the current temperature.

The numerical integration algorithm was used for their solution. As it is well known, for the frequency less than 5 Hz, the pseudo-stationary stress equations (3) may be used without taking into account the finite velocity of the stress impulse propagation. Experimental data for the generator design show that the main oscillation energy is transmitted in the frequency range of 0.02–1 Hz*. It means that the pseudo-stationary stress equations may be also used for this modelling.

The applicability of one-dimensional scheme in this case was carefully analyzed previously in [2], and it was found accurate enough, while the correct boundary conditions are adjusted. The boundary conditions were applied as follows. Heat transfer parameters at the heating fluid side (2a) were given to be steady state. The boiling crisis and thermal shocks at the water-steam side were simulated by changing the inside limits of the heat transfer coefficient h_2 from $h_2 = h_f = h_{\min}$ that correspond to film boiling until $h_2 = h_n = h_{\max}$ that correspond to nucleate boiling. The magnitudes of the particular heat transfer coefficient h_2 were simulated by a random generator for high frequency oscillations caused by rivulet appearance along the tube perimeter, using the governing equations

$$h_2(n) = RND(n)\times(h_n - h_f) + h_f, \quad (4a)$$

$$\tau(n) = RND(n)\times C_\tau + \tau(n-1), \quad (4b)$$

where n is point number; RND is random number in the interval 0–1; C_τ is the maximum time step; subscript n refers to nucleate boiling; subscript f refers to film boiling.

DNB zone shifting affects the heat transfer coefficient h_2 at the tube region as follows. If DNB zone shifting does not exceed half of the rivulet length (regular DNB zone position means that the analyzing tube region is under the middle of rivulet), the tube zone is effected to high frequency oscillation. If the upward shift of the DNB zone exceeds this value, the tube region is covered entirely with a liquid film under a stable nucleate boiling heat transfer. A similar downward shift leads to a stable transition boiling heat transfer. In both of these latter cases, oscillations do not exist. This variation of the heat transfer phase may be described using the relative rivulet length l_{rel}, defined in the following functional form:

$$l_{rel} = \frac{l}{2L_{max}}, \tag{5}$$

where l is rivulet length and L_{max} is the maximum amplitude of zone fluctuation.

The number of rivulets and the maximum rivulet length depend on several parameters, but as it was experimentally found for the particular helically coiled tubes*, usually the only rivulet takes place here, being placed at the outside of the coil perimeter with $l_{max} \cong 500$ mm. Since due to hydraulic instability, the zone fluctuation amplitude L_{max} is highly variable, it was decided to take the relative rivulet length as an independent parameter. For $l_{rel} \geq 1$, the entire tube perimeter is subjected to high frequency oscillations due to alternate formation of rivulets and dry patches. For $l_{rel} < 1$, the tube perimeter is subjected to either liquid (if the DNB zone shifts upward) or vapour cooling (if it shifts downward). Assuming these boundary conditions, a "direct" solution to the hydraulic instability problem was obtained. For each selected relative rivulet length, the matrix of boundary conditions was set, using Eqs. (4). This matrix was then used as a basis for solving the energy equation (1), while actual time steps when solving Eq. (1) were 10–100 times shorter. Temperature distributions at times (4b) were saved in the matrix for further statistical analysis and used for solving the thermoelasticity equations (3). In addition, full stresses were stored in the matrix. Figures 2 and 3 show the results of the evaluation of temperature oscillations.

Statistical analysis was then completed, comprising the following parameter determinations.

– **Standard deviation** of the stress ordinates is determined as

$$S_\sigma = \sqrt{\frac{\sum_{i=1}^{n} (\sigma_i - \sigma_m)^2}{(n-1)}} \tag{6}$$

and is defined here as oscillation intensity; subscript m refers to mean stress value, and subscript σ refers to stress oscillation.

– **Limiting parameter** is the ratio of the maximum amplitude $A_{\sigma,\, max}$ to the oscillation intensity S_σ

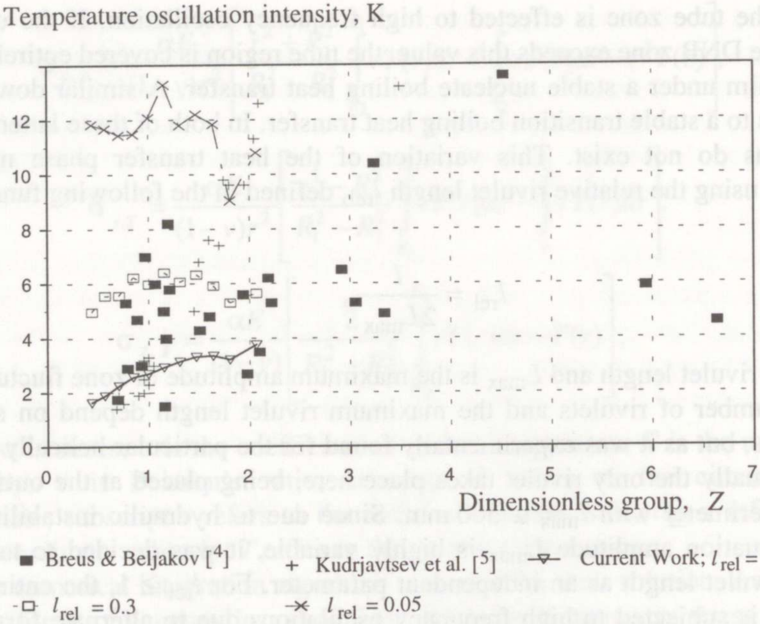

Fig. 2. Oscillation intensity of wall temperature.

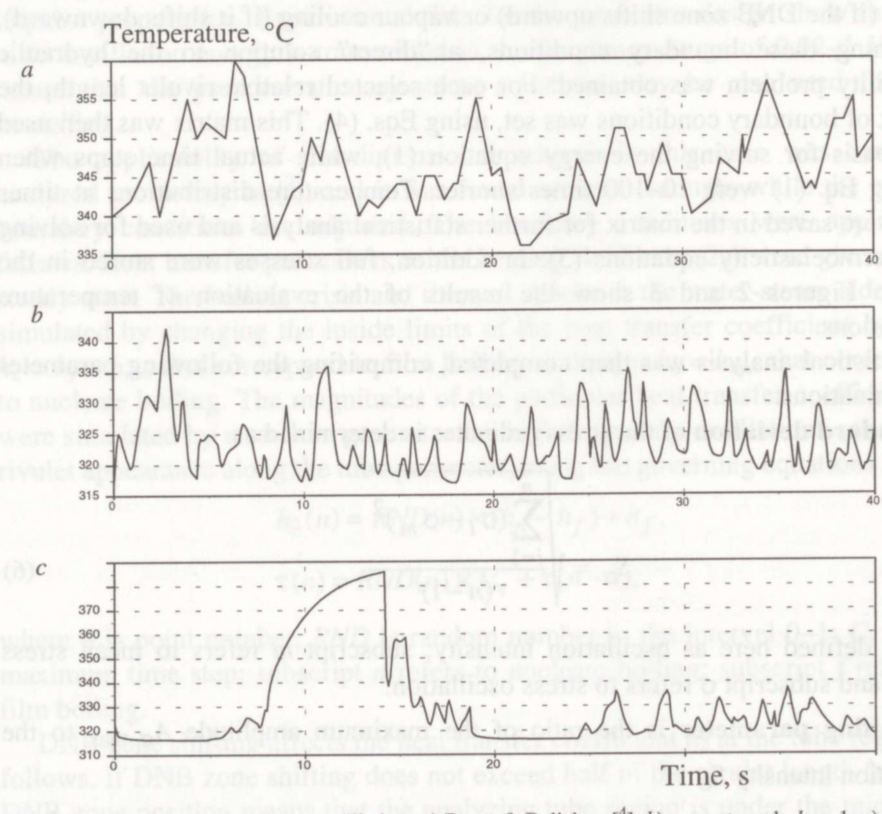

Fig. 3. Curves of wall temperature oscillation: *a*) Breus & Beljakov [4]; *b*) current work, $l_{rel}= 1$; *c*) $l_{rel}= 0.3$.

$$\delta_\sigma = \frac{A_{\sigma,\max}}{S_\sigma}. \tag{7}$$

– **Autocorrelation function** for the current time interval $\tau = m \cdot \Delta\tau$, the autocorrelation function is defined as the correlation between two oscillation curve ordinates over the corresponding time interval

$$R_\sigma(m) = \frac{1}{(n-m)S_\sigma^2} \sum_{i=1}^{n=m} (\sigma_i - \sigma_m)(\sigma_{i+m} - \sigma_m), \tag{8}$$

where m is time step number and $\Delta\tau$ is time increment.

– **Spectral density** of the oscillation distribution is defined as

$$G_t(k) = 2\Delta\tau \left[R_t(0) + 2\sum_{m=1}^{n-1} D_m R_t(m) \cos\left(\frac{\pi m k}{n}\right) \right], \tag{9}$$

where k is frequency step,

$D_m = [1 + \cos(\pi m/n)]$ for $m = 0, 1, 2, \ldots, n$,

$D_m = 0$ for $m > n$.

Statistical parameters (6)–(9), estimated both for temperature and stress oscillations, using appropriate matrices, are applied for oscillation process analysis as well as for comparing temperature and induced stress intensities and frequencies. The main idea is that by computational modelling of temperature oscillations, induced stress and fatigue damage cumulation, a researcher is able to find correlations between the above-mentioned parameters and heat-mass transfer parameters, and establish appropriate governing equations. These equations may then be used to predict the fatigue life of operating equipment on the basis of the temperature oscillation curve, fluid flow and the statistical parameters of heat transfer, and tube metal properties. As it will be shown later in this paper, the dimensionless heat-mass transfer parameter may be successfully used for rough prediction of fatigue damage as well. More precise prediction of stress oscillation parameters under a particular temperature oscillation set, induced by superimposition of high frequency rivulets and dry patch appearance and low frequency DNB zone shifting, is available by using the following transmission coefficients:

– the **transmission coefficient** of oscillation intensity defined as

$$K_s = \frac{1-\nu}{\alpha E}\frac{S_\sigma}{S_t}, \tag{10}$$

where subscript t refers to temperature oscillation.

– and the frequency **transmission coefficient** defined as

$$K_f = \frac{f_\sigma}{f_t} = \frac{\theta_t}{\theta_\sigma}, \tag{11}$$

where f is frequency; θ is the period of oscillation.

Both coefficients, (10) and (11), were estimated on the basis of modelled temperature and stress oscillations and may be used to predict operational stress oscillations, using statistical parameters of operational temperature oscillation curves.

The stress data set was then used in the *third phase program* to estimate the cumulated cyclic fatigue life. As it was described earlier, temperature and stress oscillations are due to the effect of the superimposed high and low frequency fluctuation of the heat transfer intensity. It means that all the loading stress cycles have different mean values, σ_m, and amplitudes A_σ. Fatigue damage in this case was estimated at each particular loading level on the basis of the bilinear fatigue life curve of the tube material defined by the following equation:

$$N = N_1 \left(\frac{\sigma_{-1}}{A_\sigma} \right)^{m1} \quad \text{for} \quad A_\sigma \geq \sigma_{-1}, \tag{12a}$$

$$N = N_1 \left(\frac{\sigma_{-1}}{A_\sigma} \right)^{m2} \quad \text{for} \quad A_\sigma < \sigma_{-1}, \tag{12b}$$

where N_1 is base number of cycles in the fatigue equation, $m1$ and $m2$ are exponents.

This equation gives the number of cycles N to fatigue crack initiation for this material under symmetrical load with the cycling amplitude A_σ. The equivalent amplitude $A_{\sigma,eq}$, used instead of A_σ for actual asymmetric stress oscillations, is obtained according to the following governing equation:

$$A_{\sigma,eq} = A_\sigma - \Psi_\sigma \sigma_m, \tag{13}$$

where ψ_σ is stress cycle asymmetry coefficient.

This defines an equivalent amplitude of the symmetrical stress, which would induce the same fatigue damage as an actual asymmetrical set of stresses having a mean value of σ_m, the stress cycle asymmetry coefficient ψ_σ, and the amplitude A_σ.

The parameters of material fatigue (the fatigue endurance limit σ_{-1}, the base number of cycles N_1, and the stress cycle asymmetry coefficient ψ_σ) were determined experimentally*. The damage summation for all loading levels was provided, using the Manson double-linear damage rule [3]. The main goal of this paper is to model the DNB phenomenon in a helically coiled once-through steam generating tube, to compare the found temperature oscillations with the experimentally measured ones, using an analogue operating device, to identify the type of modelled temperature oscillations and analyze their statistical parameters, to evaluate induced thermal stress oscillations and their correlations with definite heat-mass flow parameters, and to estimate the influence of the latter parameters on the tube material fatigue life.

3. RESULTS AND DISCUSSION

As noted earlier, the amplitude and frequency of temperature oscillation depend on the design and heat transfer parameters. Induced thermal stresses depend on the same parameters. Experimental data, analyzed by Breus & Beljakov [4], Kudrjavtsev et al. [5] and Kirillov et al. [6], have determined the most effective parameters. These are flow mass velocity, pressure, and heat flux. In all references, a decrease in the oscillation intensity of the tube wall temperature with the flow mass velocity and fluid pressure increasing, was observed. Kudrjavtsev et al. [5] explains this phenomenon as follows. With both flow mass velocity and pressure increasing, critical quality decreases, leading to an increase of liquid droplet concentration in the core and to an intensification of liquid film irrigation. Consequently, the boiling heat transfer coefficient of the film increases, and the difference between nucleate and film heat transfer intensity decreases, leading to a decrease in oscillation intensity. With the heat flux increasing, oscillation intensity increases to a certain maximum and then decreases. This phenomenon can be explained as follows. In the DNB zone, a liquid film breaks into rivulets with random boundaries leading to a heat transfer coefficient (HTC) variation. Under a low heat flux, these variations are small, and so is the intensity of temperature oscillation. With heat flux increasing, temperature oscillation intensity increases to a certain maximum. The following heat flux increase leads to the full evaporation of the rivulets so that above the DNB zone, heat transfer to the steam flow predominates. Temperature oscillation becomes unstable with low frequency and low intensity. The same type of HTC variation and induced temperature oscillation exists if the DNB zone itself shifts up and down along the steam generating tube due to hydraulic instability. This phenomenon will be analyzed later. On the basis of experimental data analysis, Kudrjavtsev et al. [5] proposed the dimensionless number $Z = 10^4 q\ (1 - P/P_{cr})\ /\ (r \cdot G)$, which allows determination of the influence of these parameters on temperature oscillations; here r means evaporation heat; q is heat flux; P is stream pressure; P_{cr} is critical water/stream pressure; G is flow mass velocity.

This influence was studied, using the model of a once-through steam generator with a helium-heating carrier medium, and the *first phase computer program*. The heat transfer parameters in the DNB zone were determined, using the heat balance calculation of a steam generator realized in a special Turbo BASIC computational program according to the algorithm in [1]. The Code prescribes the governing equations for the evaluation of heat carrier properties, depending on their temperature and heat transfer rate, as dependent on the fluid properties and flow velocities. This program allows for the estimation of both heat carrier's temperature distributions along heat transfer surfaces and appropriate thermal properties of the heating and cooling fluids for the operating values of temperature and pressure, conductive heat transfer through the tube wall and the replacement of the three heat transfer regions of a steam generator: economizer, evaporator, and super heater. By the program, more accurate values of heat transfer parameters and region boundaries are achieved iteratively. The DNB zone replacement is defined, using critical quality value according to the governing equation in [1] for helically coiled steam generating tubes. Convective heat transfer parameters of the appropriate DNB zone (including the nucleate and film boiling HTCs) for a particular steam generator design are then used for metal temperature and stress modelling. The effective frequency of high frequency oscillations was assumed equal to 1 Hz. The period of the low frequency DNB zone shifting was assumed equal to 60 s. The value of critical quality was simulated from 0.1 to 1. For these conditions, the dimensionless number Z was in the range of 0.4–2.1 (Kudrjavtsev et al. [5] had Z number in the range of 0.5–3.5). Comparative results of the oscillation intensity of the calculated metal temperature and those of the experimental data are shown in Fig. 3. It must be noted that there is a significant scattering of experimental data in the range of high Z values ($Z > 2$).

The most significant computed data divergence is in the range of low Z numbers and low relative rivulet values (see $l_{\text{rel}} = 0.05$). One of the reasons may be the following. In computer modelling, the hydraulic instability may be defined by defining the amplitude of the DNB zone fluctuation or relative rivulet length. In both cases, it means that the hydraulic instability phenomenon is "directly" modelled. In physical modelling, such direct modelling is impossible. The indirect way of modelling the hydraulic instability is possible by changing flow mass velocity, heat flux, and pressure. This changing leads to the DNB zone shifting and to the corresponding change in temperature oscillation intensity. As a result, part of the parameters in physical modelling becomes indeterminate, and more accurate comparison is not possible.

The statistical analysis of temperature oscillations confirms the explanation. Figure 3*a* shows the experimental data of Breus and Beljakov [4]. Figure 3*b* presents the computational results for critical quality $X_{\text{cr}} = 0.9$ for the relative rivulet length $l_{\text{rel}} = 1$, and Fig. 3*c* for $l_{\text{rel}} = 0.3$. The appropriate autocorrelation functions and spectral density of surface temperature oscillations are shown in Fig. 4.

It may be noted that both experimental data of Breus and Beljakov [4] oscillation curve and its statistical parameters demonstrate that the wall temperature difference oscillations are a pure random oscillation process as the authors declare and approximately the same as calculated with $l_{\text{rel}} = 1$ (Fig. 3*b*).

Probably a slight influence of low frequency superimposed oscillations takes place here, since it is calculated for $l_{rel} = 0.3$ (Fig. 3*c*). These results show that the computational model is able to separate the influence of the definite particular parameters on the resulting temperature oscillation intensity.

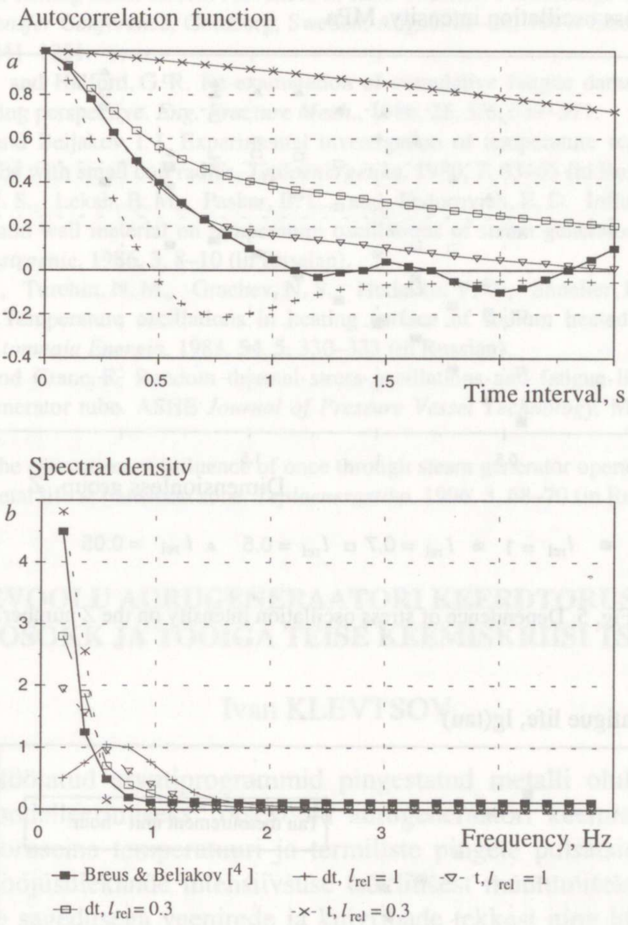

Fig. 4. Autocorrelation function of wall temperature oscillation (*a*) and spectral density (*b*).

At the same time, the dependence of the stress oscillation intensity on the dimensionless parameter *Z* can be determined (Fig. 5). It has practically the same effect as temperature oscillations (intensity increases with *Z* increasing), with definite dependence on the rivulet length. Fatigue life under these stress oscillation intensities is shown in Fig. 6. Its general trend is a slow decrease with *Z* increasing, but a sharp decrease occurs under the conditions of the fluctuations superimposed by the low frequency DNB zone, where $l_{rel} < 1$. A sharp decrease takes place practically for all the amplitudes of DNB fluctuation and depends only on the Z number.

A detailed description of the computational modelling algorithm may be found in [2, 7, 8] and is out of the scope of this paper.

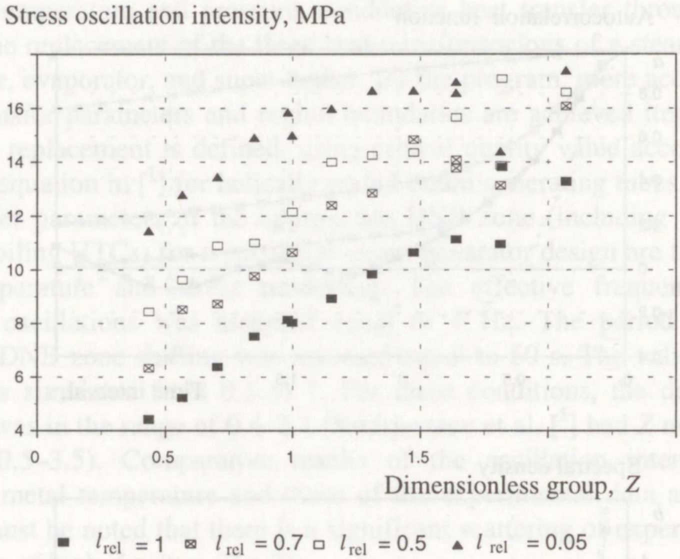

Fig. 5. Dependence of stress oscillation intensity on the Z number.

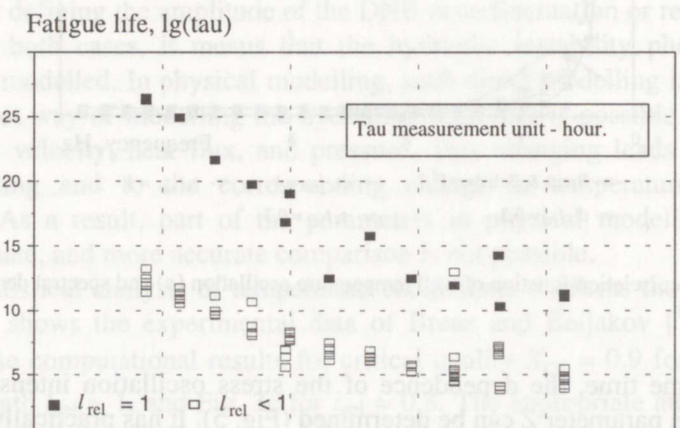

Fig. 6. Dependence of fatigue life under definite stress oscillation intensity on the dimensionless group number Z.

REFERENCES

1. Code 108.031.05-84. *Nuclear Power Plant Heat Transfer Equipment. Heat Transfer & Hydraulic Parameters Determination.* Leningrad, 1986 (in Russian).
2. Klevtsov, I. and Käär, H. Calculation of once through steam generator tube fatigue life in transition boiling zone. Recent Advances in Heat Transfer. *Proceedings of the First Baltic Heat Transfer Conference*, Göteborg, Sweden, August 26–28, 1991. Elsevier, Amsterdam, 1992, 1241–1251.
3. Manson, S. S. and Halford, G. R. Re-examination of cumulative fatigue damage analysis – an engineering perspective. *Eng. Fracture Mech.*, 1986, **25**, 5/6, 539–571.
4. Breus, W. I. and Beljakov, I. I. Experimental investigation of temperature regime of helically coiled tube with small coil radius. *Teploenergetika*, 1980, 7, 63–65 (in Russian).
5. Kudrjavtsev, I. S., Lekah, B. M., Paskar, B. L., and Fedorovich, E. D. Influence of heating method and wall material on temperature oscillations of steam generator heating surface. *Mashinostroenie*, 1986, 3, 8–10 (in Russian).
6. Kirillov, P. L., Turchin, N. M., Grachev, N. S., Hudasko, V. V., Shneller, I., Bitsa, J., and Hum, J. Temperature oscillations in heating surface of sodium heated steam generator model. *Atomnaja Energia*, 1983, **54**, 5, 330–333 (in Russian).
7. Klevtsov, I. and Crane, R. Random thermal stress oscillations and fatigue life estimation for steam generator tube. ASHE *Journal of Pressure Vessel Technology,* May 1994, **116**, 2, 110–114.
8. Klevtsov, I. The estimation of influence of once through steam generator operational parameters on the metal life at transition zone. *Teploenergetika*, 1996, 3, 68–70 (in Russian).

OTSEVOOLU AURUGENERAATORI KEERDTORUSTIKU TERMOŠOKK JA TÖÖIGA TEISE KEEMISKRIISI TSOONIS

Ivan KLEVTSOV

On välja töötatud arvutiprogrammid pingestatud metalli olukorra arvutuslikuks "otsemodelleerimiseks" otsevoolu aurugeneraatori keemiskriisi tsoonis, kus tekivad toruseina temperatuuri ja termiliste pingete pulsatsioonid, mis on põhjustatud soojusülekande intensiivsuse tsüklilisest muutumisest. Viimane on tingitud kõrge sagedusega veenirede ja kuivribade tekkest ning hüdraulika ebastabiilsusest madala sagedusega tsooni liikumisel. Pingete pulsatsiooni hinnangud on saadud ning aurugeneraatori torude tööiga enne metalli väsimist on arvutatud katla konstruktsiooni ja tööparameetrite alusel. Pingete pulsatsiooni intensiivsus ning torumetalli tööiga enne väsimist on välja toodud ka dimensioonita kriteeriumi puhul. Seda kriteeriumi on kirjeldatud kasutades keevkriisitsooni soojusülekande parameetreid, nagu soojusvoo tihedus, soojuskandja masskiirus, aurustumissoojus ja siserõhk. Üks järeldusi on, et aurugeneraatori konstruktsiooni tööiga on küllalt pikk. Pakutud arvutusliku modelleerimise metoodika ja saadud tulemused võivad olla kasulikud rõhu all töötavate keevkihtkatelde torude tööea hindamiseks. Soojusülekande intensiivsus on sellistes kateldes väga suur ja tsükliline väsimus võib oluliselt lühendada kogu katla tööiga.