Analytical Determination of Critical Flaw Dimensions in Steam Generator Tubing

I. Kadenko, N. Sakhno, R. Yermolenko, Nondestructive Examination Training and Certification Facility and Taras Shevchenko National University of Kyiv, Ukraine; O. Kharytonov, O. Kutsenko, Taras Shevchenko National University of Kyiv, Ukraine

1. INTRODUCTION

One of the actual problems of safe and reliable operation of NPP units with pressurized reactors is the development of tube plugging criterion for steam generator tubing. The basis for plugging criterion development is associated with existence of flaws with critical sizes at steam generator tubes. For effective application of this criterion the conditions of flaw instability should be analyzed and critical flaw sizes must be determined. Analytical investigation of flaws stability at steam generator tubes is carried out in this paper for semi-elliptical cracks since some of flaws at steam generator tubes for VVERs are similar to such cracks.

The analytical solution of thermo-elastic problem dealing with the determination of stresses in tubes wall is presented. It makes it possible to perform linear fracture mechanics analysis with the aim to determine the conditions of crack instability.

The approach to develop steam generator tube plugging criteria on the base of obtained results is presented and discussed.

2. PROBLEM FORMULATION

Steam generator is one of the most important elements of nuclear steam supply system with VVERs. Steam generator is safety significant element and is qualified as an equipment of Group B [1] (the second class of safety). It must be mentioned that the percentage of energy losses due to steam generator failures is very high and the problem to develop the programs of steam generator in-service inspection and generation of reasonable criteria for steam generator tube plugging are of great importance. Some of approaches to develop steam generator tube plugging criteria are associated with the existence of the flaws with critical sizes located at steam generator tubing.

The outer fibers of heat-exchange tube wall are subject to stretching at circumferential direction in view of the action of high internal pressure and temperature. The longitudinal fatigue cracks can arise at the outer surface of the tube and grow with time. Therefore it is very important to evaluate the danger associated with such cracks and to predict the tube end-of-life operation time. In our paper the semi-elliptical cracks are considered. It must be mentioned that the real flaws often have a form that is quite similar to semi-elliptical (Fig. 1).

According to fracture mechanics concept the characteristic which can provide a basis for such evaluations is a stress intensity factor (SIF). Determination of SIF dependence on crack parameters under normal operation conditions is the subject of given investigation. The model of defect in steam generator tube (Figure 1) has been developed and analyzed in this paper.

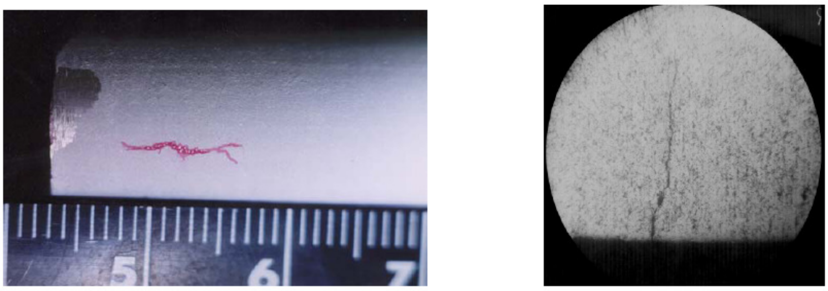

Figure 1 - Defect at steam generator PGV-1000 tube of VVER

The steam generator tube parameters and coolant properties are presented in Table 1.

Table 1 - VVER-1000 steam generator tubing parameters

Inner radius, *a*	6.5 mm
Outer radius, *b*	8 mm
Length, *L*	10-15 m
Inner (primary coolant) pressure, p_{in}	15.7 MPa
Outer (secondary coolant) pressure, p_{out}	6.4 MPa
Inlet primary coolant temperature, $T_{in}(0)$	320° C
Outlet primary coolant temperature, $T_{in}(L)$	290° C
Secondary coolant temperature, T_{out}	280° C
Velocity of the primary coolant within the tube	3.69 m/c
Reynolds number, *Re*	$\approx 4\times10^5$
Prandtl number, *Pr*	0.88
Primary coolant heat conduction, λ	0.55 W/(m·K)
Tube wall material	08Cr18Ni10Ti
Linear expansion coefficient at 300ºC, α	17.4×10^{-6} K^{-1}
Young's modulus at 300ºC, E	1.8×10^{11} Pa
Poisson's ratio, μ	0.3

The most significant problem under investigation is a determination of SIF for points along the front of semi-elliptical crack postulated at outer surface of tube wall under the normal steam generator operation conditions. The algorithm of above problem solution is as follows. At the first step the temperature problem (Figure 2) is solved. The obtained temperature field in tube wall is then used for stress calculations at the second step. The SIF values are calculated using the stress field at the final step of the solution development.

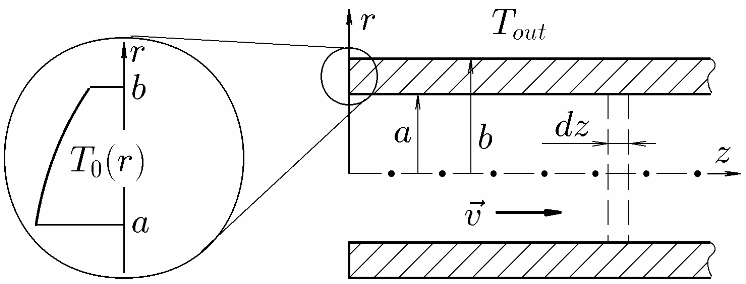

Figure 2 - Temperature field problem for heat-exchange tube

The temperature field in tube wall can be determined from heat conduction equation. In our case the steady-state conditions must be applied. The heat conduction equation reduces to Laplace's equation. In cylindrical coordinate system it can be written in a form:

$$\frac{\partial^2 T}{\partial r^2}+\frac{1}{r}\frac{\partial T}{\partial r}+\frac{1}{r^2}\frac{\partial^2 T}{\partial \varphi^2}+\frac{\partial^2 T}{\partial z^2}=0\,, \qquad (1)$$

where T - is the temperature of tube wall, r, φ, z - are cylindrical coordinates (Fig. 1).

In our case the problem may be considered as axisymmetrical. Based on that $T = T(r, z)$ and

$\frac{\partial T}{\partial \varphi} = \frac{\partial^2 T}{\partial \varphi^2} \equiv 0$.

The equation (1) must be supplemented with the appropriate boundary conditions. Note that the primary coolant motion within heat-exchange tube is strongly turbulent. Due to this feature it can be assumed that the coolant temperature T_{in} varies only in axial direction ($T_{in} = T_{in}(z)$) and is equal to the bulk temperature. The value of heat transfer coefficient α_c can be obtained from the well-known Dittus-Boelter correlation:

$$\alpha_c = \frac{\lambda Nu}{2a}, \qquad (2)$$

where $Nu = 0.023\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4}$ - is Nusselt number.

The value $\alpha_c = 2.8 \times 10^4$ W/(m^2·K), which is obtained from (2), is a very high. So, the boundary condition at the tube inner surface can be specified as

$$T|_{r=a} = T_{in}(z), \qquad (3)$$

It must be mentioned that the difference in inner temperatures corresponding to the boundary condition (3) and convective boundary condition has the order of 0.01%.

The function $T_{in}(z)$ is not preset. So, the condition (3) is not closed. The closed boundary condition can be derived from the energy balance for the small volume of the coolant in the tube with the length dz (Figure 1). The quantity of heat loosed by the coolant is proportional to the variation of it temperature:

$$dq = A\frac{dT_{in}}{dz}dz, \qquad (4)$$

where constant A depends on the coolant parameters and inner diameter of the tube.

From the other hand, the coolant-to-wall heat flux q is governed by Fourier's law:

$$dq = B\left(\frac{dT}{dr}\right)\bigg|_{r=a} dz, \qquad (5)$$

where constant B depends on tube material parameters and inner diameter of the tube.

From (3), (4) and (5), the following boundary condition at the inner surface of the tube can be obtained

$$H\frac{dT}{dr} = \frac{dT}{dz}, \qquad r = a. \qquad (6)$$

The dimensionless constant H can be evaluated using the thermophysical properties of the coolant and tube wall material. However, more preferable way is to select this constant using the outlet coolant temperature value. Taking this approach the value 2×10^{-4} was calculated.

The condition at the outer surface of the tube can be derived using the value of the temperature of the secondary circuit coolant in steam generator T_{out} =280°C.

$$T|_{r=b} = T_{out}. \qquad (7)$$

The conditions (6) and (7) must be supplemented with the condition at the tube tip

$$T|_{z=0} = T_0(r), \qquad (8)$$

where $T_0(r)$ - is initial temperature distribution with the wall depth (see Figure 1). The method for $T_0(r)$ determination is described below.

3. PROBLEM SOLUTION

The solution of the problem (1), (6)-(8) is given in [2] using Fourier's variable separation method and can be written in a form:

$$T(r,z) = T_{out} + \sum_{n=1}^{\infty} A_n U_0(\omega_n r) e^{-\omega_n z}, \qquad a < r < b, \quad z > 0. \tag{9}$$

The eigenvalues ω_n are determined as roots of the equation:

$$H U_1(\omega a) = U_0(\omega a),$$

where

$$U_i(\omega r) = \frac{J_i(\omega r)}{J_0(\omega b)} - \frac{Y_i(\omega r)}{Y_0(\omega b)}, \quad i = 1,2,$$

J_i - is Bessel's function of the first kind and i-th order, Y_i - is Bessel's function of the second kind and i-th order. Coefficients A_n can be determined from the expansion of the $T_0(r)$ by eigenfunctions:

$$\sum_{n=1}^{\infty} A_n U_0(\omega_n r) = T_0(r) - T_{out}, \qquad a < r < b.$$

The expressions for the temperature stresses (analogous to (9)) are also presented in [2].

Therefore the initial temperature profile $T_0(r)$ must be specified for the development of unambiguous solution. We used the simple iteration method for determination of $T_0(r)$. At the first iteration the heat-transfer problem (1), (6)-(8) with the arbitrary temperature distribution $T_0(r)$ is solved. At the second step the real conditions of tube attaching to hot water manifold are taken into account. The part of hot water manifold wall *AEFG* is attached to the part *ABCD* of the tube wall (Figure 3). On the sides *AB* and *BC* the temperature distributions obtained at the first iteration are specified. Besides the inlet primary coolant temperature $T_{in}(0)$ is prescribed on the sides *AG* and *FG*, the secondary coolant temperature T_{out} - on the sides *CD* and *DE*. The linear temperature distribution (from T_{out} at the point *E* to $T_{in}(0)$ at the point *F*) is specified on the side *EF*. The next approximation for $T_0(r)$ is obtained as a solution of the formulated steady-state heat conduction problem (Dirichlet's problem) at the *AD* interval.

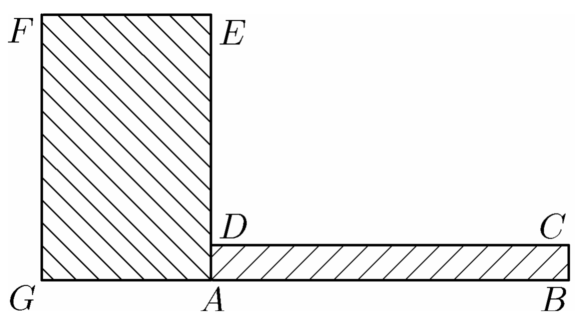

Figure 3 - The geometry of Dirichlet's problem

The appropriate solution accuracy was obtained at the second iteration when the linear distribution $T_0(r)$ (from T_{out} at the point *D* to $T_{in}(0)$ at the point *A*) was used at the first step. The obtained $T_0(r)$ profile (solid curve) is presented at Fig. 4 in comparison with linear profile (dashed line).

It can be concluded that the deviation of $T_0(r)$ from the linear profile is not essential.

The temperature field in the steam generator tube wall is developed using (9). The distribution of the temperature of the inner tube surface with the length is presented at Figure 5. From the Figures 4 and 5 it can be concluded that the temperature gradient in axial direction is negligible in comparison with the gradient in radial direction (the radial temperature gradient is approximately at fifth orders higher). So, in order to determine the temperature stresses in the given cross-section of tube wall the variation of the temperature in axial direction can be neglected.

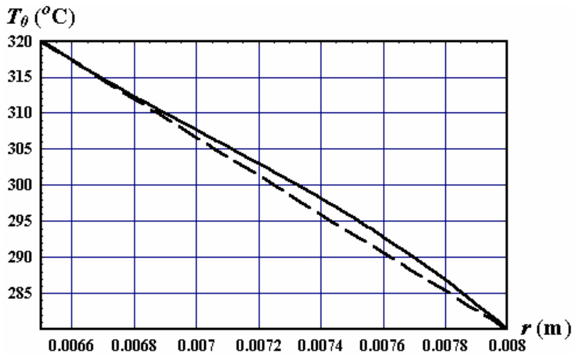

Figure 4 - The initial temperature profile in the wall of the tube

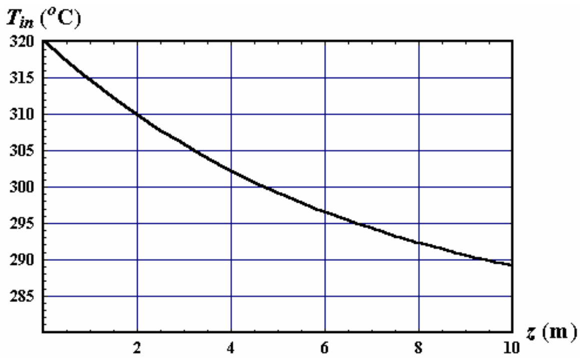

Figure 5 - Inner surface temperature distribution along tube length

The total stresses in steam generator tube wall can be presented as the sum of thermal stresses and the stresses due to pressure action at inner and outer surfaces. The thermal stresses can be determined from the solution of the thermoelastic problem for the hole cylinder with the prescribed temperature at the inner and outer faces is well-known [2, 3]. The distribution of circumferential stresses is expressed as:

$$\sigma_\varphi^T(r) = -\frac{\alpha E \Delta T}{2(1-\mu)}\left(\frac{\log\frac{b}{r} - 1}{\log\frac{b}{a}} + \frac{\frac{b^2}{r^2}+1}{\frac{b^2}{a^2}-1} \right), \qquad a < r < b, \qquad (10)$$

where α - is linear expansion coefficient, E – is Young's modulus, $\mu = 0{,}3$ - is Poisson's ratio, $\Delta T = T_{in} - T_{out}$ - is a temperature drop between inner and outer tube surfaces.

The stresses due to action of inner and outer pressure can be expressed using well-known solution of Lame's problem [3]:

$$\sigma_\varphi^p(r) = \frac{p_{in}a^2 - p_{out}b^2}{b^2 - a^2r^2} + \frac{(p_{in} - p_{out})a^2b^2}{(b^2 - a^2)r^2}, \qquad a < r < b, \qquad (11)$$

where p_{in} – is inner and p_{out} – is outer pressure.

The distributions of circumferential stresses are presented at the Figures 6 and 7 for the tip and the middle part of the tube respectively. The dash-dotted curve corresponds to thermal stresses (10), the dashed curve – to the stresses due to pressure action (11) and the solid curve corresponds to the total stresses (the sum of the stresses of above two types). The thermal stresses calculated taking into account the axial temperature variation (temperature field is determined from (9)) are marked by big dotes. Their deviation from the stresses determined according to (10) is not significant. More essential deviation at the Figure 6 can be explained by the fact of nonlinear temperature distribution with the wall depth at the tip of the tube (see Figure 4).

The obtained stress distributions make it possible to perform the analysis of brittle fracture resistance of the tube with postulated flaw. The flaw is assumed to be semi-elliptical outer crack of length $2l$ and depth d (Figure 8).

ET signal caused by semi-elliptical crack is given at Figure 9.

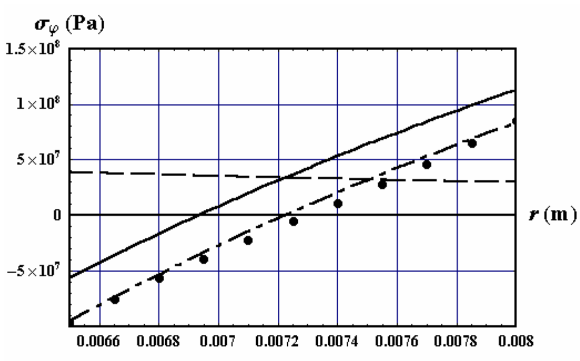

Figure 6 - Circumferential stresses distribution with wall depth at tube tip

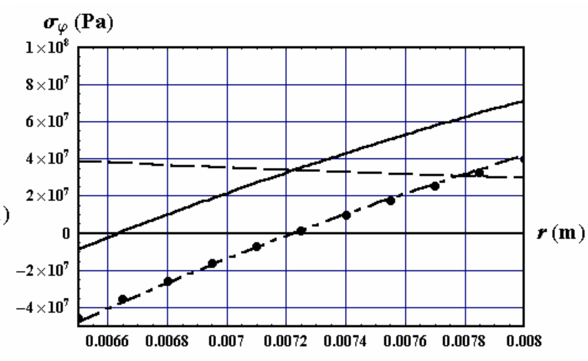

Figure 7 - Circumferential stresses distribution with wall depth at the middle part of the tube (5m from tip)

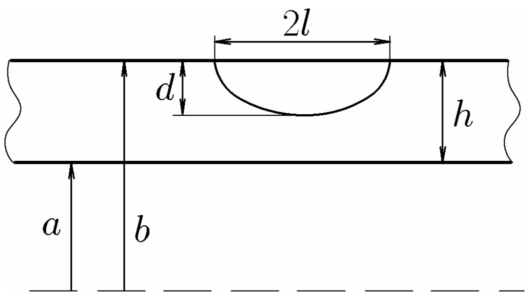

Figure 8 - Outer surface semi-elliptical crack

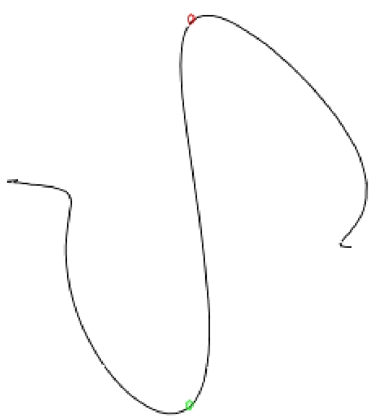

Figure 9 - ET signal caused by semi-elliptical crack

The calculation of the SIF values at the points along the front of the crack can be effectively carried out using weight function method. For the set of polynomial weight functions the SIF table data from [4] can be used. The relative error (maximal deviation relative to maximal value) is approximately equal to 0.25% if the stress distribution with wall depth is approached by the quadratic trinomial

$$\sigma_{\varphi}(x) = a_0 + a_1(b-x) + a_2(b-x)^2, \quad 0 < x < d .$$

Maximal value of SIF along the front of the crack can be used for the determination of crack stability from the condition

$$K_{I\max} < K_{ISCC}, \qquad (12)$$

where $K_{I\max}$ - is maximal SIF value, K_{ISCC} - is the critical SIF under stress corrosion cracking (SCC) conditions. As it was pointed out in [5] for steam generator tubing K_{ISCC} is normally distributed with mean value 12.0 MPa·m$^{1/2}$ and standard deviation 3.03 MPa·m$^{1/2}$. In our paper two acceptable reliability levels 95% and 99% are used. The crack stability conditions corresponding to that levels can be written in the forms:

$$K_{I\max} < 7 \text{ MPa·m}^{1/2} \text{ for the 95\% level,} \qquad (13)$$

$$K_{I\max} < 5 \text{ MPa·m}^{1/2} \text{ for the 99\% level.} \qquad (14)$$

The dependencies of maximal SIF values on the crack parameters (depth and length) are presented at the Figures 10 and 11 for the initial and middle (5m from the tip) parts of the tube respectively. Each curve consists of two parts: upper part corresponding to SIF maximum at the surface point and lower part corresponding to SIF maximum at the deepest point of the crack front. Note that according to the table data from [4] SIF at surface point is no monotone function of the length of the crack with fixed depth.

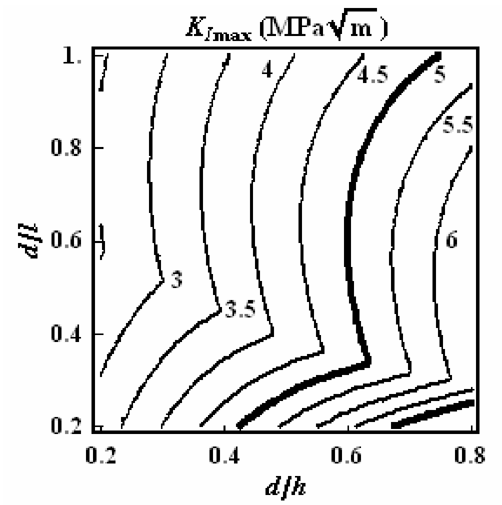

Figure 10 - $K_{I\max}$ dependence on the crack dimensions (location closed to tube tip)

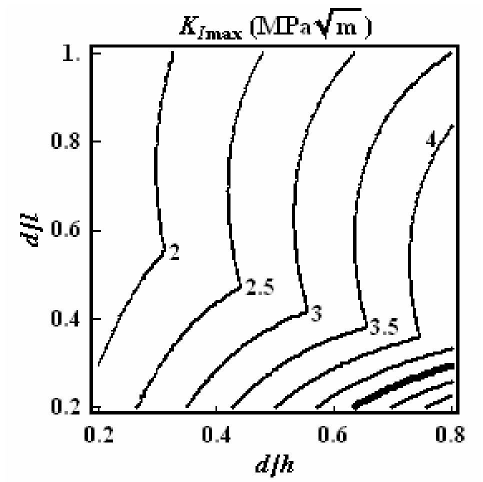

Figure 11 - $K_{I\max}$ dependence on the crack dimensions (location at the middle of the tube)

Bold curves separate the plane of crack parameters on safe and unsafe areas according to the condition (13), (14) for acceptable reliability levels 95% and 99% respectively. The cracks with parameters to the left from the bold curves can be considered as allowable from the point of view of steam generator operation with correspondent acceptable reliability level. So, the presented bold limiting curves can be used for the development of analytical steam generators plugging criteria.

4. CONCLUSIONS

The main results of above investigation are as follows:

1. The model of the steam generator heat transfer tube has been developed. The distributions of temperature and stresses in tube wall have been calculated on the base of this model in the case of steam generator normal operating conditions.
2. The linear fracture mechanics analysis has been carried out. The flaws were assumed to be outer semi-elliptical cracks. The algorithm for the calculation of the SIF values along the crack front has been developed.
3. The limiting curves that separate the plane of crack parameters on safe and unsafe areas according to the brittle fracture condition for acceptable reliability levels 95% and 99% have been developed. The cracks with parameters to the left from the bold curves can be considered as allowable from the point of view of steam generator operation. The presented methodology of limiting curves generation can be used for the development of analytical steam generators tube plugging criteria.
4. The proposed algorithms of SIF calculation can be used for prediction of fatigue crack growth.

5. REFERENCES

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