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ON TEMPERATURE DISTRIBUTION OVER THE EXTERNAL SURFACE OF SCREEN TUBES

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Аннотация—Предложено уравнение, описывающее распределение температур на наружной поверхности трубы однорядного гладкостенного экрана в зависимости от величины падающего из топки лучистого потока, термического сопротивления слоя золы, температуры нагреваемой среды и относительного шага экрана с учетом собственного излучения отложений золы и поглощения отраженных тепловых потоков при равномерном излучении обмуровки.

Решение интегрально-дифференциального уравнения произведено методом конечных разностей на электронно-вычислительной машине «Урал».

Показано, что равномерность распределения температур повышается при увеличении относительного шага, температуры нагреваемой среды, падающего теплового потока и уменьшения термического сопротивления слоя золы на трубах. Результаты расчётов сопоставлены с экспериментальными данными.

NOMENCLATURE

- s, spacing of screen tubes [m];
- D, screen tube diameter [m];
- q_{inc}, heat flux from the furnace to screen based on the plane surface of the screen;
- q_{rev}, heat flux from the screen to the furnace based on the plane surface of the screen;
- ψ, heat value;
- q_{net}, radiant flux absorbed at a certain point;
- T_0, temperature of the heated medium [°K];
- T_r, temperature at any point on the external surface of the tube [°K];
- R, thermal resistance;
- σ_s, black-body radiation coefficient;
- a, emissivity;
- θ, angle of location of a certain point on the tube periphery;
- φ, $= 1 - \sqrt{[1 - (D/S)^2]} + (D/S)\ \mathrm{arctg}\ \sqrt{[(S/D)^2 - 1]}$, angle coefficient of the screen;
- $\bar{q}, \bar{R}, \bar{T}_r, \bar{T}_0$ and $\overline{\Delta\theta}$, incident heat flux, thermal resistance, temperature at the external tube surface, temperature of heated fluid and increment of the angle θ in the machine scale, respectively;
- A_1, $= \dfrac{1 + \cos(\theta + \psi)}{2}$;
- A', $= \dfrac{1 + \cos(180 - \theta + \gamma)}{2}$;
- B_i, $= \dfrac{1 - \sin\beta_2}{2}$;
- $\bar{D}_i$, $= 0{\cdot}1 \cos\alpha \,.\, d\alpha$;
- m, $= 1 - \varphi$.

Direct measurement of intrinsic radiation of screened furnace walls of steam boilers using various fuels and heaters in petroleum refineries using mazuts has shown that sooting of the tube surfaces gives rise to powerful radiant fluxes towards the furnace flame which are commensurable with flame radiation.

The extremely low thermal conductivity of a very thin layer of deposits adjacent to the wall and the rather low mean value of the heat-conduction coefficient of soot decrease the surface heat flux, causing non-uniform peripheral temperature distributions over the external surface of screen tubes, particularly with small values of the relative spacing of the screen.

Non-uniform distributions of temperature, and consequently of heat flux restrict the allowable heat flux of the screen, increase thermal stresses in the tube wall and give rise to coking and undesirable decomposing of petroleum and petroleum products in furnaces of processing plants. The form of the peripheral distribution depends on the screen design. Unfortunately, plots of incident fluxes presented in references [1, 2] are probably constructed on the basis of speculative conclusions and are therefore applicable only to purely qualitative estimations of operating conditions of screen tubes.

In solving the problem of thermal stresses in screens, Charny [3] and Doll-Steinberg [4] considered the peripheral temperature distribution in the tubes with uniform radiation of the furnace and lining. In these works sooting of the external surface, intrinsic radiation of screen tubes, absorption of radiant fluxes from neighbouring tubes and lining were not taken into account, and for this reason the temperature distribution obtained in references [3, 4] may show important differences from the real one.

In works [5, 6] the effective screen temperature is introduced for description of reverse heat fluxes which is obtained from the assumption of uniform peripheral temperature distribution. The effective temperature and temperature of the external layer of sooting [7] cannot describe the performance of the screen.

Moreover, knowledge of the actual peripheral temperature distribution is very important for the analysis of the operation of screen tubes, since this allows calculation of the net heat flux

$$q_{\text{net}} = q_{\text{inc}} - q_{\text{rev}} \tag{1}$$

and the heat value of the screen

$$\psi = \frac{q_{\text{net}}}{q_{\text{inc}}}. \tag{2}$$

The temperature at any point of the surface of a screen tube may be found from the heat-balance equation

$$\frac{T_r - T_0}{R} = q - a\sigma_s T_r^4 \tag{3}$$

from which it follows that

$$T_r = f\left(\theta, q_{\text{inc}}, R, T_0, \frac{S}{D}\right). \tag{4}$$

In the solution of the problem considered the following assumptions are adopted:

(1) Furnace radiation is uniform in all directions;
(2) Lining is adiabatic;
(3) Heat transferred to the lining is radiated uniformly;
(4) Heat resistance of the peripheral layer of sooting does not change and the emissivity of the sooting is constant;
(5) There are no additional peripheral heat fluxes;
(6) No convective heat transfer.

In Fig. 1, a schematic drawing is presented of radiant fluxes which are taken into account in derivation of the equation of the temperature distribution over the external surface of the tube.

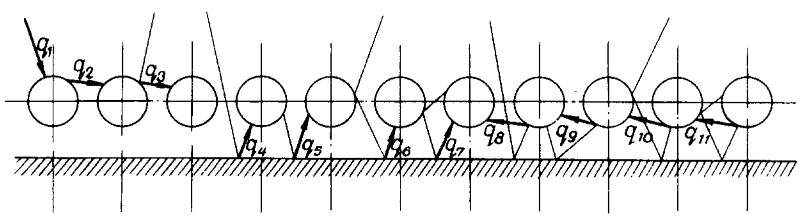

FIG. 1. Schematic drawing of heat fluxes for derivation of the temperature distribution function.

Summation of absorbed fluxes yields

$$\frac{aq_{\mathrm{inc}}}{2}[1+\cos(\theta+\psi)]+\frac{a^2\sigma_s}{2}\int\limits_{\theta_1'=\psi}^{\theta_2'=180-\gamma} T_r^4\cos\alpha\,.\,\mathrm{d}\alpha+\frac{a(1-a)q_{\mathrm{inc}}}{2\cdot 2}\int\limits_{\theta_1'=\psi}^{\theta_2'=180-\gamma}[1$$

$$+\cos(\theta+\psi)]\cos\alpha\,.\,\mathrm{d}\alpha+\frac{a}{2}\left[q_{\mathrm{inc}}(1-\varphi)+\frac{a\sigma_s}{2}\frac{D}{s}\int\limits_{\theta_1}^{\theta_2}T_r^4(1-\sin\beta_2)\,.\,\mathrm{d}\theta\right.$$

$$+\frac{(1-a)q_{\mathrm{inc}}}{2}\frac{D}{s}\int\limits_{\theta_1}^{\theta_2}[1+\cos(\theta+\psi)](1-\sin\beta_2)\,\mathrm{d}\theta+\frac{a(1-a)\sigma_s}{2}\frac{D}{s}\int\limits_{\theta_1}^{\theta_2}(1$$

$$\left.-\sin\beta_2)\int\limits_{\theta_1'=\psi}^{\theta_2'=180-\gamma}T_r^4\cos\alpha\,.\,\mathrm{d}\alpha\,.\,\mathrm{d}\theta\right][1+\cos(180-\theta+\gamma)]+\frac{a(1-a)}{2.2}\times\left[q_{\mathrm{inc}}(1-\varphi)\right.$$

$$+\frac{a\sigma_s}{2}\cdot\frac{D}{s}\int\limits_{\theta_1}^{\theta_2}T_r^4(1-\sin\beta_2)\,\mathrm{d}\theta+\frac{(1-a)q_{\mathrm{inc}}}{2}\frac{D}{s}\int\limits_{\theta_1}^{\theta_2}[1+\cos(\theta+\psi)](1-\sin\beta_2)\,\mathrm{d}\theta$$

$$\left.+\frac{a(1-a)\sigma_s}{2}\frac{D}{s}\int\limits_{\theta_1}^{\theta_2}(1-\sin\beta_2)\int\limits_{\theta_1'=\psi}^{\theta_2'=180-\gamma}T_r^4\cos\alpha\,.\,\mathrm{d}\alpha\,\mathrm{d}\theta\right]\int\limits_{\theta_1'}^{\theta_2}[1$$

$$+\cos(180-\theta+\gamma)]\cos\alpha\,.\,\mathrm{d}\alpha-a\sigma_sT_r^4=\frac{T_r-T_0}{R}.\quad(5)$$

The integro-differential equation (5) describes the peripheral temperature distribution in the range of the angle $0\leqslant\theta\leqslant 180$ (Fig. 2). The relationships between the angles in equation (5) and their ranges are as follows:

$$\cos(\theta+\psi)=1-\frac{2s}{D}\,.\sin\psi,\qquad(0\leqslant\psi\leqslant\theta_1)$$

$$\theta_1=\arcsin\frac{D}{s}$$

$$\theta_2=\pi$$

$$\cos(180-\theta+\gamma)=1-\frac{2s}{D}\sin\gamma,(0\leqslant\gamma\leqslant\theta_1)$$

$$\beta_2=\frac{\pi}{2}+\gamma-\theta,\qquad\left(-\frac{\pi}{2}\leqslant\beta_2\leqslant\frac{\pi}{2}\right)$$

$$\alpha=\frac{\pi}{2}-\theta-\mathrm{arctg}\frac{\cos\theta'-\cos\theta}{(2s/D)-\sin\theta'-\sin\theta},$$

$$\left(-\frac{\pi}{2}\leqslant\alpha\leqslant\frac{\pi}{2}\right)$$

Equation (5) was solved by a finite difference method. The half-section of the tube was divided into 60 equal sectors. The calculation was made for the following value determining T_r:

$$\theta=0\,(3°)\,180°,\qquad R=0{\cdot}00086;0{\cdot}0043;0{\cdot}0086;0{\cdot}0129;0{\cdot}0215\ \mathrm{m^2deg/W},$$

$$q_{\mathrm{inc}}=58{\cdot}1;87{\cdot}2;116{\cdot}3;174{\cdot}5;\ 232{\cdot}6\ \mathrm{kW/m^2},$$

$$T_0=373;773;973°\mathrm{K},$$

$$\frac{s}{D}=1{\cdot}1;1{\cdot}25;1{\cdot}75\text{ and }2{\cdot}5;\qquad a=0{\cdot}9.$$

The solution was obtained by a trial-and-error method on the electronic computers Ural-1 and Ural-2. The calculation was made with a fixed point to reduce machine time.

The algorithm of equation (5) at $1-a=0{\cdot}1$

is of the form:

$$
\begin{aligned}
&0{\cdot}9\bar{q}\bar{R}A_1 : 0{\cdot}1 + 0{\cdot}2308 \sum \bar{R}\bar{T}_r^4\bar{D}_i\, 10^5 + 0{\cdot}45\bar{q} \sum \bar{R}A_1\bar{D}_i\, 10 + A'[0{\cdot}9\bar{q}\bar{R}m : 0{\cdot}1 \\
&\quad + 0{\cdot}4616\frac{D}{s}\sum \bar{R}\bar{T}_r^4 B_i\,\Delta\theta : 0{\cdot}001 + 0{\cdot}18\bar{q}\frac{D}{s}\sum \bar{R}A_1 B_i\,\Delta\theta + 0{\cdot}513\frac{D}{s}\sum B_i \sum \bar{R}\bar{T}_r^4\bar{D}_i\,\Delta\theta\, 10^3] \\
&\quad + 0{\cdot}45\sum A'\bar{D}_i\Bigg[0{\cdot}9\bar{q}\bar{R}m : 0{\cdot}1 + 0{\cdot}4616\frac{D}{s}\sum \bar{R}\bar{T}_r^4 B_i\,\Delta\theta : 0{\cdot}001 + 0{\cdot}18\bar{q}\frac{D}{s}\sum \bar{R}A_1 B_i\,\Delta\theta \\
&\qquad + 0{\cdot}513\frac{D}{s}\sum B_i \sum \bar{R}\bar{T}_r^4\bar{D}_i\,\Delta\theta\, 10^3\Bigg] + T_0 = 0{\cdot}513\bar{T}_r^4\bar{R}\, 10^3 + \bar{T}_r. \qquad (6)
\end{aligned}
$$

Equation (6) may be written as

$$A = B\bar{T}_r^4 + T_r.$$

To reduce the number of iterations, the resulting value of T_i was directly used for specification of function (5) and evaluation of T_{i+1}. The resulting function of the temperature distribution was used with new values of parameters.

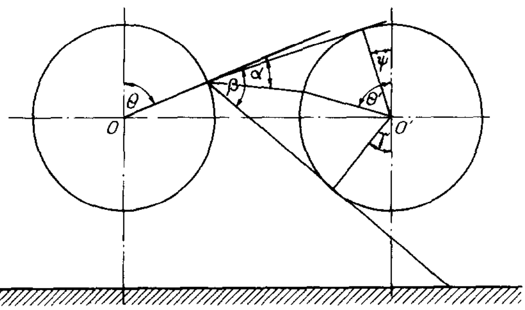

FIG. 2. Ratio between angles of screen with smooth tubes.

The mean temperature of external surface of the tube is given by

$$T_m = \tfrac{1}{60}\sum_{i=1}^{i=60} (T_r)_i. \qquad (8)$$

The heat flux absorbed by heated medium is given by

$$q_{\text{net}} = \frac{T_m - T_0}{R}\cdot\frac{\pi D}{s}. \qquad (9)$$

The values of T_m and q_{net} were also estimated from equations (8) and (9) on the electronic computer.

To verify the evaluated temperature distribution, it was used to find q'_{rev} on the electronic computer which was then compared with q_{rev} obtained from formula (1). The difference did not exceed 10 per cent, i.e. the mean error in T_r was below 2 per cent.

In Fig. 3, the usual graphs are presented showing the temperature distribution over the tube surface with different s/D and q_{inc} at $T_0 = 573^\circ$K and $R = 0{\cdot}0086$ m^{2}deg/W. In Fig. 4 similar diagrams are shown for different s/D at $R = 0{\cdot}0043$ m^{2}deg/W, $T_0 = 373^\circ$C and $q_{\text{inc}} = 58{\cdot}1$ kW/m^2.

The analysis of the graphs shows that the minimum temperature $T_r^{\min}$ is found at angles $\theta < 180^\circ$. In this case with increasing relative spacing, the temperature minimum moves in the direction of increasing θ. Simultaneously, with equal q_{inc}, the difference of T_r at $\theta = 180^\circ$ and $T_r^{\min}$ decreases. This type of temperature change is caused by the fact that in the derivation of equation (5) uniform radiation of the lining is assumed and peripheral flux is not taken into account.

The graphs and evaluations have shown that uniformity of temperature distribution is greater with increasing relative spacing, temperature of the heated medium, incident heat flux and decreasing thermal resistance of the soot layer on tubes.

Maximum value of the temperature $T_r^{\max}$ increases with incident heat flux, sooting resistance and the temperature of the heated medium and is independent of the relative spacing of the screen.

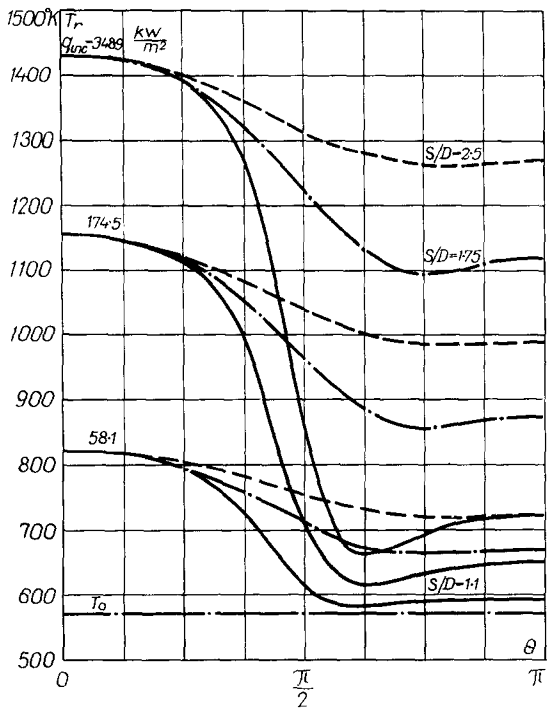

FIG. 3. Ordinary plots of temperature distribution over the external surface of screen tubes.

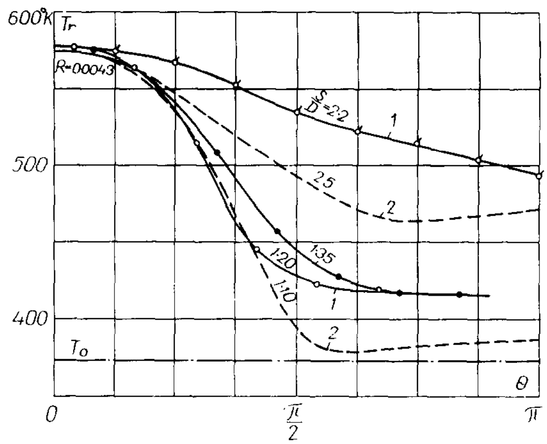

FIG. 4. Comparison of predicted and experimental temperature on the tube surface.

Experimental determination of temperature distribution was carried out on a unit simulating the operation of sooted tubes. The unit was a plane heater 180 × 180 × 100 mm. One of the walls with an internal electrical heater imbedded in chamotte was a heat-energy radiator, and the opposite wall acted as a lining. Between the walls was mounted a series of thick walled porcelain tubes (internal diameter, 20 mm; wall thickness, 5 mm). Porcelain tubes were used because it was necessary to provide considerable thermal resistances (R = 0·05 m^{2}deg/W) to make the operating conditions similar to those of sooted tubes. The porcelain tubes were cooled with water. The temperature of the external surface of the central tube was measured by a nichrome–constantan thermocouple with a thermal electrode diameter of 0·1 mm. The tube could be rotated round its axis and the temperature measured at different angles θ. The experiments were performed at $q_{\text{inc}} = 59$ kW/m^2 and s/D = 1·2; 1·35; 2·2.

In Fig. 4 (Graph 1) the peripheral temperature distribution on the experimental tube is compared (Graph 2) with the predicted temperature distribution by formula (5). Levelling of temperatures in the experimental tube is caused by non-uniform radiation of the lining and peripheral heat fluxes due to the small diameter, large thickness of the wall and rather high value of heat-conduction coefficient. In a layer of soot deposits peripheral heat fluxes are many times less.

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Abstract—An equation is presented which describes the temperature distribution over the external surface of a smooth tube of a single-row screen at different values of radiant flux from the furnace, thermal resistance of the soot layer, temperature of the heated medium and relative tube spacing. This equation also accounts for intrinsic radiation of the soot deposits and absorption of reflected heat fluxes, radiation of the lining being uniform.

The integro-differential equation is solved by the finite-difference method on an electronic computer.

Uniformity of the temperature distribution is shown to increase with the relative spacing, temperature of the heated medium, incident heat flux and with decreasing thermal resistance of the soot layer on the tubes.

The predicted values are compared with experimental data.

Résumé—Une équation est présentée, décrivant la distribution de température sur la surface extérieure d'un tube lisse situé dans une rangée de tubes, pour différentes valeurs du flux de rayonnement provenant du fourneau, de la résistance thermique de la couche de suie, de la température du milieu chauffé et de l'espacement relatif des tubes. Cette équation tient compte aussi du rayonnement propre des dépôts de suie et de l'absorption des flux de chaleur réfélchis, le rayonnement du revêtement étant uniforme.

L'équation intégro-différentielle est résolue par la méthode des différences finies sur un calculateur électronique.

On montre que l'uniformité de la distribution de température croît avec l'espacement relatif, la température du milieu chauffé, le flux de chaleur incident et lorsque la résistance thermique de la couche de suie sur les tubes décroit. Les valeurs calculées sont comparées avec les valeurs expérimentales.

Zusammenfassung—Eine Gleichung wird angegeben für die Temperaturverteilung über die äussere Oberfläche eines glatten Rohres in einem einreihigen Gitter bei verschiedenen Werten des Temperaturstrahlungsflusses vom Ofen, thermischen Widerstandes der Russ-schicht, Temperatur des beheizten Mediums und relativem Rohrabstand. Diese Gleichung berücksichtigt auch die Eigenstrahlung der Russablagerungen und die Absorption reflektierter Strahlung, bei gleichmässiger Strahlung der Ausmanuerung.

Die Integral-Differentialgleichung wird nach der Methode endlicher Differenzen auf einer elektronischen Rechenmaschine gelöst.

Es wird gezeigt, dass die Gleichmässigkeit der Temperaturverteilung zunimmt mit dem relativen Abstand, der Temperatur des beheizten Mediums, der auffallenden Strahlung und mit abnehmendem thermischen Widerstand der Russ-schicht auf den Rohren.

Die berechneten Werte werden mit experimentellen Ergebnissen verglichen.