DESIGN OF METAL MESH FIRE NOZZLES OF INFRARED RADIATION BURNERS

O. N. Bryukhanov, A. I. Pluzhnikov, S. M. Shlyakhov, and E. N. Bryukhanova

UDC 539.3

The design is based on the requirements of thermal stability for the mesh as a plate of reduced rigidity and strength of a panel according to the scheme of a U-shaped frame.

1. The working element of metal mesh fire nozzles of gaseous infrared radiation burners is a panel that is a densely perforated plate with holes of a given shape (see Fig. 1). Here xy is the coordinate plane located in the middle plane of the plate; L_1, L_2 are the dimensions of the working field of the panel ($L_2 \geq L_1$); $b, \lambda b$ are the dimensions of a rhombus-shaped hole; $\lambda \geq 1$ is the elongation parameter of the rhombus. Owing to the linearity of the perforation figures, the region occupied by the material consists of a set of crossed rectilinear rods rigidly fixed at the nodes.

We use the following notation: l, the length of the rod of a cell (measured along the middle line of the contour); h, δ, the dimensions of the cross section of the rod (δ corresponds to the thickness of the mesh); φ, the area ratio. From the geometry of the cell we obtain

$$h = \frac{b\lambda\,(1 - \sqrt{\varphi}\,)}{\sqrt{\varphi}\,\sqrt{1+\lambda^2}}, \tag{1}$$

$F_r = h\delta$ is the cross-sectional area of the rod, $\beta = \arctan\lambda$, $\gamma = \arctan(1/\lambda)$.

2. We assume that, while in use, the mesh is heated uniformly. Along the outer contour it is framed by the burner casing. Actually, the mesh is fixed nonrigidly in the burner casing. In calculations of the working (temperature) stresses that cause the panel to lose stability, we will neglect the deformation of the casing.

Owing to the thermal and geometric symmetries of the problem the nodes of the mesh are stationary. This implies that the elongations of the rods are equal to zero:

$$\Delta l = 0\,. \tag{2}$$

Condition (2) yields a formula for the stress P compressing a rod:

$$P = E\,(T)\,F_r\,\alpha_m\,(T - T_0)\,, \tag{3}$$

where $E(T)$ is Young's modulus of the mesh material at the working temperature T, α_m is the mean coefficient of linear expansion; T_0 is the initial temperature (the temperature at which the burner is mounted), whose value is neglected in what follows.

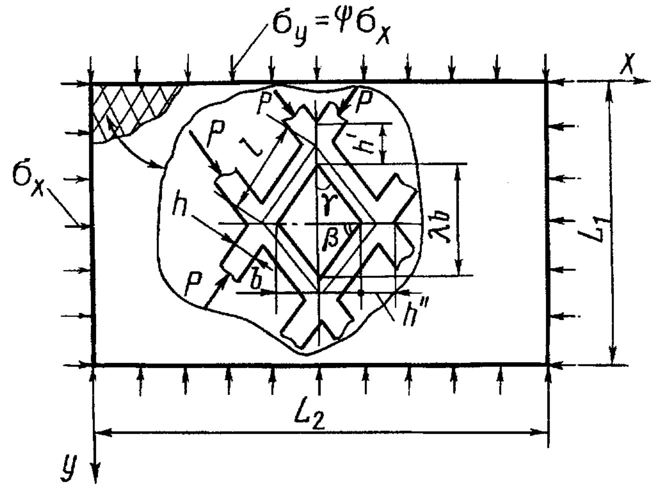
On the basis of Eq. (3) we find the pressure on the casing from the side of the mesh and conversely, on the mesh from the casing. The intensity of the latter is determined by means of the formulas

$$\sigma_x = \frac{2E\alpha_m\,Th\cos\beta}{\lambda b + h'}, \quad \sigma_y = \psi\,\sigma_x\,. \tag{4}$$

Here

Saratov Polytechnic Institute, Saratov. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 68, No. 2, pp. 253-257, March-April, 1995. Original article submitted March 18, 1992.

1062-0125/95/6802-0220$12.50



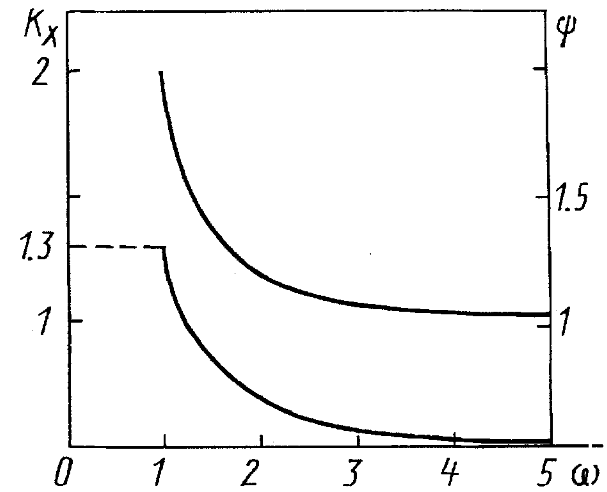


Fig. 1. Calculational scheme for the panel.

Fig. 2. Domain of values of the coefficient K_x.

$$\psi = \frac{\lambda b + h'}{b + h''} \frac{\cos \gamma}{\cos \beta}, \tag{5}$$

h', h'' are geometric parameters (Fig. 1).

From an analysis of Eq. (5) it follows that for actual gas burners with $0.5 \le h \le 1.5$ mm and $1 \le \lambda \le 1.5$ the values of ψ lie within the range

$$1 \le \psi \le 2. \tag{6}$$

3. To investigate the thermal stability of the mesh, we will determine its rigidity characteristics, performing their evaluation on the basis of the flexural strains of the mesh. The cylindrical rigidities per unit length of the plate are determined from the formulas [1]

$$D_x = \frac{E(T)\, h' \delta^3}{12(\lambda b + h')}, \quad D_y = \frac{E(T)\, h'' \delta^3}{12(b + h'')}.$$

Taking into account the actual variability of the flexural rigidities along the x, y axes and the compatibility of the flexural strains in two mutually perpendicular directions, it is worthwhile to turn to consideration of an isotropic plate with a certain reduced rigidity D. We will take the energy balance as the criterion for the equivalence between the rigidities of the orthotropic and isotropic plates. This yields

$$D = \frac{2 D_x D_y}{D_x + D_y}.$$

4. The solution of the problem of the thermal stability of the panel is reduced to integration of the differential equation describing the deflection of the plate under the loadings (4) uniformly distributed over the edges of the plate (Fig. 1) [2]:

$$\frac{D}{\delta} \nabla^4 W + \sigma_x \frac{\partial^2 W}{\partial x^2} + \sigma_y \frac{\partial^2 W}{\partial y^2} = 0, \tag{7}$$

where ∇^4 is the biharmonic operator; W is the bending deflection function.

In selecting boundary conditions, we proceed from the most unfavorable scheme of attaching the plate along the contour – a movable hinge. In this case, we obtain the least of the possible critical temperatures, and this will contribute to the stability margin.

We seek the solution of Eq. (7) in the form

$$W = A_{mn} \sin \frac{m\pi x}{L_2} \sin \frac{n\pi y}{L_1}.$$

Following [2], we obtain formulas for determining the critical stresses:

$$\sigma_{x_{cr}} = K_x \frac{\pi^2 D}{L_1^2 \delta}, \quad \sigma_{y_{cr}} = \psi \sigma_{x_{cr}}. \tag{8}$$

Here

$$K_x = \frac{\left[\left(\frac{m}{\omega}\right)^2 + n^2\right]^2}{\left(\frac{m}{\omega}\right)^2 + \psi n^2}, \quad \omega = \frac{L_2}{L_1}. \tag{9}$$

The coefficient K_x in Eq. (8) must assume the smallest values. In Fig. 2 its values are given as functions of ω for the range of ψ in (6).

Formulas (4) and (8) yield a relation for determining the critical temperature T_{cr}:

$$K_x \frac{\pi^2 \overline{D}}{L_1^2 \delta} = \frac{2\alpha_m T_{cr} h \cos\beta}{\lambda b + h'}, \tag{10}$$

where

$$\overline{D} = \frac{D}{E}; \quad \alpha_m T_{cr} = \int_0^{T_{cr}} \alpha(\tau)\, d\tau.$$

The condition for maintenance of the stability of the plate is

$$T \le T_{cr}/n_y, \tag{11}$$

where n_y is the adopted coefficient of the stability margin.

Assuming $\alpha(T) = \alpha_0 + \alpha_1 T$, on the basis of Eq. (11) we obtain from Eq. (10) a condition that should be satisfied by the length of the short side (width) of the gas burner panel to maintain its stability at the working temperature T:

$$L_1 \le L_1^*, \quad L_1^* = \sqrt{\left(\frac{K_x \pi^2 \overline{D} (\lambda b + h')}{2\delta h \cos\beta\, [\alpha_0 n_y T + 0.5\alpha_1 (n_y T)^2]}\right)}. \tag{12}$$

To ensure the stability of the panel, it is necessary to design burners with a width L_1 not exceeding the limiting dimension L_1^*, i.e., to make them as square as possible. The latter statement follows from an analysis of formula (9) and relation (10). From these relations it also follows [3] that with an increase in ω ($\omega \ge 5$) loss of stability of the panel is possible with formation of more than one half-wave along the long side. With a further increase in ω ($\omega \ge 10$) the length of the half-wave tends to approach the width L_1. Thus, the bulging panel is divided approximately into squares.

Consequently, in designing polygonal-type burners it is worthwile to divide the latter by rigidity ribs into "pads" approximating squares with a side $l_2 \lesssim L_1$ (Fig. 3). Moreover, the condition of the stability of the panel (12) should be fulfilled.

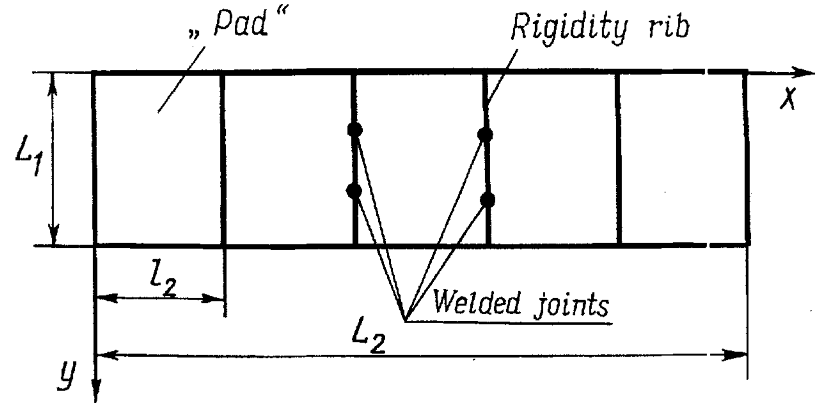


Fig. 3. Polygonal-type panel in plan.

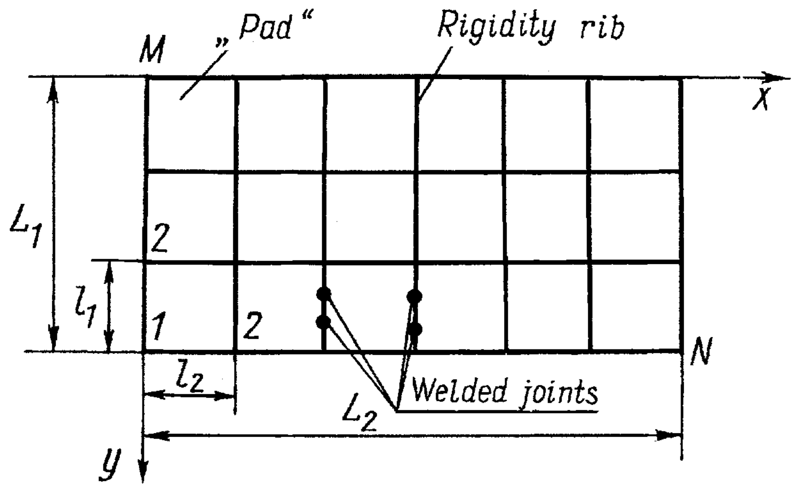


Fig. 4. Mosaic-type panel in plan.

If condition (12) is not fulfilled, the polygonal scheme must be rejected, and a panel of mosaic type should be designed with "pads" having the dimensions (Fig. 4) $l_1 = L_1/M$, $l_2 = L_2/N$, where N and M are respectively equal to the ratios $\overline{N} = L_2/L_1^*$, $\overline{M} = L_1/L_1^*$ rounded off to the larger side to an integer.

Technically, the formation of a "pad" is done by deep drawing (stamping). After the drawing of "pads" the latter constitute a boxlike U-shaped construction of height H. Structurally a "pad" consists of three meshes, with two of these being point-welded to each other [4]. The calculation of the parameters l_1, l_2 is made independently for each of the three mesh that make up the panel of the gas burner. Ultimately it is assumed that l_1 and l_2 are the smallest of the required dimensions and are identical for all three meshes (Figs. 3 and 4).

5. To estimate the temperature stresses in a "pad," we consider the latter to be a plane rectangular frame jammed at the bases of the racks. The frame is doubly statically indeterminate. Its parameters are: D, flexural rigidity; H, height of a rack (amount of drawing); l, length of a cross-bar (the dimension of the "pad" in plan, i.e., l_1 or l_2; in the case of a square pad $l_1 = l_2 = l$).

We will perform the expansion of the static indeterminacy by the method of forces [5]. We solve the problem with account for the thermal expansion of the casing, neglecting its elastic compliance.

The greatest normal stress in the frame due to the thermal effect occurs at the base of the racks and is determined by the formula

$$\sigma_{\max} = \frac{3E\delta l\,(H + l)\,(\alpha_m T - \alpha_c T_c)}{2H^2\,(H + 2l)}, \tag{13}$$

where α_c is the mean coefficient of linear expansion of the casing material; T_c is the heating temperature of the casing. If we neglect the thermal expansion of the casing, it should be assumed in formula (13) that $\alpha_c T_c = 0$.

The field of stresses in the "pad" due to the thermal effect is superimposed by the field of residual stresses resulting from the stamping. Since the operation of drawing is accompanied by annealing of the mesh, which relieves these residual stresses, the latter are not involved in strength calculations. It follows from the above that the maximum level of stresses in a "pad" is determined from formula (13). For a "pad" that is rectangular in plan we should assume in formula (13) that $l = \max\{l_1, l_2\}$.

The static condition of strength of the panel has the form

$$\sigma_{max} \leq [\sigma], \tag{14}$$

where $[\sigma]$ is the admissible stress for the working range of temperatures.

Apart from a checking calculation for strength using condition (14), it is possible to carry out design calculations consisting in the determination of the safe amount of drawing (the choice of the dimension H) from the equality $\sigma_{max} = [\sigma]$.

Thus, the procedure described above makes it possible to perform checking and design calculations of the panels of metal mesh fire nozzles of infrared radiation burners.

NOTATION

xy, coordinate plane; L_1, L_2, dimensions of the working field of the panel; b, λb, dimensions of the hole; λ, parameter of elongation of the rhombus; l, h, δ, linear dimensions of the rod of a cell; F_r, cross-sectional area of the rod of a cell; β, γ, angular parameters of a cell; φ, area ratio; P, stress; E, Young's modulus; α_m, α_c, mean coefficients of linear expansion; T, T_0, working and initial temperatures of the burner; σ_x, σ_y, intensity of stresses; h', h'', geometric parameters; D_x, D_y, D, rigidity; W, bending deflection function; A_{mn}, integration constants; ψ, K_x, coefficients; T_{cr}, critical temperature; n_y, coefficient of the stability margin; L_1^*, limiting dimension of the panel; l_1, l_2, dimensions of the "pad"; T_c, temperature of the casing; σ_{max}, maximum stress; $[\sigma]$, admissible stress.

REFERENCES

1. S. P. Timoshenko and S. Voinovskii-Kriger, Plates and Shells [in Russian], Moscow (1963).
2. A. S. Vol'mir, Stability of Deformable Systems [in Russian], Moscow (1967).
3. P. M. Ogibalov and V. F. Gribanov, Thermal Stability of Plates and Shells [in Russian], Moscow (1968).
4. O. N. Bryukhanov, E. V. Kreinin, and B. S. Mastryukov, Radiative Gas Heating [in Russian], Leningrad (1989).
5. S. D. Ponomaryov et al., Strength Calculations in Mechanical Engineering [in Russian], Vol. 1, Moscow (1956).