LITERATURE CITED

1. P. G. Daniels, "The flow about the trailing edge of a supersonic oscillating aerofoil," J. Fluid Mech., 72, No. 3 (1975).
2. S. N. Brown and P. G. Daniels, "On viscous flow about the trailing edge of a rapidly oscillating plate," J. Fluid Mech., 67, No. 4 (1975).
3. O. S. Ryzhov and E. D. Terent'ev, "Nonstationary boundary layer with self-induced pressure," Prikl. Mat. Mekh., 41, No. 6 (1977).
4. V. I. Zhuk and O. S. Ryzhov, "Solutions of dispersion equation using the theory of free boundary-layer interaction," Dokl. Akad. Nauk SSSR, 247, No. 5 (1979).
5. E. D. Terent'ev, "Nonstationary boundary layer with self-induced pressure near oscillating wall in supersonic flow," Dokl. Akad. Nauk SSSR, 240, No. 5 (1978).
6. A. I. Ruban, "Numerical solution of local asymptotic problem of unsteady separation of laminar boundary layer in supersonic flow," Zh. Vychisl. Mat. Mat. Fiz., No. 5 (1978).
7. A. V. Kazakov, "Flat plate with flap in unsteady supersonic viscous flow," Uch. Zap. Central Hydrodynamic Institute, 11, No. 4 (1980).
8. O. S. Ryzhov, "Unsteady three-dimensional boundary layer, freely interacting with external flow," Prikl. Mat. Mekh., 44, No. 6 (1980).
9. K. Stewartson and P. G. Williams, "Self-induced separation," Proc. R. Soc. London, Ser. A, 312 (1969).
10. V. Ya. Neiland, "Theory of laminar boundary-layer separation in supersonic flow," Izv. Akad. Nauk SSSR, Mekh. Zhidk. Gaza, No. 4 (1969).

TEMPERATURE DISTRIBUTION OVER THE SURFACES OF SPHERICAL SHELLS IN A PURGED DENSE LAYER WITH INTERNAL HEAT GENERATION

L. K. Vukovich, V. I. Lelekov,
A. V. Nikolaev, S. S. Timar',
and N. V. Tkach

Instruments are being developed at present in which a spherical filling is enclosed between perforated walls and its thickness amounts to three–six particle diameters. The specific conditions for the entry of gas into the filling (through holes in the perforated wall) and its relatively small thickness should have an effect on the nature of the gas motion in the filling, and consequently on heat exchange with spheres placed in various arrays. The distribution of the local characteristics of heat or mass exchange through the surface of spheres in the packings has been investigated in a series of experimental researches. The results obtained are presented in the form of a distribution of the local coefficients of heat exchange over the surface. However, a number of practical problems require knowledge of the local surface temperatures (for example, for the calculation of the thermal stresses in the casings enveloping a heat-generating sphere), which it is impossible to determine from the existing local heat transfer coefficients determined by detectors of the local thermal and mass fluxes, in connection with the interrelationship between the internal and external heat exchange problems [1]. An approximate computational dependence has been proposed in [2] for the determination of the maximum temperature nonuniformity in the casing enveloping a heat-generating core. This dependence has been derived for a single type of packing of the spheres. The absence in it of the heat-generation power remains incomprehensible. An expression for the relative maximum increase of the temperature differential in the casing caused by the different intensity of heat exchange at various points of its surface has been obtained in [3] by an alternate numerical solution of the time-independent thermal conductivity equation for a spherical heat-generating element under boundary conditions of the third kind (determined experimentally) in the range of variation 0.4–2.85 of the ratios of the thermal conductivity of the shell material and the coolant. However, this dependence has been derived for a specific packing of the spheres with a ratio of the channel and sphere diameters of less
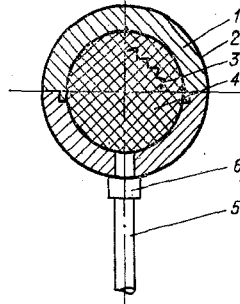
Fig. 1

than 1.9. A shortcoming of the dependences derived in [2, 3] is the absence in them of parameters of the external heat exchange. It seems important in connection with this to determine the local heat transfer coefficients through the surface of a sphere in packings of a different type with all the parameters of the internal and external heat exchange taken into account. This will permit using the data obtained to calculate the temperature fields on the surfaces of spheres in the packings, and consequently the thermal stresses in them.

The investigations were performed for three types of regular packings: (I) cubic (porosity $\varepsilon = 0.476$, number of contacts with neighboring spheres $n = 6$), (II) maximally dense ($\varepsilon = 0.259$, $n = 12$), and (III) packing with the number of contacts in the plane perpendicular to the direction of coolant flow equal to six and with two points of contact in the frontal and stern parts of the sphere with $\varepsilon = 0.396$.

The packings of spheres were assembled in boxes of rectangular and square cross sections (packings 1 and 2 − 100×100 mm, and packing 3 − 100×86.6 mm) out of spheres 50 mm in diameter, hemispheres, and quarters of spheres according to the idea of an elementary cell of spherical filling proposed in [4]. Five series of spheres were installed according to the direction of flow in the packings. Interchangeable walls with central holes 50, 40, 25, 15, and 10 mm in diameter at a zero distance from the spheres to the plane of the hole were placed in front of the first series and after the fifth one. The investigations were performed by the method of local simulation in the steady heat-exchange regime. The leakages of heat by thermal conductivity at points of contact with neighboring spheres were measured.

The spherical calorimeter (Fig. 1) consisted of a shell 1 5 mm thick assembled from two hemispheres joined by a tight fitting. Six Chromel−Copel thermocouples 2 with wire 0.2 mm in diameter sealed along the shell generatrix with an angular step of 30° are led out from within to the outer casing surface. The calorimeter casing is made out of Teflon-4 ($\lambda_m = 0.25$ W/(m·°K)), just as are the rest of the packing spheres. A spherical core 4 made out of fireproof clay with an electric spiral 3 uniformly wound onto its surface was placed into the casing. A thin insulation layer was deposited over the spiral. The uniform winding of the spiral provided for a constant heat flux density on the inner surface of the casing. A metal tube 6 mm in diameter through which the thermocouple and heater wires were led out served as the support 5. In order to reduce the heat leakage through the support, the latter is connected to the calorimeter through a heat-insulating collar 6. The support was led out of the box through a contact hemisphere, which permitted avoiding the effects of the support on the hydrodynamics in the filling. The calorimeter could be rotated about its axis with the help of the support and a coordinate mechanism, which provided for the determination of the temperature field on its entire surface.

The regulus of the differential Chromel−Copel thermocouple is sealed at the point of contact of the adjacent sphere or hemisphere with the calorimeter. The other regulus of this thermocouple was sealed at a distance of 5 mm from the surface of the sphere along its radius. The temperature differentials at the contact spheres were determined in the experiments, and from them − the heat flux from the measuring calorimeter to the contact spheres. It was arbitrarily assumed for determination of the heat leakages that the heat flux on the outer surface of the calorimeter is uniformly distributed. The specific heat flux on the outer surface of the calorimeter at the contact points with adjacent spheres was added to the heat flux in the contact sphere. The value of the temperature which would occur in the case of complete simulation was determined by multiplying the measured temperature at the contact point on the calorimeter surface by the ratio of the total heat flux to the specific flux through the calorimeter surface. Control experiments using two heated spheres placed into the packings have shown that the temperatures at the contact points differ for local simulation by no more than 5% from the temperatures measured with complete thermal simulation. Springs mounted between the channel walls and the contact hemispheres, as well as between the contact spheres and the ones next to them, have provided for an identical clamping force to the calorimeter.
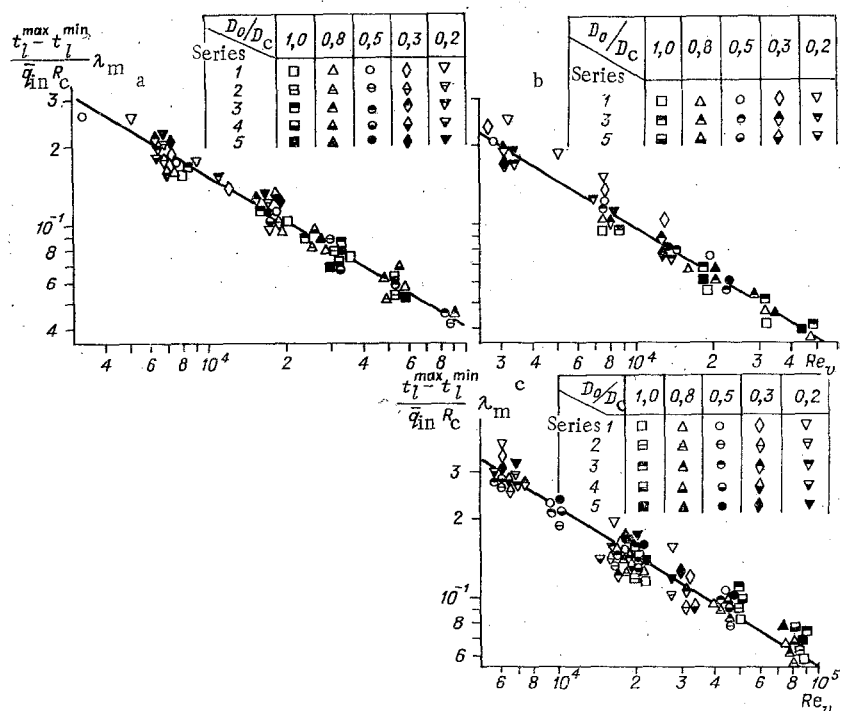
Fig. 2

The temperature fields on the calorimeter surfaces were determined in the experiments. The average surface temperature was calculated with account taken of the areas of the surface sections on which the thermocouples are mounted. The value of the average temperature of the calorimeter surface in the experiments was mainly 40–60°C; heat transfer of a sphere by radiation was inappreciable. Air was supplied to the hole through a damping chamber having a series of equalizing grids, which permitted obtaining a sufficiently uniform velocity field at the exit from the hole. The Reynolds number was calculated in three ways: for the spheres of the first packing series — from the average velocity at the exit from the hole and the calorimeter diameter, Re_c, for the subsequent series — from the average velocity of the flow coming into the packing in the cross section of a box unoccupied by spheres, $Re_{i.f.}$, and from the average velocity of the gas in the minimum straight-through packing cross section and the calorimeter diameter, Re_v. The range of variation of Re_c in the experiments was $8 \times 10^3 - 3 \times 10^5$, $Re_{i.f.} - 5 \times 10^2 - 10^4$, and $Re_v - 3 \times 10^3 - 10^5$, respectively.

In the investigated range of Reynolds numbers the nature of the temperature distribution over the calorimeter surface for each packing did not depend on the value of the criterion Re.

The nature of the distribution and size of the local temperatures on the calorimeter surface for the second, third, and fourth series in packings 1 and 3 did not depend on the quantity D_0/D_c but were functions of this ratio for the first and fifth packing series. The minimum values of the temperatures on the calorimeter surface in the first series of all the packings and in the third and fifth series of packing 2 were observed at its frontal point as well as in the most freely purged parts of the surface of the second, third, fourth, and fifth series in packings 1 and 3. The maximum values of the temperature on the calorimeter surface occurred in the first, second, third, and fourth series of all the packings at the contact points located in the intake region of the calorimeter and for the fifth series — in its front part.

The dependences of the maximum temperature nonuniformity on the calorimeter surface on the value of Re_v are presented in Fig. 2 for all series of packings 1 and 3 and for the first, third, and fifth series of packing 2. Analysis of these dependences shows that the value of the maximum temperature nonuniformity on the surface of a sphere of the first series can be larger or smaller than the maximum temperature nonuniformity for the succeeding series and depends on the ratio D_0/D_c. The maximum temperature nonuniformity of the calorimeter surface of the fifth series of packings is similar to the maximum nonuniformity on the surface of the preceding series and is practically independent of the ratio D_0/D_c.

The dependences of the maximum temperature nonuniformity on the calorimeter surface in the third series of each packing on the value of $Re_{i.f.}$ are presented in Fig. 3. Analysis of these dependences shows that the greater the porosity of the packing and the smaller the number of contacts of the calorimeter with ad-

TABLE 1

Analysis	Packing					
	1		2		3	
	A	m	A	m	A	m
with respect to Re_v	34,86	0,59	24,06	0,60	63,02	0,61
with respect to $Re_{i.f.}$	14,07	0,59	9,51	0,60	14,93	0,61

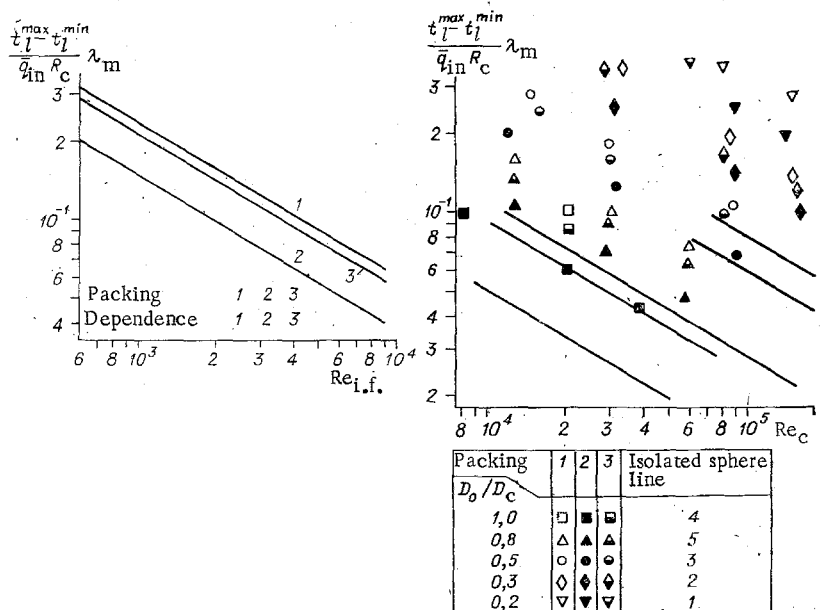| Packing D_0/D_c	1	2	3	Isolated sphere line
1,0	□	■	▣	4
0,8	△	▲	▲	5
0,5	○	●	◉	3
0,3	◊	◆	◈	2
0,2	▽	▼	▼	1

Fig. 3

jacent spheres, the higher is the maximum temperature nonuniformity on its surface. The experimental data on the maximum temperature nonuniformity in the packings are generalized by the equation $[(t_l^{max} - t_l^{min})]/ \bar{q}_{in} R_n]\lambda_m = A \ Re^{-m}$. The values of the quantities A and m for the third series of each packing upon analysis with respect to Re_v and $Re_{i.f.}$ are given in Table 1.

As follows from Table 1, the values of the exponents of Re for all the packings are similar to each other and approximately equal to 0.60. Experimental data on the three types of packings with the porosity taken into account are generalized by the dimensionless dependence

$$\frac{t_l^{max} - t_l^{min}}{\bar{q}_{in} R_L}\lambda_m = (2.85 + 26.6\varepsilon) Re_{i.f.}^{-0.6}.$$

The results previously obtained by the authors from experiments to determine the effect of the thermal conductivity of the casing material in the range $\lambda_m = 0.25\text{-}102$ W/(m.°K), the diameter ratio D_0/D_c in the range of 0.2-1.0, and the quantity Re_c in the range of $10^4\text{-}3\times 10^5$ upon axisymmetric jet flow around an isolated calorimeter in the initial section of the jet are generalized with a maximum error of 15% by the expression

$$\frac{t_l^{max} - t_l^{min}}{\bar{q}_{in} R_L}\lambda_m = 25.8 \left(\frac{\lambda_m}{\lambda_g}\right)^{0.66} \left[1 - 2.34\frac{D_0}{D_c} + 1.52\left(\frac{D_0}{D_c}\right)^2\right] Re_c^{-0.59}.$$

Analysis of the results shows an identity of the effect of the Reynolds number on the maximum temperature nonuniformity over the surface of an isolated calorimeter and a calorimeter placed in a packing. This confirms to some extent the "jet theory" of the motion of the gas in fillings of spheres and gives a basis for proposing that the effect of the thermal conductivity of the casing material on the maximum temperature nonuniformity over its surface will be the same in packings as for an isolated sphere in the case of jet streamline flow in the initial section of the jet. In this case the equation which describes the maximum temperature nonuniformity on the surfaces of casings of different thermal conductivities with internal sources of heat generation for a sphere located in the third series will be of the form

$$\frac{t_l^{max} - t_l^{min}}{\bar{q}_{in} R_L} \lambda_m = B \left(\frac{\lambda_m}{\lambda_g} \right)^{0,66} Re_{i.f.}^{-0,6}. \tag{1}$$

The values of the coefficient B are: for packing $I - B = 3.44$; $II - B = 2.16$, and $III - B = 2.97$. With the porosity taken into account expression (1) will take the form

$$\frac{t_l^{max} - t_l^{min}}{\bar{q}_{in} R_L} \lambda_m = (0.64 + 5.88\varepsilon) \left(\frac{\lambda_m}{\lambda_g} \right)^{0,66} Re_{i.f.}^{-0,6}.$$

The dependences of the maximum temperature nonuniformity on the calorimeter surface of the first series of different types of packings on the quantity Re_C are presented in Fig. 4. The dependences obtained for an isolated calorimeter in the initial section of the jet are shown by the lines. Analysis of the dependences shows that the maximum temperature nonuniformity on the calorimeter surface in the first series of packings increases as the value of the ratio D_0/D_C decreases and is practically always higher than for an isolated sphere with the identical values of D_0/D_C.

LITERATURE CITED

1. A. A. Sholokhov, I. P. Zasorin, V. E. Minashin, et al., Determination of the Temperature in the Fuel Elements of a Nuclear Reactor [in Russian], Atomizdat, Moscow (1978).
2. R. G. Bogoyavlenskii, Hydrodynamics and Heat Exchange in High-Temperature Nuclear Reactors with Spherical Fuel Elements [in Russian], Atomizdat, Moscow (1978).
3. N. G. Burdanov, L. E. Kostikov, and V. V. Lozovetskii, "Investigation of the local heat transfer coefficients and the temperature fields in spherical elements," Trudy MVTU, No. 307, Issledovanie Protsessov v Energeticheskikh Ustanovkakh, No. 4 (1979).
4. Z. R. Gorbis, Heat Exchange and Hydrodynamics of Dispersed Straight-Through Flows [in Russian], Énergiya, Moscow (1970).

MECHANISM OF LASER MAINTENANCE OF A
DEEP VAPOR CHANNEL WITHIN A LIQUID

A. A. Vedenov, G. G. Gladush,
and A. N. Yavokhin

UDC 535.214

The question of the limiting depth of penetration of a laser beam into a target material is of significant physical interest, as well as being of practical importance, for example, in increasing the efficiency of laser welding. It is known that with decrease in the velocity of beam motion its penetration into the target increases, and it can be assumed that the maximum depth is attained with a nonmoving beam. We note that in experiments with such a beam under certain conditions, a quite stable cavity has been observed, showing relatively small oscillations of its surface [1]. Therefore, it is of interest, as one stage in the investigation of laser welding, to study the model of a stationary vapor channel, formed in a liquid by a nonmoving laser beam. It is natural to commence with the simplest possible models of mechanical and thermal equilibrium of the cavity, not considering plasma phenomena, liquid hydrodynamics, light scattering, etc.

The condition for cavity stability is the equality of the pressures p_1 and p_2 (Fig. 1) at each point of the surface. In the present study it will be assumed that p_1 is composed of the hydrostatic pressure in the liquid, the external pressure p_a, and the pressure due to surface tension forces: $p_1 = d(h - z) + p_a + \sigma k$, while p_2 is composed of the pressure in the vapor p and the recoil pressure produced by mass transfer through the cavity surface. Here d is the specific gravity of the liquid, σ is the surface tension coefficient, k is the channel surface curvature, and h is its depth. In [2, 3] in a similar formulation of the problem it was assumed that $p = const = p_a$, which eliminated any effect of vapor flow dynamics in the cavity on cavity form.

In a deep and narrow channel, which is characteristic of laser welding techniques, it can be assumed that the gasdynamic quantities figuring in the problem are functions solely of distance from the cavity bottom z,