Journal of Engineering Physics and Thermophysics, Vol. 69, No. 4, 1996

COMPRESSOR-ADSORBER WITH A HEAT PIPE

V. A. Babenko, L. L. Vasil'ev, and
L. E. Kanonchik

UDC 621.577

A design and mathematical model of a compressor-adsorber employing a heat pipe are described. Based on numerical solution, the effect of the finning parameters on adsorber efficiency is analyzed.

The development of modern heat-engineering equipment is accompanied by increased heat generation and a simultaneous rise in the demands on environmental safety and rational use of primary energy. In view of this, of current interest is the creation of improved systems for producing heat and cold that are based on sorption processes. Adsorbers can be considered alternatives to heat pumps and refrigerators based on vapor-compression cycles and Freon, whose use is prohibited by the 1992 Montreal Protocol.

The operation of adsorption systems to produce cold and heat depends on many parameters. In order to analyze the laws governing heat and mass exchange and thermodynamic optimization, mathematical simulation of such complex systems, primarily of their key element, i.e., the compressor-adsorber, was carried out. Of particular interest for the study of adsorption processes are the numerous constant-temperature models. In [1], based on the assumption of a homogeneous temperature field, a diffusion process in a separate microporous particle was considered with allowance for adsorption heat. A more general case, taking into account diffusion heat and mass transfer in an adsorbent granule, was described in [2]. Nonisothermal diffusion and kinetics in a thin layer of a solid sorbent whose temperature was constant throughtout the volume but varied with time were studied in [3]. An integral method for the design of two adsorbers connected by a loop with a liquid was presented in [4-5], where thermal-wave motion was analyzed.

A review of recent publications [6-8] concerning the creation of adsorptive solar collectors has shown that the so-called constant-pressure model is probably best for describing modern sorbents with high porosity. Usually, such highly porous layers are obtained from separate sorbent granules, as a result of which they have a complex bimodal porous structure in which the spaces between the sorbent particles form large pores and small pores are located inside the particles. In [6], a two-dimensional model of heat and mass exchange in an adsorber at a constant pressure is proposed. The model ignores diffusive resistance, while thermal resistance is taken into account by the coefficients of heat exchange between the adsorbent layer and fin. On the basis of this model a thermodynamic and economic optimization of a finned adsorber for a solar collector was performed in [7].

The authors of [8] modified the constant-pressure model to describe the rapidly occurring adsorption processes that are common in refrigerators. They added to the heat-conductivity and mass-balance equations a kinetics equation that contained a first derivative and thus took into account the influence of the nonequilibrium state of sorption processes on the fields of temperatures and concentrations.

A more general statement of the problem is given in [9]. The basic system included the continuity equation, momentum-transfer equation, mass-diffusion equation, heat-transfer equation, and state equation for an elementary volume of sorbent. Numerical investigations were conducted for the influence of permeability and porosity on the distributions of temperature and pressure in the sorbent. It is shown that for specimens with $r/l \ll 1$ the problem becomes one-dimensional, and the constant-pressure model is a particular case applicable to media with high permeability (for example, $>10^{-9}$ m^2 for water filtration and $>10^{-12}$ m^2) and porosity ($>70\%$).

In the present work we suggest a new design of a heat-pipe-based compressor-adsorber, carry out its mathematical simulation, and verify its reliability experimentally.

Academic Scientific Complex "A. V. Luikov Heat and Mass Transfer Institute of the Academy of Sciences of Belarus," Minsk, Belarus. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 69, No. 4, pp. 531-539, July-August, 1996. Original article submitted November 3, 1994.

In the adsorber we applied a unique method for supplying and removing heat to the sorbent by using a heat pipe (HP) on which both an electric heater and a liquid heat exchanger were simultaneously mounted. The copper pipe (of length 650 mm, outer diameter 40 mm, and inner diameter 38 mm) with lengthwise grooves was filled with water. Adsorbers that employ heat-conducting devices with phase transitions have considerably extended potential (from domestic to space applications). Compared to single-phase systems of heat transfer, HPs are distinguished for their low thermal resistance, isothermicity, high heat-transmitting capability, and reliability.

The adsorber is made of steel in the form of a coaxial cylindrical container with an inner diameter of 56 mm and a length of 350 mm, into which a portion of the HP is inserted with a 6 mm-thick and 300 mm-long layer of sorbent deposited on it. The sorbent consists of a mixture of 80% zeolite (MgA) and 20% china clay; it is made by compression at a pressure of several tens of mPa with subsequent heating in a furnace for 3 h. Powdered zeolite has a low density (0.52 kg/m^3) and thermal conductivity (0.05–1 W$\cdot$m$^{-1}\cdot$K^{-1}). The density of the compound obtained was equal to 0.75 kg/m^3; the mean macropore diameter was 7000 Å; the total pore volume, which was measured using mercury, was $0.28 \cdot 10^{-3}$ m^3/kg; the permeability was of the order of 10^{-9} m^2. The mass of the sorbent, which was fastened by a metal grid, amounted to 200 g. Water was used as the heat-transfer agent.

Finning of the outside of the HP housing enhances heat transfer to the sorbent and increases the effective thermal conductivity of the sorbent layer. We analyzed the heat transfer surface formed by round copper fins with the following parameters: thickness $2\delta_f = 0.5$ mm; height $H_f = 6$ mm, spacing $2S_f = 12$ mm, inner diameter 40 mm, outer diameter 52 mm, and length of finned portion 0.3 m.

Mathematical Model. For mathematical simulation of the HP-based adsorber, we use a computational element which is half of a cell bounded by the axis of the heat pipe ($r = 0$) and the symmetry planes passing through the middle of the fin ($z = 0$) and the middle of the sorbent layer between two adjacent fins ($z = S_f$).

The two-dimensional mathematical model for adsorber design is based on the statement given in [6, 7]. However, unlike in [6, 7], the specimen was heated and cooled from the inside of the cylindrical layer from the heat pipe with a high internal heat transfer coefficient of the order of 10^4–10^5 W/m$^2\cdot$K. This coefficient can be considered infinite compared to the contact heat-transfer coefficient on the heat pipe surface α_{HP}. In calculations we assumed that $\alpha_{HP} = 500–1000$ W/m$^2\cdot$K.

The following difference is associated with the nonadiabaticity of the nonworking surfaces of the adsorber. A laboratory experiment demonstrates that because of the liberation or absorption of the adsorbate vapors on the sorbent surface it is not easy to attain complete heat insulation of the adsorber. Therefore, just as in the previous calculations by a one-dimensional model [10], we investigated the effect of insufficient heat insulation of the adsorber surface resulting in a nonzero coefficient of heat-exchange α_{env} with the environment.

In the case of heat supply and removal by a two-phase heat carrier with temperature T_{HP} (that of the heat pipe), the mathematical model of the heat and mass exchange in a fixed layer of solid sorbent in a finned adsorber is based on the following assumptions [6, 7]:

1) the pressure in the adsorber is uniform throughout the entire volume;

2) the resistance to diffusion is insignificant;

3) there is thermodynamic equilibrium at each time instant and at each point of the adsorber. The sorbent particles act similarly to heat sources distributed in the adsorber with volumetric density

$$Q = q_{st}\rho_s \frac{\partial a}{\partial t}. \tag{1}$$

A thermodynamic cycle consisting of two isobars and two isosteres is simulated assuming that the vapor temperature in the HP in the first two stages coincides with the temperature of the external source and in the subsequent two stages coincides with the adsorption temperature:

$$\begin{aligned} T_{HP} &= T_d \quad \text{when} \quad \tau_0 < \tau \le \tau_2, \\ T_{HP} &= T_a \quad \text{when} \quad \tau_2 < \tau \le \tau_4. \end{aligned} \tag{2}$$

From a mathematical standpoint, the problem is described by a two-dimensional parabolic partial differential equation that represents the energy conservation law

$$\rho_s (C_{ps} + aC_{pa}) \frac{\partial T_s}{\partial \tau} = \frac{1}{\partial r} \frac{\partial}{\partial r} \left(r\lambda_{\mathrm{eff}} \frac{\partial T_s}{\partial r} \right) + \frac{\partial}{\partial z} \left(\lambda_{\mathrm{eff}} \frac{\partial T}{\partial z} \right) + Q . \tag{3}$$

with initial conditions

$$T(r, z, \tau = 0) = T_0 \tag{4}$$

and boundary conditions

$$\left(\lambda_{\mathrm{eff}} \frac{\partial T}{\partial r} \right) \Bigg|_{r=r_0} = \alpha_{\mathrm{HP}} (T_{\mathrm{HP}} - T|_{r=r_0}) , \tag{5}$$

where α_{HP} is the coefficient of heat exchange between the inner surface of the cylindrical sorbent layer and a two-phase heat carrier with temperature T_{HP}:

$$- \left(\lambda_{\mathrm{eff}} \frac{\partial T}{\partial r} \right) \Bigg|_{r=r_1} = \alpha_{\mathrm{env}} (T|_{r=r_1} - T_{\mathrm{env}}) , \tag{6}$$

where α_{env} is the coefficient of heat losses by the adsorber on the outer surface:

$$\frac{\partial T}{\partial n} \Bigg|_{z=0} = 0 ; \quad \frac{\partial T}{\partial n} \Bigg|_{z=S_f} = 0 . \tag{7}$$

For points located on the fin and the wall of the HP, Eq. (3) takes the form

$$\rho_m C_m \frac{\partial T}{\partial \tau} = \lambda_m \Delta T .$$

The heat conduction Eq. (3) is supplemented with a mass-balance equation written in the form of the Dubinin–Radushkevich state equation [11]:

$$da = n \left(\frac{RT}{E} \right)^n a \ln \left(\frac{P_s(T)}{P} \right)^{n-1} \left(d \ln P - \frac{q_{st}}{RT^2} dT \right) , \tag{8}$$

and with the saturation curve for a coolant

$$P = P_s(T) , \tag{9}$$

where the initial conditions are

$$P(r, z, \tau = 0) = P_0 . \tag{10}$$

In the isosteric operating stages of the adsorber, the boundary conditions for Eq. (8) have the form

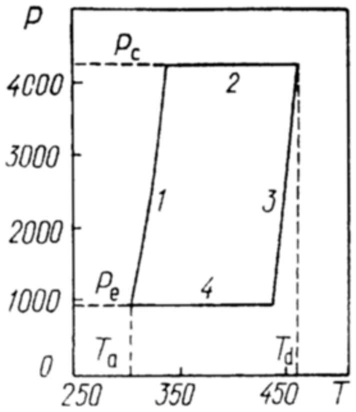
$$\frac{d}{d\tau} \int \int a(T(r, z), P) \, drdz = 0 , \tag{11}$$

which implies constancy of the total mass of the adsorber. For isobaric stages of desorption and adsorption, the pressure in the sorbent layer is determined by the constant pressure of condensation and evaporation:

$$P = P_c \quad \text{or} \quad P = P_e . \tag{12}$$

A numerical solution was obtained by using an implicit multidimensional difference advancing scheme constructed by an integrointerpolation method [12]. We solved the difference equations by reducing them to a system of three-point vector equations that was solved by a method of matrix fitting. In order to accelerate calculation, we used an algorithm for an automatic increase in the time step.

Results of Numerical Investigation. Calculations for a finned adsorber were carried out for a full thermodynamic cycle of its operation that included the following successive stages: I) isosteric heating, II) isobaric



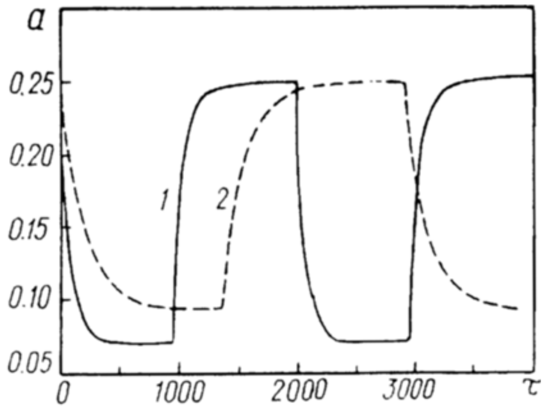


Fig. 1. Dependence of pressure in the adsorber on the mean temperature of the sorbent during an operating cycle (the coefficient of heat loss by the adsorber is $\alpha_{env} = 20\ \mathrm{W/m^2 \cdot K}$; the parameters of the fin are: $H_f = 0.006$ m, $2S_f = 0.0012$ m, $2\delta_f = 0.0005$ m). P, Pa; T, K.

Fig. 2. Change in the mean value of equilibrium adsorption with time for finned (1) and unfinned (2) adsorber ($\alpha_{env} = 20\ \mathrm{W/m^2 \cdot K}$; the parameters of the fin are: $H_f = 0.006$ m, $2S_f = 0.0012$ m, $2\delta_f = 0.0005$ m). a, kg/kg; τ, sec.

heating, III) isosteric cooling, and IV) isobaric cooling. The calculation of the isosteric stages was stopped when the pressure in the adsorber equaled that in the evaporator (stage II) or in the condenser (stage IV). The criterion for the termination of the calculation of the isobaric stages was a small change in transition to the next temporal temperature layer on the outer cylindrical surface of the sorbent ($\Delta T/T < 0.001$). A graph of the change in adsorber pressure versus the mean temperature of the sorbent layer in an operating cycle is given in Fig. 1a.

The calculation corresponds to the following initial data: the heat-pipe temperature in the case of desorption $T_d = 463$ K; the heat-pipe temperature in the case of adsorption $T_a = 303$ K; the ambient initial and condensation temperatures $T_c = 303$ K; the evaporation temperature $T_e = 278$ K; the heat exchange coefficient on the external boundary 20 $\mathrm{W/m^2 \cdot K}$, $\alpha_{env} = 0$; the heat exchange coefficient on the heat pipe (internal boundary) $\alpha_{HP} = 1000$ $\mathrm{W/m^2 \cdot K}$; the heat capacity of the zeolite $C_s = 1000$ J/kg; the density of the zeolite $\rho_s = 1000\ \mathrm{kg/m^3}$; the effective thermal conductivity $\lambda_{eff} = 0.4\ \mathrm{W/m \cdot K}$; the heat pipe radius $r_0 = 0.02$ m; the radius of the external sorbent layer $r_1 = 0.026$ m; the parameters of the copper fin are: the fin thickness $2\delta_f = 0.5$ mm; height $H_f = 6$ mm, and spacing $2S_f = 12$ mm.

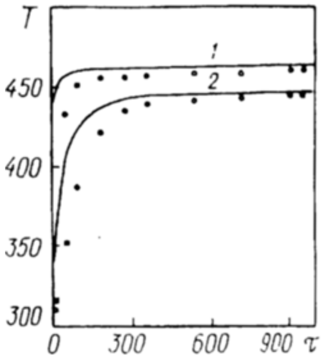
To determine the magnitude of adsorption, we used the Dubinin–Radushkevich equation in the form:

$$a = a_0 \exp\left[-\left(\frac{RT}{E}\ln\frac{P_s}{P}\right)^n - B(T)\right],$$

where the empirical coefficients $B(T) = \ln \rho_0/\rho_b (T - T_0)/(T_{cr} - T_b)$, $a_0 = 0.263$, $E = 18.5 \cdot 10^3$ J/mole, $T_0 = 293$ K, $T_{cr} = 547.15$ K, $T_b = 373.15$ K, $n = 2$ were found by using the method of least squares to process the experimental data for the zeolite (MgA)–water pair presented in [13] and obtained at the Laboratory of Porous Media of the Academic Scientific Complex "A. V. Luikov Heat and Mass Transfer Institute on the Academy of Sciences of Belarus."

Under these conditions the layer-average value of adsorption (Fig. 2) varies from 0.25 to 0.007 kg/kg. It is evident the duration of the isosteric periods is small in comparison with the duration of the isobaric periods. The use of fins makes it possible to increase the sorption potential for the water-zeolite pair and, consequently, to shorten the cycle duration by almost half compared to an unfinned adsorber.

Figure 3 presents a comparison of experimental data of [10, 14] for the desorption stage with calculations by a two-dimensional equilibrium model under conditions of a nonzero coefficient of heat exchange with the environment ($\alpha_{env} = 20\ \mathrm{W/m^2 \cdot K}$). The figure shows the change with time of temperatures at two characteristic



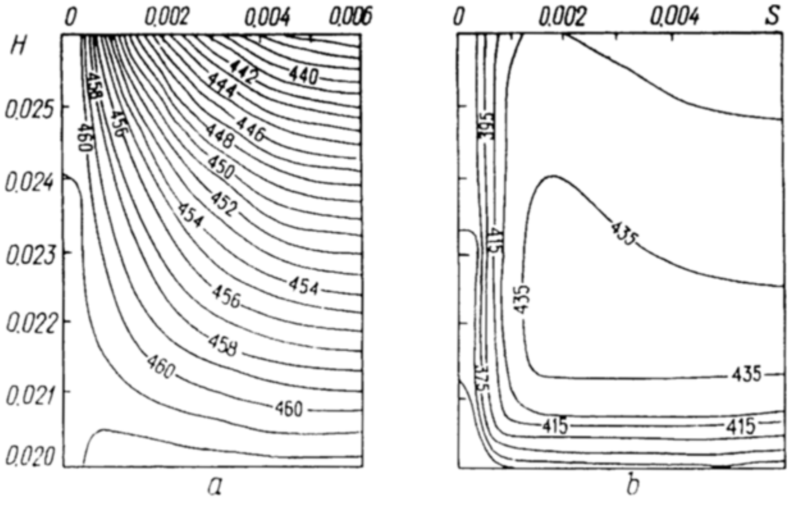


Fig. 3. Experimental (points) and calculated (curves) values of temperature on inner (1) and outer (2) surfaces of sorbent as functions of time in the desorption stage ($\alpha_{env} = 20$ W/m$^2\cdot$K).

Fig. 4. Two-dimensional temperature field in sorbent layer: a) at the end of the isobaric heating stage (the interval between isolines is 5 K), b) at the end of the isosteric cooling stage (the interval between isolines is 10 K). S, H, m.

points located near the inner ($r^* = (r - r_0)/(r_1 - r_0) = 0.01$) and outer surfaces of the sorbent layer ($r^* = 0.99$) at a half of its length. Note the coincidence of the experimental and calculated values of temperature in a stationary state. However, in an experiment the time required to attain a stationary state is higher. This is explained, on the one hand, by the fact that the nonequilibrium character of the sorption-desorption processes is ignored and, on the other hand, probably by the fact that the change of isosteric and isobaric heating periods was not precisely registered.

Figure 4 illustrates two-dimensional distributions of temperature fields in the calculated element at the end of the stages of isobaric heating and isosteric cooling. Due to the high thermal conductivity of the fins and HP, their presence (in the graphs, the fin is located on the left and the HP at the bottom) in the adsorber makes it possible to smooth the temperature field in the axial direction and decrease the radial temperature gradient over the sorbent layer, for example, at the end of the desorption stage it decreases from 39 to 17 K. The heating effect of the fins is greatest at distances of less than 2 mm.

The calculation of a full operating cycle makes it possible to determine such generalizing functions of processes in the adsorber as the specific cooling power

$$Q_m = Q_c/M ,$$

where M is the sorbent mass, Q_c is the heat flux removed from the object being cooled, and the coefficient of the conversion of the cooling power is

$$\mathrm{COP} = Q_c/Q_H ,$$

where Q_H is the heat flux supplied to the sorbent of the HP during an operating cycle.

We calculated the heat fluxes Q_c and Q_H in the following manner:

$$Q_c = (a_a - a_d)\,(L - C_{pa}\,(T_c - T_e))\,\rho_s\,\pi\,(r_1^2 - r_0^2)\,l_s/\tau_a ,$$

where a_a is the mean value of adsorption at the end of the isobaric cooling stage; a_d is the value of adsorption at the beginning of the stage of isosteric cooling; L is the latent evaporation heat; C_{pa} is the mean heat capacity of the heat carrier in the range of temperatures (T_c, T_e);

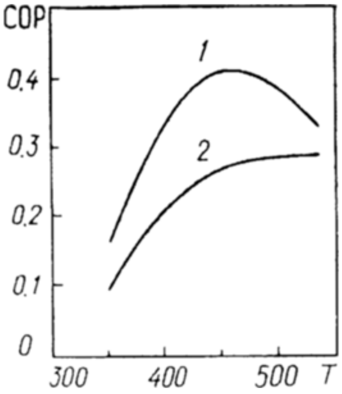


Fig. 5. Dependence of the coefficient of performance COP on the desorption temperature for a coefficient of heat loss by the adsorber $\alpha_{env} = 20$ W/m$^2 \cdot$K; 1) finned adsorber (the parameters of the fin are: $H_f = 0.006$ m, $2S_f = 0.0012$ m, $2\delta_f = 0.0005$ m), 2) unfinned adsorber, COP; T, K.

$$Q_H = \sum_{i=1}^{i=N} \alpha_{HP} (T_{HP} - T_{si}) \, 2\pi r_0 \Delta z_i ,$$

where T_{si} is the temperature of an elementary sorbent volume that contacts the heating surface, and index i is the stage number.

Using the calculated data, we analyzed the change in the COP for the water–zeolite pair with the desorption temperature for finned and unfinned adsorbers (Fig. 5). As follows from the calculations performed, fins in the sorbent layer increase the adsorber efficiency by about a factor of 1.5 over the entire desorption-temperature range at $\alpha_{env} = 20$ W/m$^2 \cdot$K.

In order to determine the influence exerted on the efficiency of the adsorber by the parameters of fins whose height coincides with that of the cylindrical sorbent layer, we consider the results of calculations carried out for adiabatic conditions on the outer surface of the adsorber (Fig. 6). An increase in the half-thickness δ_f of a copper fin within the interval from 0.1 mm to 1 mm (Fig. 6a) leads to an insignificant decrease in the COP of from 0.46 mm to 0.41 mm. This agrees with the data of [7]; in this work it is recommended that copper fins be of the minimum technologically possible thickness in order not to decrease the COP. For $\delta_f < 0.4$ mm, the specific cooling power decreases sharply below 380 W/kg. The calculations performed as well as technological considerations allow us to recommend copper fins with a half-thickness of from 0.1 mm to 0.4 mm.

Figure 6b illustrates the effect of fin spacing on the characteristics of a cell of the cylindrical adsorber. On the one hand, as the number of fins grows, their heating effect increases, and simultaneously the dimensions of the adsorber cell decrease. On the other hand, in this case the thermal resistance of the sorbent layer on the whole decreases, thus causing an increase in the heat flux supplied to the sorbent. As a result, the specific cooling power increases, and the COP for a given combination of fin height and spacing decreases. As is evident from Fig. 6b, it is not expedient to use a fin spacing larger than 10 mm, since this does not give a considerable gain in the COP.

Graphs of the COP and Q/M versus fin height are given in Fig. 6c. For fins higher than 20 mm, the COPs asymptotically approach 0.5, and the specific cooling power decreases from 200 W/kg to 100 W/kg. The choice of fin height is determined by design considerations and by the magnitude of the heat flux to be removed.

Here, it is important to note that fins are a necessary element in adsorber design, since they are required to increase the layer thickness and, consequently, the cold efficiency without a noticeable decrease in the efficiency characteristics.

Thus, the proposed model of heat and mass exchange processes in an HP-based finned adsorber made it possible to calculate the two-dimensional temperature and concentration fields in an operating cycle and on this basis to investigate the efficiency characteristics of the adsorber. The data obtained in the work can be useful for improving the specifications of such devices.

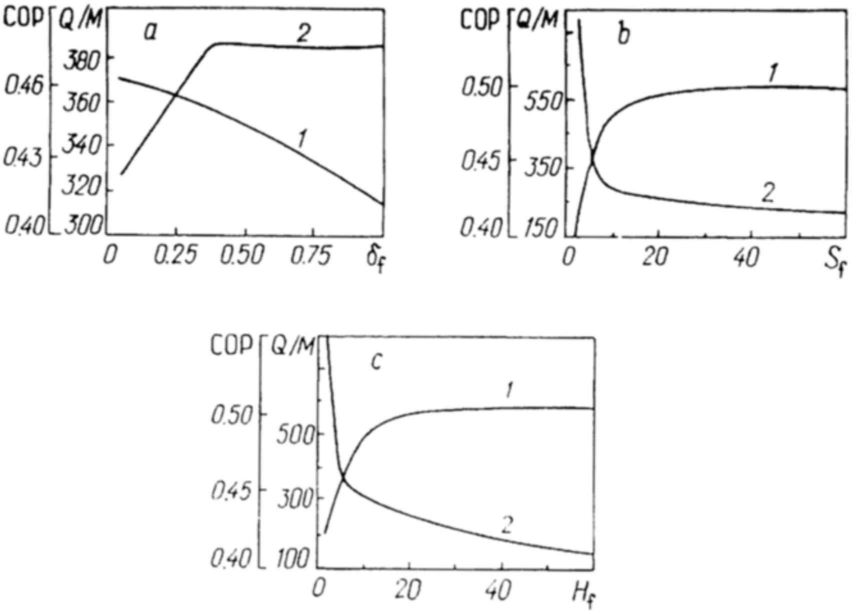


Fig. 6. Dependence of the coefficient of performance COP (1) and specific cooling power Q/M (2) on the parameters of copper fins: a) thickness, b) spacing, c) fin height; COP. Q/M, W/kg; δ_f, S_f, H, m.

NOTATION

α, heat exchange coefficient; λ, thermal conductivity coefficient; $2S_f$, fin spacing; H_f, fin height; $2\delta_f$, fin thickness; q_{st}, latent adsorption heat; C, heat capacity; ρ, density; T, temperature; τ, time; Q, heat flux; r, radius; P, pressure; R, universal gas constant. Subscripts and superscripts: HP, heat pipe; s, sorbent, a, adsorbate, adsorption; d, desorption; eff, effective; c, condenser; e, evaporator; m, metal; f, fin.

REFERENCES

1. L. P. Lee and D. M. Ruthven, J. Chem. Soc. Faraday Trans., No. 239, 2406 (1987).
2. L. M. Sun, F. Meunier, and B. Mischler, Int. J. Heat Mass Transfer, **29**, 1393-1406 (1986).
3. D. M. Ruthven and L. K. Lee, AIChE J., **27**, 654 (1981).
4. D. J. Miles, S. V. Shelton, and W. J. Wepper, Heat Recovery Systems, **9**, No. 3, 233-247 (1989).
5. J. M. Chinch and D. I. Tcherner, International Solar Ener. Conference ASME (1989).
6. J. J. Guilleminot, F. Meunier, and J. Pakleza, Int. J. Heat Mass Transfer, **30**, No. 8, 1595-1605 (1987).
7. J. J. Guilleminot and F. Meunier, Solar Energy, No. 7, 11-25 (1989).
8. E. F. Passos, J. F. Escobedo, and F. Meunier, Solar Energy, **42**, No. 2, 103-111 (1989).
9. N. B. Amar, L. M. Sun, and F. Meunier, in: Proc. Symposium Solid Sorption Refrigeration. Paris, 18-20 November, (1992), pp. 64-69.
10. L. L. Vasiliev, L. E. Kanonchik, A. A. Antux, A. G. Kulakov, and I. Rosin, in: Proc. 24th Int. Conf. on Environmental Systems. Friedrichshafen, Germany, June 20-23, No. 941580 (1994).
11. M. M. Dubinin and V. A. Asthakov, Adv. Chem. Series, **102**, 69 (1971).
12. L. F. Shampine and C. W. Gear, SIAM Review, **21**, 1-17 (1986).
13. P. K. Maier–Laxhuber, Sorptionswarmepumpen und sorptionsspeicher mit dem stoffpaar zeolith-H_2D. Diss. Dokt. Naturwiss. Fak. Phys. Techn. Univ. Munchen (1983).
14. L. L. Vasil'ev, L. E. Kanonchik, A. G. Kulakov, and A. A. Antukh, in: Heat and Mass Transfer-93/94, Inst. Heat and Mass Transf. Acad. Sci. Belarus [in Russian], Minsk (1994), pp. 29-32.