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Heat Exchangers for Utilization of the Heat of High-Temperature Geothermal Brines

A. B. Alkhasov* and D. A. Alkhasova

Institute for Geothermal Research, Dagestan Science Center (DNTs), Russian Academy of Sciences, Makhachkala, 367030 Dagestan, Russia

**e-mail: alibek_alhasov@mail.ru*

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Abstract—The basic component of two-circuit geothermal systems is the heat exchanger. When used in geothermal power systems, conventional shell-and-tube and plate heat exchangers cause problems related to the cleaning of the latter from salt-deposition and corrosion products. Their lifetime does not exceed, as a rule, 1 year. To utilize the heat of high-temperature geothermal brines, a heat exchanger of the "tube-in-tube" type is proposed. A heat exchanger of this design has been operated for several years in Ternair geothermal steam field; in this heat exchanger, the thermal potential of the saline thermal water is transferred to the fresh water of the secondary circuit of the heating system for apartment houses. The reduction in the weight and size characteristics of the heat exchangers is a topical problem that can be solved with the help of heat transfer enhancers. To enhance the heat transfer process in the heat exchanger, longitudinal ribbing of the heat exchange surface is proposed. The increase in the heat exchange surface from the heat carrier side by ribbing results in an increase in the amount of the heat transferred from the heating agent. The heat exchanger is easy to manufacture and is assembled out of components comprised of two concentrically positioned tubes of a definite length, 3–6 m, serially connected with each other. The method for calculation of the impact of the number and the size of the longitudinal ribs on the heat transfer in the well heat exchanger is presented and a criterion for the selection of the optimal number and design parameters of the ribs is formulated. To prevent the corrosion and salt deposition in the heat exchanger, the use of an effective OEDFK (oxyethylidenediphosphonic acid) agent is proposed. This agent has a long-lasting corrosion-inhibiting and antiscaling effect, which is explained by the formation of a strongly adhesive chelate layer difficult to wash off the surface. The passivating OEDFK layer is restored by periodical pulsed introduction of the agent solution into the brine at the heat exchanger inlet.

Keywords: geothermal resources, high-temperature brines, heat exchanger, heat transfer, heat transfer enhancement, salt deposition

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The large-scale exploitation of geothermal energy is associated primarily with high-temperature geothermal brines that constitute a considerable portion of the existing energy sources [1]. However, a high content of salts and gases dissolved in the brines determines their high corrosive activity and inclination to salt deposition, as a result of which one of the essential tasks of using the waters of this kind is the development of methods for corrosion and salt deposition control. To prevent the above adverse processes in the power-generating equipment and service lines, two-circuit systems are used; in the first circuit, saline thermal water circulates, while fresh water or a low-boiling working medium circulate in the second circuit.

The basic component of the two-circuit system is the heat exchanger manufactured of alloyed steel grades in very short supply and titanium, which impairs the technical and economic characteristics of the geothermal heat supply systems. Therefore, the necessity of manufacturing heat exchangers from cheap carbon steels capable of resisting the corrosive impact of geothermal brines, which is achieved by protecting the metal with corrosion-resistant coatings, is economically sound.

The practice of operating the geothermal systems shows that the use of both shell-and-tube and plate heat exchangers in the second circuit causes great difficulties. The performance of such heat exchangers becomes impaired with time; they fail frequently and require periodical cleaning from salt-deposition and corrosion products. This results in considerable extra costs and causes frequent stops of geothermal wells, which also reduces the cost-efficiency of the geothermal production.

To utilize the heat of geothermal brines, a heat exchanger of a simple design of the "tube-in-tube" type is proposed. It is assembled out of components

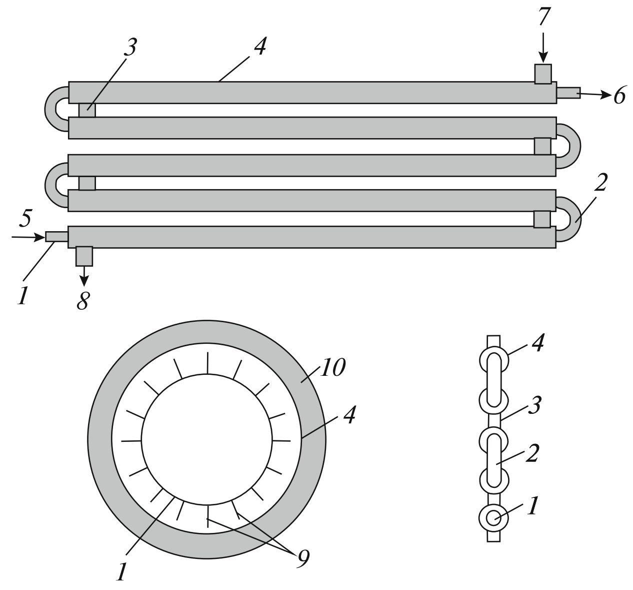

Fig. 1. Tube-in-tube heat exchanger with longitudinal ribbing of the heat exchange surface: *1*—internal tube; *2*—return bend; *3*—connecting branch; *4*—heat-insulated external tube; *5*—delivery of the geothermal heat carrier; *6*—cooled heat carrier drainage; *7*—delivery of the heat carrier to be heated; *8*—heated heat carrier withdrawal; *9*—longitudinal heat transfer ribs; *10*—thermal insulation.

that represent two concentrically positioned tubes of a definite length, 3–6 m, serially connected with each other and placed one above the other (see Fig. 1). The internal tubes are joined with return bends and the external tubes are joined with connecting branches. A heat exchanger of this design was installed in Ternair geothermal steam field in Makhachkala and has been successfully operated there for several years. The heat of saline (to 23 g/dm^3) thermal water (100°C) is transferred to fresh water of the second circuit that circulates in the heating system of apartment houses.

The reduction in the weight and size characteristics of the heat exchangers is a topical problem that can be solved with the help of heat transfer enhancers [2]. One of the methods for enhancement of the heat transfer is longitudinal ribbing of the heat exchange surface. The increase of the heat exchange surface from the side of the heat carrier by ribbing results in an increase in the amount of the heat transferred from the heating agent. In [3], a technique for calculating the effect of the number and size of the longitudinal ribs on the heat transfer in the well heat exchanger and a criterion for the selection of the optimal number and design parameters of the ribs are provided.

The total flux through a tube with longitudinal ribs can be presented in the form of the sum as follows:

$$q = q_1 + q_2 + q_3.$$

The heat flux through the inter-rib surface of the tube is

$$q_1 = \frac{\pi R - n\delta}{\dfrac{\delta_t}{\lambda_w} + \dfrac{\delta_l}{\lambda_m}}(t_1 - t_2).$$

The heat flux through all surfaces of n ribs is

$$q_2 = \frac{2n\lambda_w (t_* - t_2)\left(1 - e^{-kl}\right)\left(A + e^{kl}\right)}{\delta_t \left(A e^{-2kl} + 1\right) k e^{kl}}.$$

The heat efflux through the end part of the ribs is

$$q_3 = 2n\frac{\delta}{\delta_t}\lambda_w \frac{t_1 - t_2}{2\left(1 + \dfrac{\lambda_w}{\lambda_m}\dfrac{\delta}{\delta_t}\right)} \frac{A + 1}{A e^{-kl} + e^{kl}};$$

$$A = \frac{1 - \dfrac{\lambda_w}{\lambda_m}\dfrac{1}{k\delta_t}}{1 + \dfrac{\lambda_w}{\lambda_m}\dfrac{1}{k\delta_t}};$$

$$k = \sqrt{\frac{\lambda_w}{\lambda_m \delta \delta_t}}.$$

In the above equations, R is the radius of the ribbed tube; n is the number of the ribs; δ is the rib wall thickness; δ_l is the thickness of the ribbed tube walls; δ_t is the thickness of the temperature wall layer on the tube and rib surfaces—its values are assumed to be equal in the calculations—λ_w and λ_m are the coefficients of the water and metal heat conductivity, respectively; t_1 is the temperature of the primary heat carrier in the internal tube; t_2 is the temperature of the secondary heat carrier in the annular gap; t_* is the temperature of the outer surface of the internal tube, which is considered to be equal between both the rib ends adjacent to the internal tube's outer surface and the ribs themselves; and l is the rib height.

The thickness of the temperature wall layer can be found according to the formula [4]

$$\delta_t = \frac{2R}{\mathrm{Nu}},$$

where Nu is the dimensionless Nusselt number that characterizes the intensity of the convective heat exchange.

In Fig. 2, the dependences of the thickness of the temperature wall layer δ_t on the velocity of water v at different tube diameters is shown. A reduction in the heat carrier flow rate results in a reduction in the Reynolds and Nusselt numbers, which causes an increase in the thickness of the temperature wall layer. This leads to an increase in the thermal resistance and, consequently, to a decrease in the absolute value of the heat efflux. However, according to the calculation made in [4], an increase in the ratio of the heat flux in the ribbed heat exchanger to the heat flux in the heat exchanger without ribs is observed; the longitudinal ribbing is more efficient at low liquid flow rates.

The designs of the tube-in-tube heat exchangers, the methods for calculation of their parameters and the heat transfer enhancement, the results of the thermal and hydrodynamic analysis, and the selection of the optimization criterion for the design parameters of ribbed well heat exchangers are considered in detail in previous works of the authors [3, 4].

The maximum ratio of the reduced heat flux—the ratio of the heat flux through the ribbed tube to the heat flux through the tube without ribs—to the reduced well flow rate was taken as the criterion for selection of the optimal number and the height of the ribs. Since the well flow rate under fully developed turbulence is proportional to the square root of the pressure difference, we take the ratio of the reduced heat flux to the square root of the value of the reduced pressure difference—the ratio of the pressure difference for the ribbed annular section to the pressure difference for the annular section without ribs [4].

According to the performed calculations, the longitudinal ribbing increases the coefficient of the heat exchange between the fluxes by several tens of percent to several times depending on the number and the height of the ribs. When deriving the thermal conductivity equation for a rectangular rib with a length-constant cross section, the change in the temperature only along the rib was considered, while the temperature across the section was either averaged or assumed to be equal. With respect to the tube, its thickness was accounted for, while the radial temperature distribution was considered the same for all points. The new method for calculation of the heat transfer through a ribbed surface accounts for the rib thickness and the temperature change across the rib. The tube thickness and both the radial and circumferential temperature changes in the tube were also taken into account. This means that a general two-dimensional problem of the temperature distribution is considered. The temperatures of the rib and tube change in both longitudinal and transverse directions.

The solution of this problem with a large number of the ribs can be modeled by one symmetry element, which represents an L-shaped figure. It comprises a half of the rib dissected in the middle and the adjacent half of the tube piece. The L-shaped figure is divided into three rectangular parts: the first part is the rib, the second part is the tube piece at the joint with the rib, and the third part is the tube piece between the ribs, which has height h and thickness 2δ. For each of the parts, the temperature distribution is determined under the boundary conditions that model the heat exchange, the symmetry, and the adopted change in the temperature along the joint lines. In this two-dimensional problem of the temperature distribution in a ribbed tube, it is assumed that the rib of an indefinite height is flown around by a turbulent water flow

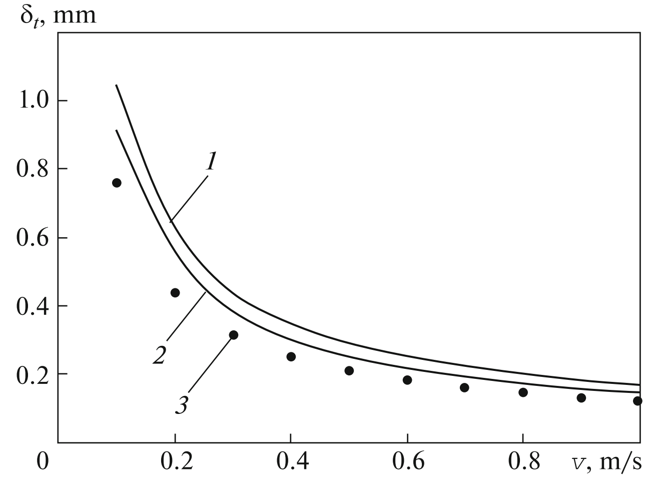

Fig. 2. Thickness of the temperature wall layer depending on the water flow rate at different tube diameters: *d*, mm: *1*—100; *2*—50; *3*—20.

with a constant temperature, which is taken as $t_w = 0$ for the sake of simplicity, and the entire heat added to the rib will be transferred into the ambience, i.e., $t(\infty, y) = 0$. The boundary condition for a piece of the tube at the joint with the rib and for the inter-rib piece of the tube is $t(0, y) = 1$. The temperature distribution across the joint between the second and third parts is presented in the form of a linear function, while the temperature distribution at the joint between the first part—the indefinitely long rib—and the second part is an even function, a second-order parabola. The method is sufficiently efficient and reliable for the solution of the problem under consideration. The problem is solved in the Fourier series for each of the parts individually. At the joints between the first and second parts, as well as the second and third parts, the temperature profiles are approximated by polynomials with coefficients not known in advance. These coefficients are found from the condition of the equality of the heat fluxes at the joints. In Fig. 3, the obtained temperature distribution pattern is shown in three regions of the contact problem in accordance with the adopted boundary conditions and the parameters of the ribbed surface.

The obtained solution allowed determining anew the coefficients of the heat transfer through a ribbed surface considering the thicknesses of both the tube and the rib. Comparison with previously obtained results [4] show that the influence of the averaging of the temperature across the tube and the rib is insignificant for millimeter-scale thicknesses. The averaging of the temperature across the rib can be well performed with thicknesses of several millimeters.

Below, we present the calculated characteristics of tube-in-tube heat exchangers with the ribbing and without it intended for picking up the heat from the

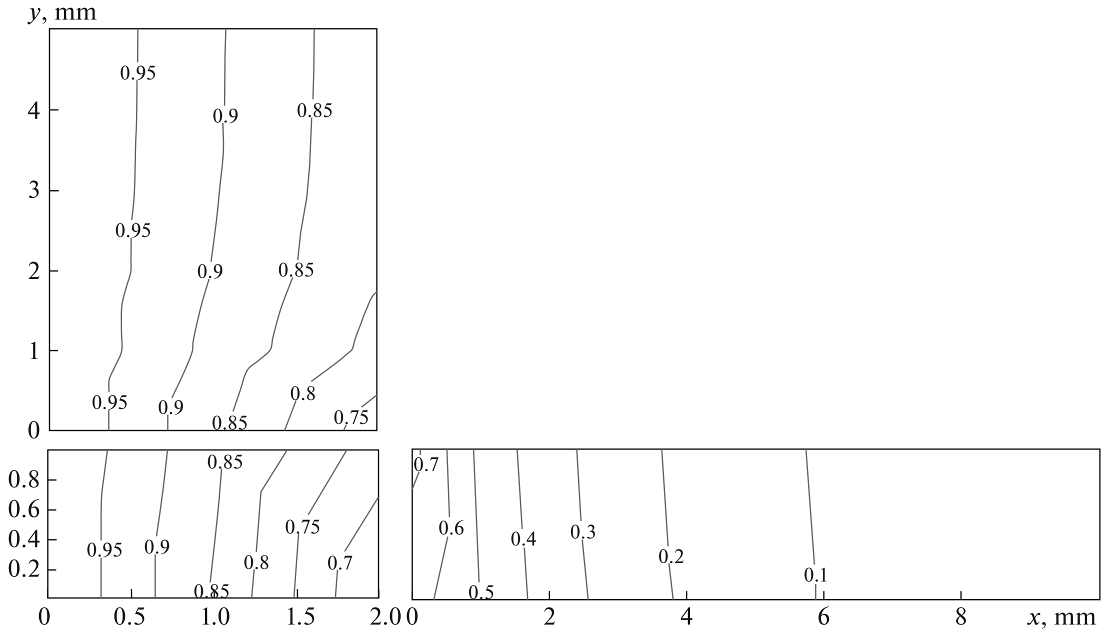

Fig. 3. Temperature distribution (°C) in the regions of the contact problem at $\lambda_m = 45$ W/(m K); $\delta = 1$ mm; $h = 5$ mm; $\delta_t = 0.125$ mm.

saline thermal water of Ternair geothermal steam field:

Number of the branches	2
Mass thermal water flow rate, kg/s	45
Flow rate in the internal tube, m/s	3.0
Mass flow rate of the heated water, kg/s	40
Inner diameter (wall thickness), mm:	
of the internal tube	100 (7)
of the external tube	142 (5)
Inlet and outlet temperature, °C:	
of the thermal water	98; 60
of the heated water	50; 90

In Table 1, the parameters that depend on the heat exchanger type are provided.

The thermal water field is operated in the intermittent mode during the heating season to supply heat to a local community with apartment houses. The mass flow rate of two wells is 45 kg/s; the thermal saline water is delivered to the heat exchange system at a temperature of 98°C. Upon extraction of the heat in the heat exchanger, the thermal water with a temperature of 60°C is discharged into a drainage duct. The water that circulates in the heating system of apartment houses is heated to 90°C before being feed into the heating facilities and, then, with a temperature of 50°C, it is delivered again to the heat-exchange system to be heated.

Table 1. Characteristics of the tube-in-tube heat exchangers

Parameter	Heat exchanger	
	ribbed	without ribbing
Flow rate in the intertubular space, m/s	4.0	3.7
Longitudinal ribs on the outer surface of the internal tube:		–
number, pcs	16	
height, mm	12	
thickness, mm	2	
Total length of the heat exchanger, m	72	84
Number of 6-m long sections, pcs	12	14
Heat exchanger weight, kg	5800	6250

Comparative analysis of the variants of the heat exchange surface with longitudinal ribbing and without it was conducted. It follows from the data presented in Table 1 that the weight and size characteristics of the heat exchanger are improved by ribbing it.

To inhibit the salt deposition in the internal tube, the flow rate of the brine should be approximately 2.5–3.0 m/s. According to the studies conducted directly on the well of Ternair geothermal field, the salt deposition rate is determined by the Reynolds number, the concentration of the suspended particles in the geothermal brine solution, and its calcium-carbonate oversaturation, which depends on the brine temperature and the pressure in the system [5]. It has been shown that, with all other conditions being equal, the salt deposition rate is lower at high Reynolds numbers. For the proposed tube-in-tube heat exchangers, the Reynolds number from 5×10^5 to 1×10^6 corresponds to the velocity of the brine in the internal tube of approximately 2.5–3.0 m/s. A further increase in the flow rate is unreasonable, since it leads to an increase in the hydraulic resistance in the heat exchanger and extra power inputs for pumping the heat carrier.

The heat exchanger should be cleaned from carbonate deposits with the thermal water that passes through it with a carbon dioxide concentration in the water higher than the equilibrium value, which enables dissolution of the deposits. This technique is being successfully applied at OAO Geotermneftegaz in the wells of Ternair thermal steam fields.

In addition, to prevent corrosion and salt deposition in the heat exchanger, the effective OEDFK (oxyethylidenediphosphonic acid) agent can be used. This agent has a long-lasting corrosion-inhibiting and antiscaling effect, which is explained by the formation on the surface of a strong adhesive chelate layer difficult to wash off the surface. The passivating OEDFK layer is restored by periodical, twice a month, pulsed introduction of the agent solution into the brine at the heat exchanger inlet [1]. Oxyethylidenediphosphonic acid is available in the form of a powder that is easy to dissolve in water in an agent to water ratio 60 : 100 g at room temperature. The agent is used in heat supply systems, hot water supply systems, closed-circuit cooling systems, steam boilers, evaporators, and distillation evaporation plants. The application of oxyethylidenediphosphonic acid is regulated by the SO 34.37.536-2004 methodology guidelines on the use of antiscalaing agents and corrosion inhibitors tested and certified at RAO Unified Energy Systems of Russia and power enterprises.

CONCLUSIONS

(1) The tube-in-tube heat exchangers are most efficient for the pickup of the heat from thermal brines.

(2) Longitudinal ribbing increases the heat exchange surface and the amount of the heat transferred from the heating agent and reduces the weight and size characteristics of the heat exchanger.

(3) Longitudinal ribbing is more efficient at low fluid flow velocities with the optimal rib height ranges within 5–15 mm and the number of the ribs ranging from 16 to 32. With the increasing number of the ribs, the heat extraction is enhanced; the hydraulic resistance, however, increases. The thermal insulation of the outer heat exchanger surface allows reducing the environmental heat losses.

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