

EXPERIMENTAL INVESTIGATION OF A MANIFOLD HEAT-PIPE HEAT EXCHANGER

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Results of experimental investigations of a heat exchanger on a manifold water heat pipe are given. An analysis is made of the temperature distribution along the heat-transfer agent path as a function of the transferred heat power. The influence of the degree of filling with the heat transfer agent on the operating characteristics of the construction is considered.

The difference in the design of a heat exchanger on a manifold heat pipe (MHP) from a traditional heat pipe or thermosiphon limits using currently available theoretical and experimental data. It is noteworthy that to date the MHP-related direction invites further study. Individual theoretical and experimental investigations of MHP and heat exchangers on their basis are reported in [1, 2]. The present investigation is concerned with an experimental study of special features of energy and material transfer both inside the considered heat exchanger type and in the external air paths of the MHP heat exchanger.

To achieve the purpose in hand, we produced a gas-gas-type manifold heat exchanger of a contour design (Fig. 1a) calculated numerically according to an analytical procedure of [1, 3] that involves:

- Prescribing the initial data, i.e., geometric, thermophysical, and physicochemical parameters of the heat-transfer agents and structural materials.
- Numerical calculations of : 1) thermal resistances of individual portions of the heat exchanger including heat-exchange elements, 2) thermal balance including the corresponding heat fluxes, the output thermal parameters of gas flows, 3) thermal efficiency of the heat exchanger.
- A hydraulic calculation of the internal (in MHP) and external (gas flows) mass transfer.

When designing the MHP heat exchanger we selected polymodal high-efficiency heat-exchange elements as evaporation and condensation sections [4] (Fig. 1b). Flat aluminium elements are finned using a nonwaste technology that consists in cutting and turning back thin metal layers (fins) with the prescribed parameters from the ingot surface with their firm bond to the bases retained (Fig. 1c). The developed technology enables us to vary the height and thickness of the fins, to ensure their optimum arrangement, makes the thermal resistances at the sites of the fin-bases transition minimal. Furthermore, the method enables us to produce elements with curvilinear rows of fins, each being divided into individual petals which ensures intense turbulence of the air flow that interacts with the fins and a high heat-transfer coefficient at low energy consumption. A decrease in the required surface area of the heat exchanger, overall dimensions, and specific quantity of metal of the exchanger is made possible.

Based on numerical precalculations we produced two heat-exchange sections with the following parameters: the external overall dimensions are $0.22 \times 0.126 \times 0.032$ m, the distance between the fins is 2 mm, the number of fins is 888, the dimensions of the external fins of the evaporator and the condenser are $4 \times 6 \times 1$ mm, the fin efficiency is 6.3, the total fin area of the heat exchanger is 2×0.2754 m², the parameters of condensation and evaporation channels are 2×5 mm, the number of flat modules in one section is 7, the number of channels in one module is 4, the geometric parameters of one module are 18×32 mm, the working external intake clearance of the evaporator and the condenser is 0.02105 m², the intake clearance of the evaporator and the condenser is 0.02272 m², and the relative clearance of the area is 0.7513.

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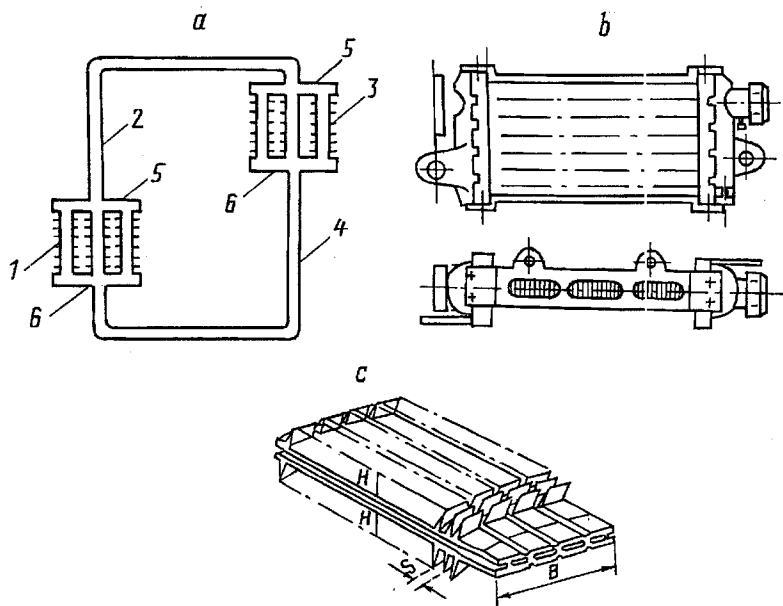


Fig. 1. Design of MHP heat exchanger: 1) evaporator, 2) vapor channel, 3) condenser, 4) liquid channel, 5) distributing manifolds, 6) collecting manifolds (a); projections of the heat-exchange sections of the MHP heat exchanger (b), and flat elements of polytube modules with cut fins (c).

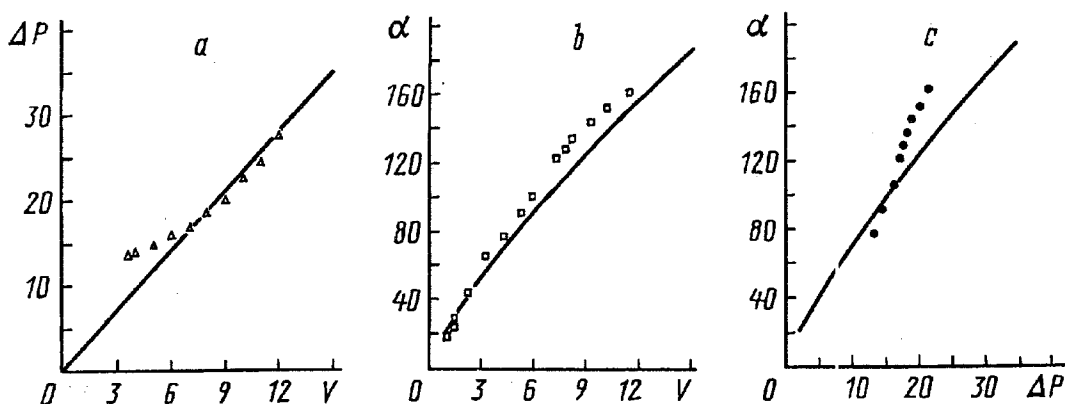


Fig. 2. Theoretical and experimental results of calculating the heat-exchange portion (evaporator and condenser): the hydrodynamic resistance vs the air flow velocity (a); the heat-transfer coefficient on the outer surface vs the air flow velocity (b) and pressure drop (c). ΔP , Pa; α , $W/(m^2 \cdot K)$; V , m/sec.

Results of numerical calculations and experimental tests of hydraulic characteristics of the evaporation and condensation heat-exchange sections are given in Fig. 2.

First we investigated the hydraulic resistance as a function of the mass rate of the air flow in the section (Fig. 2a). The maximum disagreement of the experimental data with the calculations was 14%. The results obtained in this experimental run permit the conclusion of insignificant pressure losses along the air path of heat-exchange elements with cut fins. We can explain the disagreement between the theoretical and experimental data at small velocities by inaccurate measurements.

In the second part of the preliminary experiments we studied the heat-transfer coefficient in the interfin space as a function of the pumped air mass rate (Fig. 2b). We compared the obtained experimental data with the results of the numerical calculation by the developed procedure; the error was no more than 15%. The performed investigations yield substantial values of the heat-transfer coefficient of up to 200 $W/(m^2 \cdot K)$. In a qualitative comparison of the data obtained with the heat transfer in rolled-on fins for the cut finning we also obtained a 12-45% excess for α that increases as the mass rate increases.

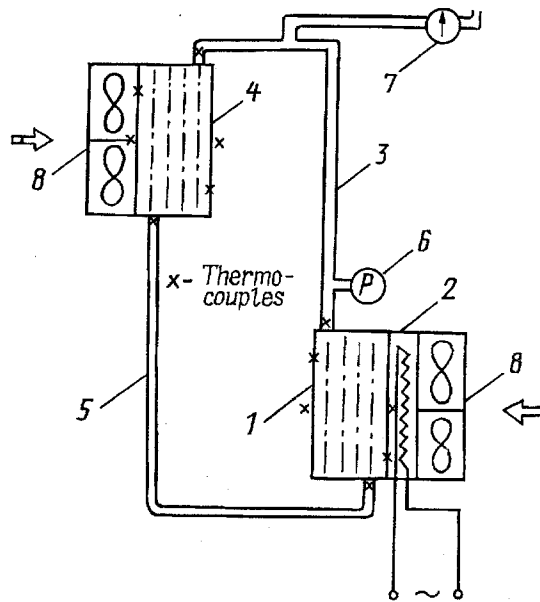


Fig. 3. Experimental setup of MHP heat exchanger: 1) evaporator, 2) electric heater, 3) vapor channel, 4) condenser, 5) liquid channel, 6) manometer, 7) vacuum pump, 8) fans.

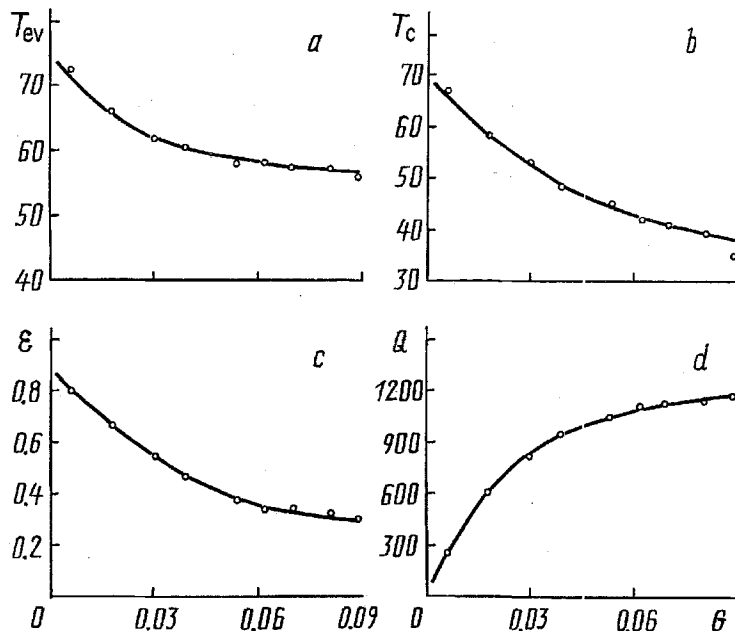


Fig. 4. Experimental heat-engineering characteristics of MHP heat exchanger (the hot air mass rate $G = 0.09$ kg/sec; $T_{inl.ev} = 25^\circ\text{C}$; $T_{inl.c} = 25^\circ\text{C}$): the air temperature at the outlet from the evaporator (a) and condenser (b) as well as the heat exchanger efficiency (c) and the transferred heat flux (d) vs. the change in the air mass rate in the condenser. T_{ev} , T_c , °C; Q , W; G , kg/sec.

The gain in α attained on account of the increased pressure losses should be considered an effective characteristic of the heat-exchange sections in question (Fig. 2c). The shape of the obtained curve $\alpha = f(\Delta p)$ permits the conclusion of an intensifying effect that occurs in the fins in question.

The structurally investigated MHP heat exchanger incorporated the two above-considered heat-exchange sections, which for visualization purposes are connected by transparent glass channels for the vapor and liquid (Fig. 3). Water served as the heat-transfer agent. The air flows were varied, the temperature of a cold incoming air flow being maintained at a constant level of $T_{inl.air} = 25^\circ\text{C}$, while that of the hot one was varied from 25 to 100°C . We

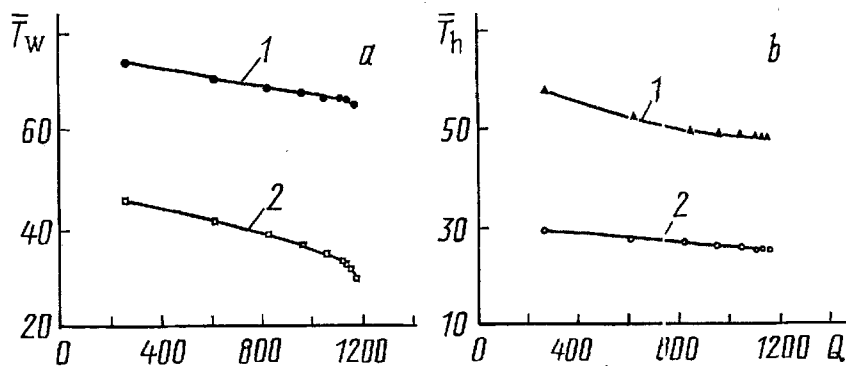


Fig. 5. Average wall temperature vs. the transferred heat flux of MHP heat exchanger: 1, evaporator, 2, condenser (a) and the vapor and liquid temperature in MHP vs. the transferred heat flux: 1, vapor, 2, liquid (b). T_w , T_h , °C.

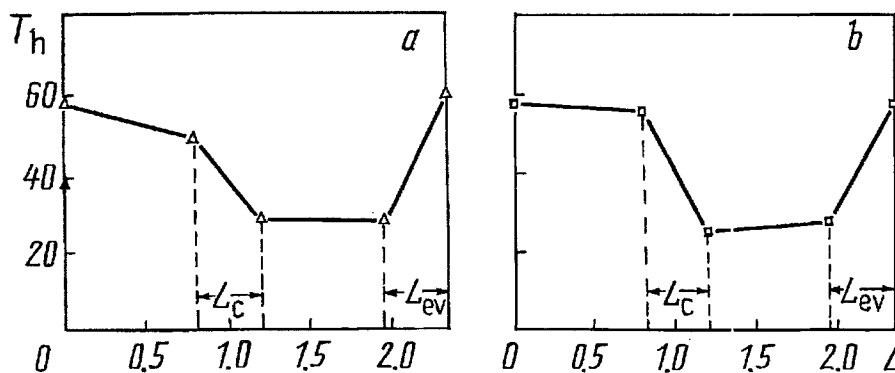


Fig. 6. Diagram of the temperature distribution of the heat-transfer agent in MHP: a) $Q = 615$ W, b) 1120 W.

recorded the vapor and liquid temperatures, the hot and cold air flow temperatures as well as the temperature fields over the heat-exchange element surface using a set of thermocouples. Furthermore, we measured mass and linear rates of the vapor, liquid, and air; a number of parameters were numerically determined.

As a result of investigating we obtained experimental data on the temperature of the hot and cold air flows at the outlet from the finned sections of the MHP heat exchanger condenser and evaporator with variation of the mass rate in the condenser and at fixed temperatures of the flows at the inlet to the sections (Figs. 4a and 4b). Satisfactory results of comparing the experimental data with the numerical calculations enable us to recommend the proposed procedure for analyzing and designing MHP-based heat exchangers.

A substantial part of the work was studying the operating parameters of the heat exchanger, namely, the thermal efficiency $\epsilon = Q/Q_{\max} \cdot (t' - t'')/(T' - t'')$ and the transferred heat flux. Figures 4c and 4d give the results of the tests as the dependences $\epsilon = f(G)$ and $Q = f(G)$ and compare them with calculation. It is evident that as the mass rate increases the thermal efficiency ϵ of the heat exchanger decreases and the transferred heat flux increases. It is noteworthy that for relatively small overall dimensions and flow-rate characteristics G the maximum power of the manifold-type heat exchanger attains $Q_{\max} = 1200$ W at thermal efficiency $\epsilon = 0.3$. The present series of investigations also shows a satisfactory agreement of the calculations with experiment; the disagreement was 10%.

Figure 5 gives the results of investigating the temperature distribution over the manifold heat exchanger surface. Analysis of the obtained curves for averaged wall temperatures (Fig. 5a) and a two-phase heat-transfer agent (Fig. 5b) for the evaporation and condensation heat-exchange sections as functions of the transferred heat flux showed that there is a substantial decrease in T with increasing Q .

Heat losses in the vapor and liquid channels lead to insignificant changes in the heat-transfer agent temperature in the channels as shown in Fig. 6. It is evident that there are large temperature changes in the evaporator and condenser. This is explained by the fact that the heat-transfer agent is cooled by a certain value

of ΔT in the condenser; correspondingly, the heat-transfer agent is heated in the evaporator with the supercooled-to-saturated state transition.

It is noteworthy that the above experimental results are obtained at a constant degree of MHP filling with the heat-transfer agent. Therefore, to determine the field of application of the results obtained, we considered the influence of the degree of MHP filling with the heat-transfer agent on heat exchanger operation. Experimental investigations were performed at constant values of the heater power and the mass rate of the air in the evaporator and the condenser.

The tests showed that there is a dependence of the transferred heat flux on the degree of filling. The dependence $Q = f(\Omega)$ is of a pronounced character at small Ω and the drop is insignificant at large Ω it is only with the filling $\Omega > 90\%$ that a drastic drop in Q occurs. This can be physically explained by the influence of the degree of filling on the heat exchanger thermal resistance. This character of the dependence is substantially affected by the phenomenon that occurs when, at a certain value of Ω , a vapor-liquid mixture hits the upper manifold and the vapor channel. This leads to increased hydrodynamic resistance in the upper manifold and vapor channel. If the vapor-liquid mixture arrives at the condenser the transferred heat flux decreases.

In summary, we can make the following conclusions from the experimental data obtained:

- MHPs have a high heat-transfer property with certain design advantages.
- The procedure of [1, 3] enables us to calculate the class of heat exchangers in question.
- A set of experimental data is obtained, in which we should highlight the optimum degree of filling the evaporation section volume with the liquid phase of the heat-transfer agent (40–55%).

NOTATION

Q , heat flux; T, t , temperature; G , mass rate; ΔP , pressure drop; V , flow velocity; α , heat-transfer coefficient; ϵ , thermal efficiency of the heat exchanger; L , length; Ω , degree of filling with the heat-transfer agent. Subscripts: c, condenser; ev, evaporator; h, heat-transfer agent; w, wall; air, air; inl, inlet; out, outlet.

REFERENCES

1. S. V. Konev and Wang Tszin' Lyan', Manifold Heat Pipes [in Russian], Minsk (1992). (Preprint ITMO AN BSSR, No. 7), pp. 1-44.
2. S. V. Konev and J. L. Wang, The Proc. of the 8th Int. Heat Pipe Conf., Beijing, China (1992).
3. S. V. Konev and Wang Tszin' Lyan', Vestsi Akad. Nauk Belarusi, Ser. Fiz. Tekhn. Navuk, No. 3, 11-16 (1994).
4. I. I. D'yakov, S. V. Konev, G. V. Vasilieva, and E. M. Kosmachova, Heat Recovery System and CHP, 10, No. 1, 49-53 (1990).