Experimental research of heat recuperators in ventilation systems on the basis of heat pipes

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Abstract. The paper presents the results of experimental studies of heat pipes and their thermo-technical characteristics (heat power, conductivity, heat transfer resistance, heat-transfer coefficient, temperature level and differential, etc.). The theoretical foundations and the experimental methods of the research of ammonia heat pipes made of aluminum section AS – KRA 7.5 – R1 (made of the alloy AD - 31) are explained. The paper includes the analysis of the thermo-technical characteristics of heat pipes as promising highly efficient heat transfer devices, which may be used as the basic elements of heat exchangers - heat recuperators for exhaust ventilation air, capable of providing energy-saving technologies in ventilation systems for housing and public utilities and for various branches of industry. The thermo-technical characteristics of heat pipes (HP) as the basic elements of a decentralized supply-extract ventilation system (DSEVS) and energy-saving technologies are analyzed. As shown in the test report of the ammonia horizontal HP made of the section AS-KRA 7,5-R1-120, this pipe ensures safe operation under various loads.

1 Introduction

The purpose of this research is to create the scientific and technical basis for the design, manufacturing technology and tests of energy-saving decentralized supply-extract ventilation systems (DSEVS) based on new generation HP with tear-drop structure of the core and optimal constructive-processing and performance characteristics [1].

The objective of this research is to select optimal constructive-processing characteristics [2] for creating DSEVS based on high conductivity ammonia heat pipes (HP).

Special attention is paid to the rational and complex use of a flat HP with biphase heat-transfer fluid as a partition, and the organization of heat input in the case of flow-over of warm, exhausted air (the evaporation zone below) and heat removal (the condensation zone above), using fin fan computer heat exchangers (aluminum, aluminum-copper radiators) [3, 4]. The objects of the research are HP and fans-heat recupretors based on high-conductivity heat pipes [5].

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Notation system

σ	coefficient of surface tension
ρ_l	liquid density
ρ_v	steam density
μ_l	liquid dynamic viscosity
μ_v	steam dynamic viscosity
d_{hp}	outside diameter of the pipe
d_{in}	inside diameter of the pipe
l_e	length of evaporation zone
l_c	length of condensation zone
l_{hp}	geometrical length (height) of the heat pipe
$l_{add,\,e}$	length of evaporation zone under additional or bottom heater
$l_{add,\,c}$	length of condensation zone under additional heater
T_1	temperature of evaporator shell (T_1=T_u)
T_2	temperature of condenser shell (T_2=T_κ)
λ_l	heat conductivity of ammonia
L	evaporation heat
g	intensity of gravity
Q	heat transfer ability of the thermal syphon or heat pipe
ΔP_s	capillary pressure of liquid
ΔP_f	capillary pressure of film
D_v	diameter of steam passage channel
G	amount of heat-transfer fluid put in the interior of the heat pipe

2 Materials and methods

The paper presents the integrated development of methodological support for thermovacuum tests of honeycomb panels with built-in heat pipes, with simulators of heat-emitting equipment: physical analogues of vertically and horizontally located heat pipes for thermophysical tests; the formation of excess liquid in the HP and its influence on the thermotechnical characteristics of the HP are analyzed; the technique for performance calculation of the HP with heat input in the bottom part that operates in the biphase thermosyphon regime; the technique of performance calculation of the HP with several simulators of heat-emitting equipment that operates in the biphase thermal syphon regime under terrestrial conditions, and the main features of the thermal operating mode are revealed; the experimental values of heat transfer coefficients in the evaporation and condensation zones of the HP are given; the example of calculating the thermo-technical characteristics of the HP made of the section AS-KRA7.5-R2; the experimental results of the tests of the HP made of the section AS-KRA7.5-R1 with several heat sinks are analyzed; the experimental results of tests of the horizontal HP made of the section AS-KRA7.5-R1 with several simulators of heat-emitting equipment are analyzed [6].

Formation of liquid droplets. Due to the high speed of steam, it is impossible that free-floating liquid droplets exist for a long time. Such droplets, if created as a result of liquid film stripping from the wave crests by countercurrent flow of liquid and steam, will be carried away by steam and will settle at the end of the condensation zone, forming an inactive leg.

Formation of a liquid slug. Under overload conditions (in 0-g medium), excess liquid may accumulate at the end of the condensation zone of the heat pipe and form a liquid slug, that completely fills the cross section of the steam passage channel of the pipe. This so-called slug reduces the active effect of high temperature of the steam flow on the remaining active leg of the heat pipe.

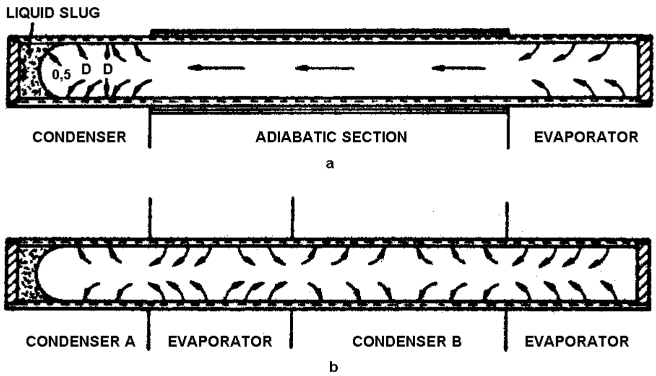

Fig.1. a,b. The configuration of a liquid slug in the heat pipe with different location of condenser and evaporator.

Formation of a liquid film. The liquid film layer on the condenser's wall may have a stable configuration if the condenser is not located close to at least one end of the manufactured heat pipe. Due to the low heat conductivity of liquid, the thermal characteristics (transfer ability, power) of the heat pipe can be significantly reduced.

However, combined with the slug, the liquid film is not a stable configuration. Let us consider a simple configuration of the HP with an evaporator at one end and a condenser at the other (Figure 1,a).

In the case of a liquid slug formation, the stable surface of the steam passage channel's interface would be a hemisphere (although there will be a form somewhat different from this ideal contour, due to the influence of the steam flow). If the excess liquid forms a film layer in the condensation zone adjacent to the slug, then the two separate and communicated (having hydraulic connection) liquid formations exist as a system [7, 8]. However, the capillary pressure of these two surfaces is absolutely different. The capillary pressure for the liquid ΔP_s and the cylindrical layer of film ΔP_f is expressed respectively as

$$\Delta P_S = \frac{4\sigma}{D_v} \tag{1}$$

$$\Delta P_f = \frac{2\sigma}{D_v} \tag{2}$$

That is, the capillary pressure of the formed liquid slug is twice more than that of the liquid film layer. So, we can make the conclusion that the liquid film layer in the condensation zone located close to the end of the pipe is not stable. That excess fluid will be involved in the more stable configuration of the liquid slug.

In Fig. 1,b we can see another, though typical situation - the condenser placed in the middle of the pipe is between two evaporation zones. The only liquid bond between this area and the area of the slug in the end of the pipe is through the capillary grooved structure. However, the capillary pressure of these grooves in the evaporation zone is higher than on the slug's surface, so the supply of the possible mass of liquid from the film layer to the slug can not take place. For this configuration it is necessary to consider completely different positions of various areas of the condenser and evaporator.

To provide the given heat power *Q,* besides the temperature drop T_1-T_2 along the interior wall of the thermal syphon (heat pipe), the thermo-physical characteristics L, ρ_l, λ_l, μ_l, the geometry of the pipe, it is necessary to provide the geometry of the evaporation and condensation zones and the correlation between them. Moreover, this dependence is non-linear and not arbitrary. The condensation zone l_c is located above the evaporator (heating area), and the condensate film moves only downward under the action of gravity.

In the horizontal position of HP ($\alpha = 0^0$), the number of evaporation zones along the pipe's length may be arbitrary, when the mass inflow of liquid condensate into the evaporation zone (simulator of heat-emitting equipment) takes place under the influence of capillary forces both to the left and to the right of the evaporator.

The summary temperature di ΔT between the evaporation and condensation zones along the exterior surface of the heat pipe consists of the temperature drops through the thickness of the shell's wall in evaporation $\Delta T_{w,e}$ and condensation $\Delta T_{w,c}$ zones, through the thickness of the condensate film in evaporation $\Delta T_{f,e}$ and condensation $\Delta T_{f,c}$ zones:

$$\Delta T=\Delta T_{w,e}+\Delta T_{w,c}+\Delta T_{f,e}+\Delta T_{f,c} \tag{3}$$

The sum $\Delta T_{f,e}+\Delta T_{f,c}$ is the temperature different along the interior surface of the heat pipe lengthwise, i.e. $\Delta T_{f,e}+\Delta T_{f,c}= T_1$-$T_2$.

The temperature different s $\Delta T_{w,e}$ and $\Delta T_{w,\kappa}$ are defined from the equations of heat transfer via heat conductivity through cylindrical wall:

$$\Delta T_{w,e} = \frac{Q}{\pi l_e}\frac{1}{2\lambda} ln\frac{d_{in}}{d_{hp}} \tag{4}$$

$$\Delta T_{w,c} = \frac{Q}{\pi l_c}\frac{1}{2\lambda} ln\frac{d_{in}}{d_{hp}} \tag{5}$$

The heat pipe has got a uniform cross-section and a thin wall. Having substituted the values of temperature drop components in equation (3), we will obtain

$$\Delta T = \frac{Q}{2\pi\pi} ln\frac{d_{in}}{d_{hp}}\left(\frac{1}{l_e}+\frac{1}{l_c}\right)+\left(T_1-T_2\right) \tag{6}$$

When making calculations for a smooth-wall heat pipe, the geometry, material of the shell, type of heat-transfer fluid, operating conditions of the heat pipe in the evaporation zone or in the condensation zone are presupposed.

The dependence $Q=f_1(T_e, \Delta T)$ by T_c = const or $Q=f_1(T_e, \Delta T)$ is established, here by T_e = const T_c, T_e are the temperatures on the exterior surface of the heat pipe in the evaporation and condensation zones. Having made the calculations for a number of values of Q, the dependencies $Q=f_2(T_c, \Delta T)$ are established.

As shown by the experimental research [9, 10], the temperature drop in the steam passage channel depends on the amount of heat-transfer fluid filling the pipe, and on the tilt angle of the axis of the heat pipe to the horizon. The three operation modes of the smooth-wall heat pipe (Fig.2) differ in the amount of filling heat-transfer fluid.

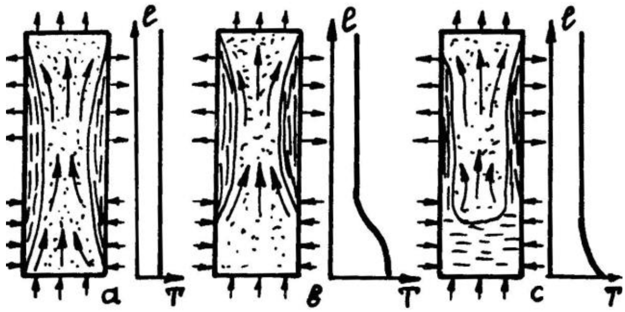

Fig.2. Influence of the heat pipe's filling degree on its operation mode.

1. Isothermal mode of the heat pipe's operation (Fig. 2,a). In this mode the amount of the filling heat-transfer fluid fully corresponds to the value of the transferred heat flow.
2. The mode of insufficient filling (Fig. 2,b). In this case there is not enough liquid for the film to cover the interior surface of the walls of the heat pipe's shell completely. In bare areas there is localized overheating of the wall.
3. The mode of excessive filling (Fig. 2,c). By excessive filling, a "puddle" forms on the bottom of the heat pipe. Excessive filling with heat-transfer fluid is less dangerous for the operation of the heat pipe, but there is significant temperature drop along the height of the "puddle".

For the HP made of section AS-KRA 7.5-R2, built into the thermopanel, the third mode (c) is typical, with excess heat-transfer fluid (up to about 10%) accumulating on the bottom of the thermal syphon, as the wall temperature in the condensation zone practically doesn't change, and in the evaporation zone it falls, starting from the lower limit.

For optimal filling of the heat pipe, we get

$$G = \left(\frac{4}{5}l_c + l_{hp} + \frac{4}{5}l_e\right)\sqrt[3]{\frac{3Q\mu_l\rho_l\pi^2 d_{hp}^2}{Lg}} \tag{7}$$

The calculation of the characteristics of the heat pipe with several simulators of heat-emitting equipment that operate in the biphase thermosyphon regime under terrestrial conditions is carried out with the help of the method described below.

When the heat pipe (HP) operates with several simulators of heat-emitting equipment in the thermal syphon regime, the heat flows differ in size and density

$$Q_i = \frac{4}{3}\pi d_{hp}\sqrt[4]{\frac{L_i\rho_{l,i}^2 g\lambda_{жс,i}^3\left(T_{1,i}-T_{2,i}\right)^3 l_{e,i}^3 l_{c,i}^3}{4\mu_{l,i}\left(l_{e,i}+l_{c,i}\right)^3}} \tag{8}$$

I.e. the biphase thermosyphon as if consists of several component thermosyphons, each with its own evaporation $l_{u,i}$ and condensation $l_{к,I}$ zones, joined together in the same shell.

$$\sum_{i=1}^{n}\left(l_{e,i}+l_{c,i}\right)+\left(l_{addi}+l_{addi}\right)=l_{hp} \tag{9}$$

By vertical arrangement of the honeycomb panel, the role of the additional heater may be played by the simulator of heat-emitting equipment, shifted to the lower limit position. Then the total heat load from the simulators of heat-emitting equipment (heaters), including the load on the additional heater, will be:

$$Q_\Sigma = \sum_{i=1}^{n} Q_i + Q_{add,h} = \sum_{i=1}^{n}\left[\frac{4}{3}\pi d_{hp}\sqrt[4]{\frac{L_i\rho_{l,i}^2 g\lambda_{l,i}^3\left(T_{1,i}-T_{2,i}\right)^3 l_{e,i}^3 l_{c,i}^3}{4\mu_{l,i}\left(l_{e,i}+l_{c,i}\right)}}\right]+Q_{add,h} \tag{10}$$

In this case, the total heat load may be determined from the heat release cyclogram of the heat-emitting equipment. The optimal filling value is calculated at the total heat load Q_Σ with the help of the formula:

$$G=\left(\frac{4}{5}l_{c,i}+l_{hp,i}+\frac{4}{5}l_{e,i}\right)\sqrt[3]{\frac{3Q_\Sigma\mu_{l,i}\rho_{l,i}\pi^2 d_{hp}^2}{L_i g}} \tag{11}$$

During tests the value $Q_{add,h}$ is a variable, and it's the part ξ of $\Sigma^n_{i=1}Q_i$. Then the total heat load for the thermal syphon with simulators of heat-emitting equipment will be:

$$Q_\Sigma = \sum_{i=1}^{n} Q_i + \xi\sum_{i=1}^{n} Q_i = (1+\xi)\sum_{i=1}^{n}\left[\frac{4}{3}\pi d_{hp}\sqrt[4]{\frac{L_i\rho_{l,i}^2 g\lambda_{l,i}^3\left(T_{1,i}-T_{2,i}\right)^3 l_{e,i}^3 l_{c,i}^3}{4\mu_{l,i}\left(l_{e,i}+l_{c,i}\right)}}\right] \tag{12}$$

The complex ratio ξ depends on the thermal conditions of the heat-emitting equipment (from the cyclograms of internal heat release), on the conditions of the launch modes of the honeycomb panel with built-in HP, on the location of the controls on the honeycomb panel, and on other factors, and it's obligatory to determine it experimentally during testing.

During the operation of the i-th heat-emitting device, the liquid condensate film on the top of the condenser flows to the i-th evaporator zone, and the liquid condensate film on the bottom part of the condenser summarizes, after the condensation of saturated steam, with the condensate film from the $(i + 1)$-th device, and flows under the action of gravity into the evaporation zone of the $(i + 1)$-th device. It can not move up to the evaporation zone of the top device. Thus, when arranging simulators of heat-emitting equipment, it is necessary to implement the principle: the increase in the heat power of devices from top to bottom when they are positioned along the height of the biphase thermosyphons.

3 Results

As the results of testing the HP made of the section AS-KRA7.5-R1-120 or R2 show, the symmetrical distribution of thermal power between two simulators of heat-emitting equipment lengthwise in horizontal position and heightwise in vertical position, when Q_1=Q_2=5; 10; 15; 20 W contributes to the increase in the degree of isothermality of the honeycomb panel's temperature pattern. The surface temperature does not exceed 40 °C, and the longitudinal and transverse temperature drops in the evaporation and condensation zones are quite acceptable.

Table 1. Development of the input control method of the thermo-physical characteristics of the HP with grooved structure of the core, in case of several heat sinks by way of computer fans in DSEVS

№	τ, hrs	$T_{air,}$ ^{0}C	Q, W	T_1		T_3		T_2		ΔT	Note
				мВ	^{0}C	мВ	^{0}C	мВ	^{0}C	^{0}C	
1	16^{15}	21	20	0.58	29.857	0.57	29.714	0.52	29.0	0.857	With exit fan № 1
2	16^{30}	21	20	0.57	29.714	0.58	29.857	0.51	28.833	0.881	With exit fan № 1
3	16^{45}	21	20	0.55	29.429	0.56	29.429	0.5	28.66	0.769	With two fans № 1, 2
4	17^{00}	21	20	0.44	27.714	0.46	28.0	0.41	27.286	0.428	With three fans № 1, 2, 3

The temperature sensor T_1 is installed in the heating (evaporation) zone in the middle of the flange in the place where the heater is fixed; the temperature sensor T_2 is in the zone of heat removal (condensation) in the middle of the flange along the same generatrix in the places where the cooler is fixed; the temperature sensor T_3 is in the middle of the transport zone.

Table 2. Development of the input control method of the thermo-physical characteristics of the HP with tear-drop grooved structure of the core and several computer fans

№	τ, hrs	T_{air}, ^{0}C	Q, W	T_1		T_2		ΔT	Note
				мВ	^{0}C	мВ	^{0}C	^{0}C	
1	11^{45}	19	15	0.33	24.0	0.28	23.286	0.714	With all the four fans
2	12^{00}	19	15	0.31	23.714	0.27	23.143	0.571	With all the four fans
3	12^{15}	19	15	0.30	23.571	0.27	23.143	0.428	With three fans № 1, 2, 3
4	12^{30}	19	15	0.34	24.167	0.31	23.714	0.429	With three fans № 1, 2, 3
5	13^{00}	19	15	0.31	23.714	0.28	23.286	0.428	With three fans № 1, 2, 3
6	13^{45}	20	15	0.24	23.667	0.21	23.167	0.50	With three fans № 1, 2, 3
7	14^{00}	20	15	0.25	23.833	0.22	23.333	0.50	With three fans № 1, 2, 3

4 Discussion

Compared to the regenerative heat recuperator of ventillation air with a massive copper heat exchanger ($\lambda_c \sim 308$ W/(m·K)), the use of the ammonia HP intensifies the transfer of heat.

The heat transfer coefficient for the ammonia aluminum pipe with a grooved core related to the cross-sectional area and to the surface area of the evaporator reaches 1.06×10^5 W/(m^2·K) and 2660 W/(m^2·K).

To obtain the best thermo-technical characteristics of the HP, it is necessary to use such constructions of capillary systems (CS), which will at the same time provide high mass velocity and high level of heat exchange in the heat input and removal zones. At the same time, there are conflicting requirements to the CS: large capillary potential, low hydraulic resistance, high effective heat conductivity.

These conditions are met by the ammonia aluminum HP with a tear-drop grooved core, that combines a round artery with a narrow split of the type AS-KRA7.5-R1 or R2. The heat transfer in the areas of input and removal of heat of the HP is intensified by the use of computer finned aluminum radiators with fans on top, providing the air capacity of up to 125 m^3/h in DSEVS (w ~ 0.5 m/s).

Conclusions

Basing on the mathematical model for the heat pipe operating in biphase thermosyphon mode, the research of heat pipes with different Ω-shaped grooves was carried out [11]. When using various technical means of heat input and removal in Keldysh Center and in the laboratory of Samara State University, the thermo-physical characteristics on trials of the HP made of AS-KRA 7.5-R1-30 section 1762 mm long correlate well with each other. The maximum heat transfer capacity of the HP made of the section AS-KRA7.5-R2 is slightly different from the value (Q_{max}) for the HP made of the section AS-KRA7.0-R2.

This is the effect of the unequally spaced mechanical roughness on the upper surface of a large number of grooves in the core, which affects the formation of steam bubbles, thus intensifying heat exchange. This fact is confirmed by a sharp increase of heat transfer coefficient during evaporation (α_e = 15800 W/(m^2·K) for the HP made of the section AS-KRA7.0-R2.

When testing the HP, large values of heat transfer coefficients during evaporation and condensation were obtained. The revealed self-similarity of the heat transfer coefficient values in the evaporation and condensation zones of the HP with high heat conductivity from the density of heat flow opens up the possibility for the more reliable calculation of effective heat conductivity for structural cores with tear-drop grooved structure, which is significantly higher than for rectangular and trapezoidal grooved cores.

The value of heat transfer resistance of the HP is 0.01-0.02 K/W by maximum transmitted heat flow. The heat transfer coefficient values for evaporation are much higher than the same values for the known analogues of the HP. The HP qualification test program was carried out in strict compliance with the European standard [12]. Mechanical, thermal, radiation and service life tests of the pipes were performed.

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