THE INTENSIFICATION OF HEAT-TRANSFER CHARACTERISTIC OF HEAT PIPES.

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Abstract

The work presents a part of the complex rated experimental research of the heat-transfer characteristic intensification of the heat pipes of medium temperature range. At the butt (axial) direction of heat flow at the inlet into the heat pipe a possibility of the jet nozzle with a specified jet direction appears that intensifies heat transfer under the low heat loads and increases the heat pipe efficiency. The heat tube steam channel realization as a whole as a gas dynamics confuser-diffuser nozzle analogous to the Laval nozzle and surrounded by a capillary porous insertion, layer along the whole length, increases the parameter limit and heat transfer factor of the heat pipe.

KEYWORDS

Heat pipe of the medium temperature range, capillary porous insertion, capillary steam injectors, Laval nozzle, steam flow twist.

INTRODUCTION

In this work results of the rated experimental modeling of the heat pipes of medium temperature range is presented. The problem of the heat-transfer characteristic intensification of the heat pipes of medium temperature range used for cooling of the heat-loaded elements of electronic technique is at present exceptionally urgency.

The heat transfer and operating efficiency of the heat pipes with capillary porous insertions are defined by a closed rotational motion of the working body undergoing to the fluid-steam phase transition with heat liberation in the condensation zone and fluid return through the capillary porous insertions into the evaporation zone of the heat pipe.

At the butt (axial) direction of the heat flow at the inlet into the heat pipe a preferable constructive solution is a flat evaporator made of capillary porous material tightly adjoining to the flat bottom cap.

With small diameters of the heat pipes made of stainless steel the application of the flat caps are possible because of their moderate thermal resistance, simplicity and technological effectiveness of manufacturing. The Fig. 1 shows the heat pipe scheme with the flat evaporator, confuser-diffuser steam channel and turbulator.

The steam phase transfer analysis was carried out by a turbulent gas dynamics model. The steam phase transfer proceeds through the steam channel arranged along the central axis of the capillary porous insertion, which in its turn is tightly set and mechanically fixed in the thin-walled cylindrical body of the heat pipe with top and bottom caps.

The central location of the steam channel, which is made as a gas dynamics confuser-diffuser nozzle, surrounded with a capillary porous insertion layer along the whole length of the heat pipe, allows to realize a considerable reduction of heat loss. Without the capillary porous insertion layer the steam nozzle is becomes supersaturated. As a result of the normal behaviour of the steam jet outflow is broken: in the
flow in the diffuser zones near the critical section a premature large amount of microdrops of the working fluid condensate appears. Simultaneously an important expansion steam jet divergence and fastening of this jet on the walls of the capillary porous insertion takes place that reduces maximum permissible parameters of the heat pipe heat transfer. The condensation jumps being formed in the extending part of the nozzle is followed by heat evolution and under the influence of two factors (heat and geometrical once) loose stability.

The variable section of the capillary porous insertion with a maximum thickness value near the critical section of the nozzle and a decreasing thickness in the evaporation and condensation zone protects the steam flow from supercooling in such a manner, that the maximum condensation takes place on the inner surface of the heat pipe top cap, at the steam jet braking temperature exceeding the condensation temperature.

The compression ratios and degrees of expansion of the gas dynamics confuser-diffuser nozzle are selected the same as by the Laval nozzle, the confuser part estimated profile is performed concave in such a manner, that the reflected steam jets converge to the critical section center of nozzle.

Overheating \( \Delta T \) of the steam flow on going out of the nozzle depended on the temperature exceeding against the working fluid boiling point in the heat pipe evaporator is rated by means of introduction of the heat condensation effective value in accordance with the formula [1]:

\[
\text{\( r_{\text{eff}} = r + C_p^{\prime} \cdot \Delta T \) }
\]

where: \( r_{\text{eff}} \) - heat condensation effective value, J/kg; \( r \) - heat condensation value, J/kg;
\( C_p^{\prime} \) - steam heat capacity, J/kg·K; \( \Delta T \) - difference of the overheated steam temperatures, K, and its saturation temperature at given pressure.

The overheating value \( \Delta T \) under high heat loads on the heat pipe reaches 50K.
In the preliminary experiments with model equipment it is determined that the condensation intensity of the broadening steam jet on the perpendicular surface of heat exchange is proportional to the steam flow forticity.

The motion type of the condensate microdrops in the two-phase dispersed flow is complicated, drops can be accelerated get additional energy owing to what not only coagulation but also crushing of drops into still smaller ones [2,3]. The increase of condensation nucleus amount of supersaturated steam promotes the steam condensation intensification both on the heat exchange surface and on the drops.

The heat-mass exchange within heat pipes is determined not only by hydraulic and gas dynamics features of steam and condensate motion, but by interaction with solid body surfaces as well in order to increase the heat transfer coefficient from the steam flow to the inner surface of the top cap and the intraphase mass exchange of the condensate film, the heat pipe top cap is made with a taper turbulator.

The turbulator surface, and the top cap inner surface are covered with elongated grooves [4-6], the longitudinal axes of which in the azimuthal plane are turned concerning the longitudinal axis of the steam channel at specified angle, which provides maximum twist of the steam wet flow with condensate microdrops.

The twist value effects on the thermal resistance of the layer adjacing to the walls including a viscous sublayer.

For the steam generation intensification in the evaporation zone under low heat loads some injecting steam outlet channels are used piercing the bottom flat insertion capillary porous insertion tightly adjoining the flat bottom cap of the heat pipe.

The injection channel diameter reaches 1 mm, and reduced capillary pressure results in temperature lowering of boiling and steam generation of the working fluid in these channels that was found as very important under low heat loads of the heat pipe.

Fig.2 shows the calculation result of velocity distribution at the steam injecting channels over the bottom flat insertion of the capillary porous insertion under low heat load.

The injectors are especially effective with small heat flows and accordingly with low initial velocities of the steam flow over the evaporator.

Fig. 3 shows the same calculation result, but under high heat load on the evaporation butt surface of the heat pipe, when operation effects of the jet injectors virtually become not apparent and steam generation becomes uniform on the whole section of the nozzle confuser part.

The injecting capillary channels are made as Laval nozzles that increases the fluid working body supply to them on the capillary porous insertion layer along the bottom, the inner surface of which is additionally covered with radial and circular grooves. The groove profile is selected as a dihedral angle.

Realization of the steam channel of the heat pipe wholly as a gas dynamics confuser nozzle analogous to the Laval nozzle and surrounded with a capillary porous insertion layer results in the flow velocity increase of the steam turbulent jet.

To illustrate the application effectiveness of the confuser-diffuser nozzle as a steam channel of the heat pipe, Fig. 4 shows differences of velocity, m/s, steam jet flow in the steam channels of two structures – as a confuser-diffuser nozzle and in a standard cylindrical channel, in both cases without use of the injecting capillary steam channel.

The steam jet velocities were calculated at a height of the nozzle critical section, the nozzle diameters in the broad part and of the cylindrical steam channel are the same.

The axial heat flow and initial steam velocities over the evaporators are the same as well.

The experimental definition of the steam flow velocity of C₂H₅O diethyl ether was carried out in the nozzle critical cross-section using the hot-wire anemometer with a heated filament according to the standard methods [7].

The gold-iron wire diameter is 5 μm, specific electric resistance is 2.8·10⁻⁸ Ω·m, resistance temperature coefficient is 3.9·10⁻³ 1/K. The boiling temperature of diethyl ether is 35.4°C ≈ 308.55K under atmospheric pressure.

The sensor with one wire filament of 2 mm length is vacuum-tightly put into the steam channel, the filament is oriented perpendicularly to the longitudinal axis in the central zone of the nozzle critical cross-section.
Fig. 2. Steam velocity distribution, m/s, over the steam injecting channels in the evaporation part of the heat pipe under low axial heat load.

Fig. 3. Steam velocity distribution, m/s, over the steam injecting channels above the evaporator of the heat pipe under high axial heat load.
The preliminary calibration of the hot-wire anemometer was carried out on a flow meter washer and standard critical nozzle, the steam pressure drop was defined using the pressure difference converter SAPPHIRE-22 DD in the steam channel. The membrane unit of SAPPHIRE-22 DD and inlet lines when operating were at the 35°C temperature.

The hot-wire anemometer measured the average in the time longitudinal component of velocity \( u \) of the free steam flow with filament superheating coefficient 1.8, the calibration proper was carried out using the modified King equation [7]:

\[
    u = k_1 \cdot \frac{E^2}{E_0^{1/n}} + k_2 \cdot \frac{E}{E_0^{1/2}}
\]

where: \( E \) and \( E_0 \) - output voltages of the hot-wire anemometer by the steam flow velocity and in the absence of it, \( k_1 \), \( k_2 \) and \( n \) - are constants.

The exponent \( n \) is near the value 0.5, the second constant \( k_2 \) takes into account a free convection on the wall at very low velocities of the steam flow.

The maximum error by calibration of the hot-wire anemometer did not exceed 2-3% of the velocity value \( u \).

The signal from the hot-wire anemometer filament is intensified tenfold and supplied to the input of the 12 bit-analog-to digital converter ADS and further into computer, where the measurement data are stored in the memory and then subjected to the program processing.

The measurement error of the average according to time value of the steam flow velocity longitudinal component by means of the hot-wire anemometer does not exceed ± 0.3 sm/s.

The measurement results of the steam flow velocity in the nozzle critical cross-section depending on thermal head \( \delta T \) on the external surface of the evaporator, \( T_B \) - temperature of boiling, K:

\[
    \delta T = T - T_B
\]

are adduced on the Fig. 5.

The black points depict the measurement results of steam velocity in the critical cross-section of the confuser-diffuser nozzle, the light ones depict the measurement results in the standard cylindrical channel of the heat pipe.
Fig. 5. The comparison of the velocities, sm/s, steam jet flow in the nozzle critical section and in the standard cylindrical steam channel depending on the temperature head in the evaporator.

Here the calculated approximating values that are the full line is nozzle, the curved dotted one is the cylindrical channel.

Reynolds number is \( \text{Re} = 0.06 \), Prandtl number is \( \text{Pr} = 0.77 \).

Fig. 6. Calculation result, pressure difference Pa, in the critical section of the steam channel as a confuser-diffuser nozzle and a standard cylindrical steam channel, depending on the initial steam velocity over the evaporators.
The steam channel made as a confuser-diffuser nozzle possesses a distance action that results in the velocity exceeding of the steam flow in comparison with the standard cylindrical steam channel in the frontal point of the turbulator when come nearer to the heat pipe condensation zone.

A complex nature of the pressure behaviour in the frontal point of the turbulator, depending on the initial velocity of the steam flow over the evaporators \( \text{Reynolds numbers Re of evaporators} \), shows a presence of some modes of the steam flow in this point, depending on the heat load on the evaporator and the heat pipe. This fact means a necessity of some supplementing each other arrangements of the condensation intensification on the turbulator including the steam flow twist.

One can see that the nozzle is found as a very effective accelerator of the steam flow under low axial heat loads. The pressure differences, \( \text{Pa} \), of the confuser-diffuser steam channel and standard cylindrical steam channel calculated on the height of the nozzle critical section is shown in the Fig. 6.

**Fig. 7.** Calculation result, velocity differences \( \text{m/s} \), of the steam flow in the turbulator frontal point in the confuser-diffuser steam channel and in the standard cylindrical steam channel, depending on the initial steam velocity over the evaporators.
Fig. 8. Calculation result, pressure difference Pa, in the turbulator frontal point in the confuser-diffuser steam channel and in the standard cylindrical steam channel, depending on the initial steam velocity over the evaporators.

An additional dynamic pressure of the condensate at the input into the top edge of the capillary porous insertion enlarges the limiting parameters of the heat pipes.

The condensate return is realized by the capillary porous insertion, the transient characteristics of which are expanded by introducing the longitudinal microcapillary channels passing along the whole length of the pipe, the average diameter of which doesn't exceed 10 µm. The layers of the sintered granulated material adjoining the microcapillary channels form the additional porous channels passing along the whole length of the capillary porous insertion as well. The estimated evaluations of the heat mass transfer of the heat pipe carried out with programs CF Design (Blue Ridge Inc., USA) and ANSYS and also model analytic evaluations show increase of the heat - transfer factor of the heat pipe of the selected design.

References