HEAT AND MASS TRANSFER IN LOW-TEMPERATURE HEAT EXCHANGER PIPES

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Results are shown of a study concerning a heat exchanger pipe with Freon as the active medium. A relation has been established which describes the effect of the coolant temperature and of the pipe inclination on the heat transfer capacity of such a pipe.

In high-temperature heat exchange piping known today, with liquid metals used as the heat carrier, the axial temperature drop during operation is generally small and of the order of a few degrees only [4, 5]. The heat transfer in the condensation zone here is usually radiative. In low-temperature piping with water, alcohol, ammonia, or any other low-conductivity high-viscosity cryogenic liquid used as the heat carrier the temperature drop in the condensation zone is often as high as a few tens of degrees. The reason for this is that the heat is removed from low-temperature piping either by convection while the cold liquid circulates through the condenser or by a phase transition (evaporation, fusion, sublimation, etc.), i.e., the thermal resistance at the outside condenser surface of low-temperature piping can be comparable to or lower than the thermal resistance of the phase transition zone in the vapor–liquid–wick (the wick saturated with liquid) system. The conditions of an isothermal operation of heat exchanger piping depend on the model of heat transfer at the outside condenser surface. In [1-3] the temperature field was measured at the outside surface of heat exchanger pipes with low-boiling liquids as the heat carriers. Here the condenser surface on the heat exchanger side was immersed directly in the cooling liquid and, consequently, the temperature drop across the surface of the pipes could be taken as approximately equal to:

$$\Delta T = \frac{T_s - T_x}{L},$$

with L denoting the length of the thermally insulated zone separating the cooler and the evaporator.

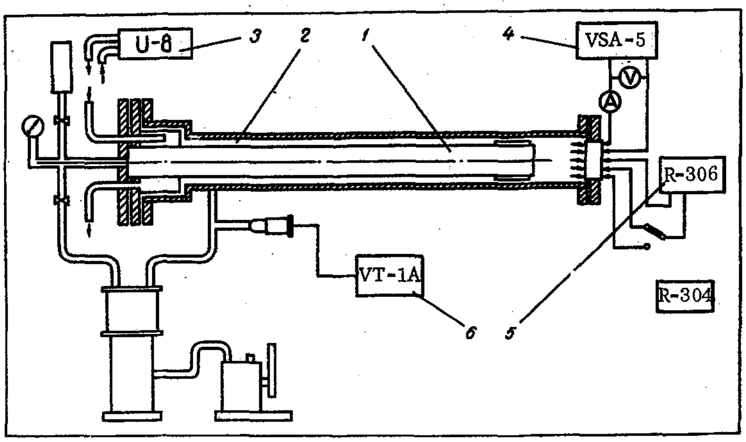

Fig. 1. Schematic diagram of the test apparatus: 1) heat exchanger pipe; 2) vacuum insulation; 3) U-8 thermostat; 4) VSA-5 power supply; 5) R-306 potentiometer; 6) VT-1A vacuumeter.

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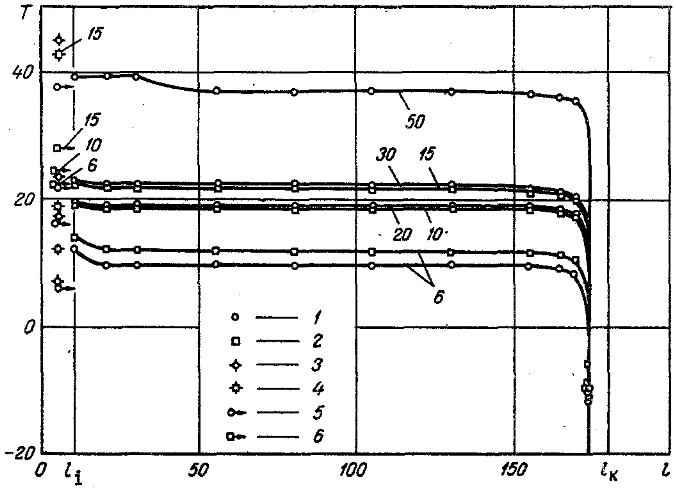

Fig. 2. Temperature distribution (°C) along the pipe (cm) with Freon-22: 1) $\alpha = +5°$; 2) 0°; 3, 4) temperature of the liquid inside the wick within the evaporation zone; 5, 6) temperature of the vapor within the evaporation zone. Numbers at the curves and the test points indicate the power, W.

Experiments had shown, however, that the temperature of those heat exchanger pipes was almost constant along most of the distance between evaporator and condenser but changed rapidly at the condenser entrance section. Thus, it is here where most of the temperature drop occurred.

The purpose of our study was to establish the peculiarities of the temperature distribution in the wick of a heat exchanger pipe equipped with a liquid cooler in the condensation zone. Freon-22 and Freon-11, both low-conductivity and high-viscosity liquids, were used as the heat carrier.

The pipe was designed to the following specifications: length 1.8 m, outside diameter 19.5 mm, wall thickness 0.25 mm, material stainless steel. Inside the pipe was placed a wick of glass wool 3.5 mm thick. This wick had the following characteristics: minimum radius of meniscus $R_{min} = 4 \cdot 10^{-5}$ m, hydraulic drag coefficient $K_1 = 0.25 \cdot 10^{11}$ m^2. The temperature was checked with copper–constantan differential thermocouples pasted to the pipe surface and with copper resistance thermometers, the latter placed on the pipe surface and inside the wick within the evaporation as well as the condensation zone. Furthermore, the vapor temperature within the evaporation zone was also measured with a thermistor. In order to eliminate any convective heat transfer between the pipe surface and the ambient medium, the pipe was placed inside an insulating vacuum layer. The rarefaction inside this vacuum jacket was maintained at 10^{-4} mm Hg.

An overall view of the test apparatus is shown in Fig. 1. The heat transfer capacity of the pipe with various working liquids was calculated by the Cotter formula [4]:

$$Q = \frac{\pi\sigma\rho_L (d_W^2 - d^2) r'}{2\mu_L R_{min} l K_1}. \tag{1}$$

Water and ammonia were found to have the maximum heat transfer capacity, while the Freons were found to transfer the smallest quantities of heat. Ammonia has several drawbacks, however, which make its use problematic. It is toxic and requires special equipment, it also reacts with certain metals. Water is unsuitable at temperatures below zero. We chose Freon-11 and Freon-22 as the heat carriers in this experiment. Formula (1) yielded the following values for the maximum transmitted thermal power at 30°C: $Q_{max} = 1.75$ W for Freon-11 and $Q_{max} = 1.26$ W for Freon-22. These values referred to the temperature of the insulation zone. It ought to be noted that the maximum transmitted power increased as the operating temperature of the pipe dropped. Depending on that temperature, the thermal flux radiated to the pipe surface varied from 4 to 1.5 W.

The temperature distributions along the pipe are shown in Fig. 2 for various levels of transmitted power and for various angles of pipe inclination. The working liquid here was Freon-22. A temperature of −25°C was maintained inside the condenser. The temperature drop near the condenser governed, almost exclusively, the entire temperature drop along the transport zone.

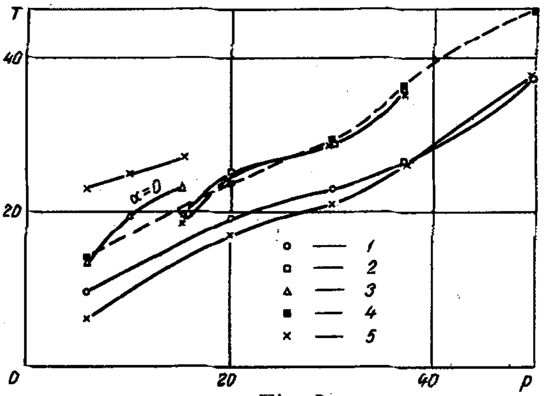

Fig. 3

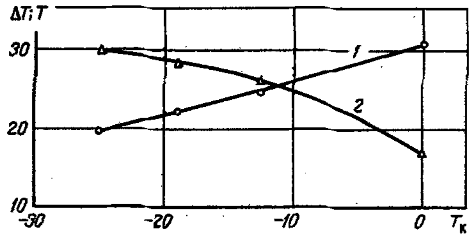

Fig. 4

Fig. 3. Operating temperature (°C) as a function of the power (W): 1) $\alpha = +5°$ and Freon-22 as the working liquid; 2) $\alpha = +2°$ and Freon-22; 3) $\alpha = 0$ and Freon-22; 4) $\alpha = 0°$ and Freon-11; 5) temperature of the vapor within the evaporation zone.

Fig. 4. Temperature drop along the transport zone and operating temperature of the pipe as functions of the condenser temperature, °C (working liquid Freon-22 and $\alpha = +5°$): 1) operating temperature of the pipe; 2) temperature drop along the transport zone.

Empirical curves which relate the operating temperature of the pipe to the supplied power are shown in Fig. 3 for Freon-11 and Freon-22 in the pipe respectively. The maximum power transmitted by the pipe in a horizontal position was 15 W, i.e., almost 10 times higher than according to the Cotter formula.

The vapor temperature and the temperatures measured at several points on the pipe surface were all close with the pipe inclined at a positive angle.

When the pipe operated in a horizontal position ($\alpha = 0$), the vapor of the working liquid superheated considerably above the wall temperature. This excess was as high as 10°C.

The operating temperature of the pipe and the temperature drop along the transport zone are shown in Fig. 4 as functions of the coolant temperature in the condensation zone of the pipe, with the latter inclined positively and transmitting a power of 20 W. By raising the condenser temperature, we succeeded in increasing the effective thermal conductivity of the pipe by almost a factor of 2 and, as a result, in appreciably increasing the temperature of the pipe surface. It was impossible to raise the condenser surface higher still, because of the excessive operating pressure.

The thermophysical properties of Freon-11 resemble those of Freon-22. This explains why the temperature-power characteristic of the pipe was about the same whether the latter carried Freon-11 or Freon-22 (Fig. 3). These curves were plotted for the same positive inclination angles and the same condenser temperatures. The operating temperatures of the pipe in both cases differed by approximately 10°C.

It must be taken into account, however, that a replacement of Freon-22 with Freon-11 means also a drop in the operating temperature from 15 atm to 1 atm, which improves the reliability and the safety of the process.

On the basis of the performed tests, one may draw the following conclusion. When low-conductivity liquids are used as the heat carrier in heat exchanger piping, then an appreciable temperature drop develops in the condensation zone of such piping. This is true when the heat-transfer coefficient at the outside pipe surface within the condensation zone is high.

As the temperature in the heat exchanger becomes lower, the temperature drop in the condenser increases and the performance of the piping deteriorates. Design formulas given in the literature for calculating the maximum thermal power transmittable by a low-temperature heat exchanger pipe yield values quite different from those obtained in tests.

Heat exchanger piping with Freons used as the heat carrier are recommended for cooling radio-electronic apparatus where the thermal flux densities are relatively low. Freons are chemically inert and do not react with either the pipe material or the apparatus material. After a comparison of the pipe performance with Freon-22 and with Freon-11, preference should be given to Freon-11 from the safety standpoint.

The heat transfer capacity can be increased considerably by modifying the wick characteristics and by shortening the pipe.

NOTATION

Q is the thermal flux;
W is the flow rate of liquid or vapor;
h is the enthalpy;
q is the thermal flux density;
u is the velocity;
σ is the surface tension;
R is the radius of liquid meniscus;
ε is the porosity;
K_1 is the hydraulic drag coefficient;
μ is the dynamic viscosity;
ρ is the density;
r' is the latent heat of evaporation;
x is the pipe length coordinate.

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