STUDY OF THE HEAT TRANSFER UNDER LARGE TEMPERATURE DROPS DURING COOLING WITH A STREAM OF LIQUID NITROGEN MIST

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UDC 536.24

Results are shown of feasibility studies concerned with increasing the coefficient of heat transfer to liquid nitrogen under large temperature drops.

In many cryogenic engineering devices the cooling is effected during film boiling of the liquid, when the heat-transfer coefficient is low. The authors undertook to establish the feasibility of increasing the heat-transfer coefficient under temperature drops corresponding to steady film boiling, by cooling a surface with controlled streams of liquid mist.

The test apparatus is shown schematically in Fig. 1. It consisted of a helical coil 1, 50 mm in diameter, made of 6 mm (diam.) × 1.0 mm (thick) copper tubing. Hydrogen at room temperature was passed through it under a pressure of 100 atm. The flow rate could be varied by shunting some gas through valve 14 into a receptacle. The hydrogen inlet and outlet temperatures were measured with copper–constantan thermocouples 4, 5. The outside surface of the helix as well as the contact areas between the thermocouple junctions and the tube were thermally insulated with glass fiber 6 and BF-4 glue added on the outside. Inside the helix was placed a liquid-nitrogen sprayer 2 in the form of a tube with 0.7 mm diameter holes. The helix was placed into a container above the nitrogen level. A centrifugal pump 3 with an approximately 35 liters/min capacity and a 1.5 atm maximum pressure fed liquid nitrogen into the sprayer. The pressure in the sprayer was varied from 0.25 to 1.1 atm by shunting some nitrogen through valve 16. The pressure was measured with manometer 17. Unevaporated nitrogen was drained into the container and then returned to the pump. The number of holes in the sprayer tube was varied from test to test between 16 and 150. The rate of gas flow through the heat exchanger was checked on the basis of the hydraulic drag along the calibrated tube segment after throttling to atmospheric pressure and heating in the heat exchanger 11 to a 25°C temperature. At least 50 m^3/h of hydrogen was pumped through. At that minimum rate the temperature along the entire heat exchanger stabilized above the film boiling point of nitrogen.

The heat-transfer coefficient k was measured as a function of the nitrogen trickle-jet pressure, of the number of jets, and of the temperature difference. For comparison, we also measured the coefficient k_0 of heat transfer during simple immersion of this heat exchanger into liquid nitrogen. Within the range of stable film boiling $k_0 \approx 125$ kcal/$m^2 \cdot h \cdot$deg. The heat-transfer coefficients k and k_0 were calculated by the Newton formula [1]

$$Q = kF\Delta T. \tag{1}$$

The quantity of heat Q here was equal to the difference between the heat content in hydrogen at the inlet to and at the outlet from the coil. It was determined from the T–S diagram [3]. Since in our test the temperature varied along the entire heat exchanger, as the temperature drop ΔT in Eq. (1) we used the logarithmic mean temperature difference [1]:

$$\Delta T = \frac{\Delta T_i - \Delta T_e}{\ln \dfrac{\Delta T_i}{\Delta T_e}}.$$

Institute of Engineering Physics, Academy of Sciences of the Ukrainian SSR, Khar'kov. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 23, No. 2, pp. 207-211, August, 1972. Original article submitted November 3, 1971.

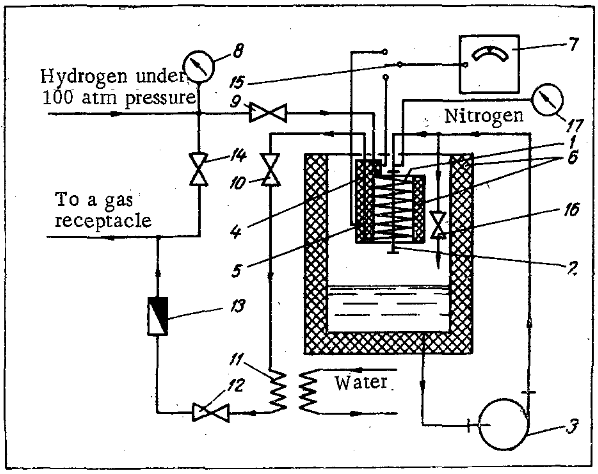

Fig. 1. Schematic diagram of the test apparatus: 1) helical coil; 2) liquid-nitrogen sprayer; 3) centrifugal pump; 4, 5) thermocouples; 6) glass fiber; 7) potentiometer; 8, 17) manometers; 9, 10, 12, 14, 16) valves; 11) heat exchanger; 13) flow meter; 15) switch.

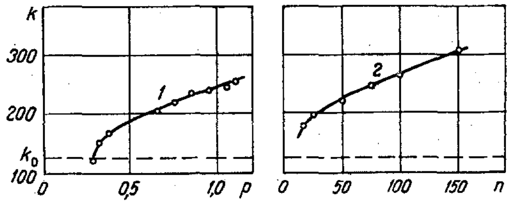

Fig. 2. Heat-transfer coefficient k (kcal / $m^2 \cdot h \cdot deg$) as a function of the nitrogen jet supply pressure (1) with n = 50 and as a function of the number of jets n (2) at p = 0.55 atm.

The temperature drops at the inlet ΔT_i and at the outlet ΔT_e were found by measuring the surface temperature of the heat exchanger, while the temperature of the cooling liquid was assumed equal to the boiling point of nitrogen under atmospheric pressure. The error of the temperature determination in these tests did not exceed 2.5%.

In Fig. 2 is shown the heat-coefficient k as a function of the nitrogen pressure p at the sprayer inlet (curve 1). It seems logical to assume that the higher heat dissipation rate under higher pressure was due to a higher heat-transfer coefficient at places of direct contact with the nitrogen mist jets. The effective cooling surface of a single jet was determined experimentally. The nitrogen jet from a 0.7 mm diameter hole was aimed at a given surface of a nickel silver box under conditions analogous to those of the trickle-sprayed heat exchanger. The water stream through the box was adjusted so as to minimize the ice area on the back wall. The size of this area was estimated visually through a transparent front wall and assumed 7 mm^2.

An additional test was performed, in order to prove directly that a higher coefficient k results from a sharp increase in the heat-transfer rate at places of contact with the jets. The test will be understood from Fig. 3. A copper foil (1) 0.5 mm thick and 20 cm^2 in area was heated by radiation from an electric furnace 3 and cooled by freely boiling nitrogen. Furthermore, a nitrogen jet was aimed at the foil from a hole 3 mm in diameter. With the aid of a calibrated contact thermocouple 2, a sharp-edge 0.5 mm diam. constantan conductor (the other conductor was the foil), the temperature was measured inside and outside the jet zone. The insert temperature was found to be about 70°C lower inside than outside. This indicated more heat dissipation at places directly trickle-sprayed by the jets. That this was not a boundary effect could be demonstrated as follows. The surface cooled with liquid nitrogen was not sprayed with the nitrogen jet. The temperature was measured inside as well as outside the jet zone. The temperature was found not to vary from place to place (nearer to or farther from the foil edge) but was found to possibly fluctuate with time at any given location. These fluctuations were apparently due to the instability of the vapor film at such heights of the liquid column (≈20 mm) and its discontinuity in some places. The readings of the contact thermocouple did not depend on the pressure on the constantan thermoelectrode. Its accuracy of temperature measurement was the same as that of conventional copper–constantan thermocouples.

No increase in the heat-transfer coefficient due to the dripping of liquid down the helix was noted. This was verified on an apparatus with a heat exchanger in the form of an identical tube but not thermally insulated and placed at the edge of a container 800 mm in diameter. The heat-transfer coefficient was measured while the container remained stationary and while the container revolved. While the container revolved, the velocity of the oncoming nitrogen stream was 1 m/sec. No difference between the coefficients of heat transfer during film boiling in both cases was noted.

The heat-transfer coefficient as a function of the number of jets is shown in Fig. 2 (curve 2). With the number of holes increased above 25, one notes a proportionally increasing heat-transfer coefficient. The sharp decrease of k at n = 25 is explained by liquid nitrogen covering the entire heat-exchange surface.

In Fig. 4 the heat-transfer coefficient k is shown as a function of the temperature drop ΔT with n = 150 (curve 1) and with n = 75 (curve 2). It is well known [2] that the heat-transfer coefficient remains constant during film boiling of nitrogen into a large volume. In our case it remained constant only up to a certain magnitude of the temperature drop, above which it decreased.

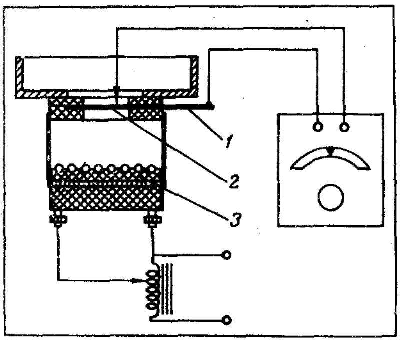

Fig. 3

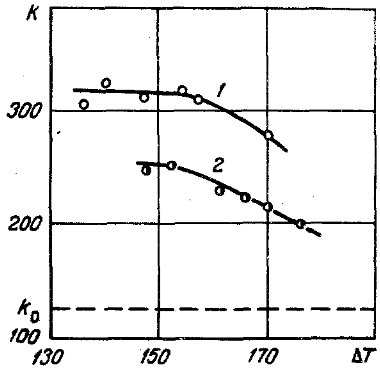

Fig. 4

Fig. 3. Schematic diagram of the additional test: 1) copper foil; 2) contact thermocouple; 3) electric furnace.

Fig. 4. Heat-transfer coefficient k (kcal/$m^2 \cdot h \cdot deg$) as a function of the temperature drop ΔT (°C): p = 0.55 atm and n = 150 (1); p = 0.55 atm and n = 75 (2).

If one assumes that the entire increase in the heat-transfer coefficient during cooling with mist jets occurs at the contact areas with the jets, while at the rest of the surface it remains equal to k_0, then a calculation according to the formula $k_1 = (k-k_0)(F/F_1)$ yields $k_1 \approx 8500$ kcal/$m^2 \cdot h \cdot deg$ at p = 0.55 atm and with n = 150. The logarithmic mean temperature difference has been assumed the same over the entire heat-exchanger surface. As the number of holes is decreased, the heat-transfer coefficient calculated in this manner becomes higher (k_1 = 11,000 with n = 75). This increase can be explained by the lesser effect of unevaporated nitrogen dripping down the heat-exchanger surface.

Such a large boost of the heat transfer at places of contact with the jets can be explained by the presence of droplets in the liquid mist stream which are capable of evaporating, when in contact with the surface, without formation of a vapor film. The size of such droplets may quite possibly vary depending on the temperature difference. An explanation for the character of the relation between heat-transfer coefficient and temperature drop (Fig. 4) is, then, that most droplets formed in the jet under a 0.55 atm pressure cease to wet the surface as the temperature drop increases.

NOTATION

k is the measured heat-transfer coefficient;
k_0 is the coefficient of heat transfer during film boiling into a large volume;
k_1 is the heat-transfer coefficient at an effective cooling area;
F_1 is the effective cooling area;
F is the heat-exchanger surface area;
n is the number of holes in the sprayer tube;
p is the nitrogen-jet supply pressure;
Q is the quantity of heat;
ΔT is the logarithmic mean temperature difference;
ΔT_i is the temperature drop at the heat-exchanger inlet;
ΔT_e is the temperature drop at the heat-exchanger outlet.

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