HEAT TRANSFER DURING THE COOLING OF HUMID UNSATURATED VAPOR–GAS MIXTURES

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UDC 536.248.2

It has been experimentally verified that the heat transfer during cooling of unsaturated gases can be calculated by the equation for saturated mixtures with a correction factor Z characterizing the departure of a mixture from the saturated state.

During the cooling of saturated mixtures, the quantities of heat transferred from the gas and from the vapor are in a definite ratio which depends only on the mixture temperature, and it is thus possible to

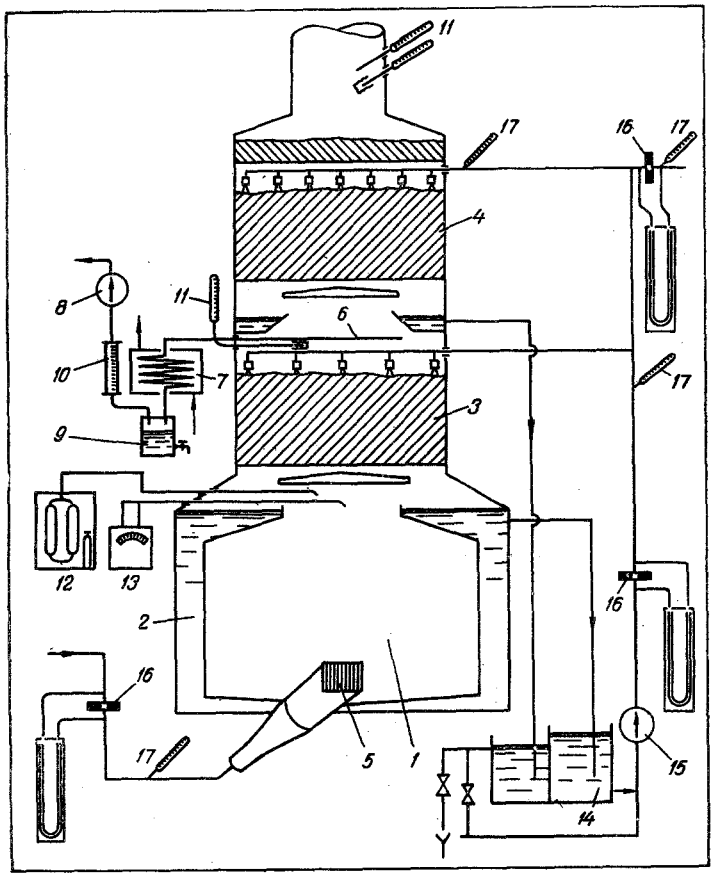

Fig. 1. Schematic diagram of the test apparatus: 1) furnace; 2) water carrying jacket; 3) humidification zone; 4) condensation zone; 5) burner; 6) gas exhaust pipe; 7) cooler coil; 8) vacuum pump; 9) burette; 10) rotameter; 11) psychromoters; 12) gas analyzer; 13) potentiometer; 14) water tank; 15) pump; 16) diaphragms; 17) thermometers.

All-Union Correspondence Polytechnic Institute, Moscow. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 22, No. 1, pp. 19-24, January, 1972. Original article submitted June 8, 1970.

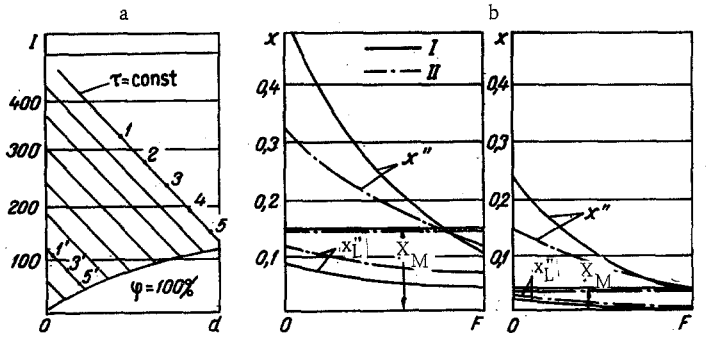

Fig. 2. Plotting the gas and the water parameters in the process of testing, at points 1-5 of a series: points 1, 1' (I), points 5, 5' (II); I (kcal/m^3), x (kg/m^3), d(kg/kg), F (m^2).

calculate the process with only one – any one – driving action: heat transfer or mass transfer [1]. When a mixture departs from the saturated state, this ratio changes and this requires an appropriate correction. In order to be still able to calculate an unsaturated mixture with only one driving action, it has been suggested in [1] that the unsaturated mixture be reduced to an equivalent (with respect to enthalpy) saturated state by introducing into the calculation formula a correction factor

$$Z = \frac{x'' - x_G}{x'' - x''_L},$$

which would account for the departure of the humidity in a mixture from the saturation humidity.

With this correction factor, the equation of heat transfer becomes

$$Nu^* = Nu \frac{1 + \xi X_M^m}{1 + \xi X_M^m Z^p}; \quad (1)$$

When Z = 0, corresponding to the saturated state, Eq. (1) becomes $Nu^* = Nu(1 + \xi X_M^m)$; as Z departs from zero, the denominator increases and, when Z = 1 (absence of condensation), Eq. (1) becomes the equation of pure heat transfer, i.e., $Nu^* = Nu$.

The feasibility of calculating the heat transfer process for saturated mixtures by this method has been demonstrated in [2]. In this article we will attempt, on the basis of experimental data, to apply this method to an unsaturated mixture.

The test apparatus (Fig. 1) consisted of a gas furnace 1 inside a water carrying jacket 2 and a contact-type heat exchanger split into a lower part (humidification zone 3) and an upper part (condensation zone 4). The burner 5 burns natural gas whose combustion products are cooled in the furnace down to t = 700-1000°C.

In this apparatus we could vary the temperature of the gases at the furnace outlet by adding air through special vents in the furnace and its exhaust duct.

The hot gases were passed into zone 3, where they were humidified to the necessary level. For humidification we used hot water at the wet bulb temperature, which was fed through a sprinkler set above zone 3. In earlier experiments [2] the gases at the exit from zone 3 became humidified almost to a state of saturation, when the water flow rate was sufficient for a complete wetting of the furnace hood ($L > 15 \cdot 10^3$ kg/m$^3 \cdot$h). For this reason, in order to produce an unsaturated mixture in those tests, the sprinkling intensity was regulated within 12,000 to 2,000 kg/m$^2 \cdot$h, which resulted in an incomplete wetting of the furnace hood and a lower humidity of the gases.

The gases, after having been humidified in zone 3, were passed into the condensation zone 4, while cooling water was fed into the upper part of this zone. The active volume of zones 3 and 4 was filled with a pile of ceramic Raschig rings 25 × 25 × 3 mm in size. The quantity of cold water was maintained constant so as to ensure a dew density $L = 5 \cdot 10^3$ kg/m$^2 \cdot$h on the furnace hood.

The hot water from zone 4 was drained off. The temperature and the composition of gases at the exit from the furnace was checked in these tests with a platinum – rhodium/platinum thermocouple and a

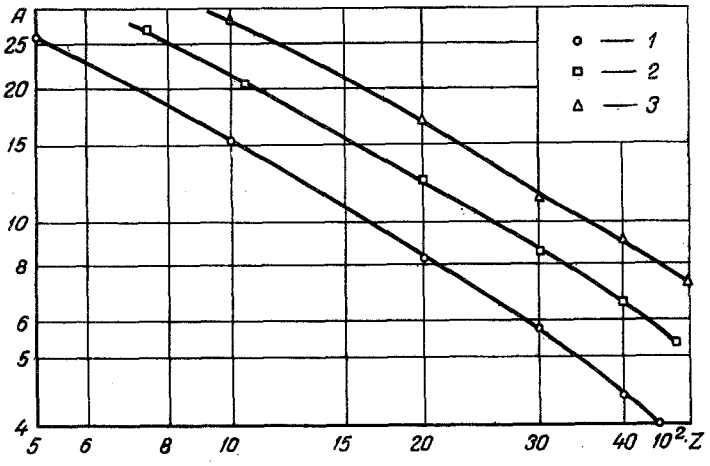

Fig. 3. Group $A = 10^2/(1 + \xi X_M^{0.55} Z)$ as a function of the parameter Z: $X_M = 0.25$ (1), 0.14 (2), and 0.065 (3).

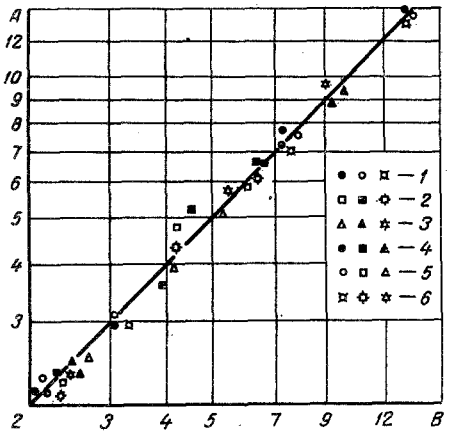

Fig. 4. Generalized curves of $A = Nu^*/Re_G^{0.52} Pr_{0.55}^{0.33}$ as a function of the group $B = (1 + \xi X_M^{0.55})/(1 + \xi X_M^{0.55} Z)$: 1) $X_M = 0.25$; 2) 0.14; 3) 0.065; 4) $Re_G = 500$; 5) 350; 6) 200.

gas analyzer. The humidity of the gases at the exit from the humidification zone was determined by two methods: psychrometrically with mercury thermometers 11 and by directly measuring the weight content of the gases. For this purpose, a gas sample was sucked out along pipe 6 and the cooler coil 7 by means of the vacuum pump 8. The condensate precipitating in the coil was collected in the burette 9, while the dry gas entered the rotameter 10.

In order to establish how the rate of heat transfer is affected by the departure of a mixture from saturation, we performed four series of tests, each yielding a number of points on the same adiabatic saturation line (τ = const) on the I–d diagram (Fig. 2a). Each series of tests was performed at a different initial temperature of the gases and correspondingly different mean humidity levels X_M. Within each test series the value of X_M was maintained constant. What made the tests difficult was that, with all other conditions the same, both the parameter Z and the value of X_M varied from point to point in the series. For this reason, we artificially adjusted the temperature of the cooling water so that X_M should remain constant during a change of Z. This was achieved by raising the water temperature during a decrease in the initial humidity of the mixture (Fig. 2b).

The parameters were varied in these tests within the following ranges: temperature of the gases from 105 to 650°C, temperature of the cooling water from 5 to 41°C, humidity of the gases from 0.05 to 0.68 kg/m^3, velocity of the gases from 0.2 to 1.1 m/sec, mean humidity of the boundary layer from 0.06 to 0.25 kg/m^3, Reynolds number from 160 to 550, and parameter Z from 0.05 to 0.53.

In Fig. 3 is shown the parameter

$$N = \frac{1}{1 + \xi X_M^{0.55} Z}$$

as a function of Z, characterizing the change in the heat transfer as the humidity of the gases departs from the saturation level, at constant Re_G and Pr numbers for three different values of X_M.

From the expression

$$N = \frac{1}{1 + \xi X_M^m Z^p}$$

follows

$$p = \ln(1-N) - \ln(N\xi X_M^m) - \ln Z.$$

Assuming, according to [2], that m = 0.55 and evaluating the data in Fig. 3, we obtain p ≈ 1, from which it follows that the cooling of an unsaturated vapor–gas mixture may be treated as the cooling of an equivalent (with respect to enthalpy) saturated mixture by inserting into the calculation formula the correction factor

$$1/1+\xi X_M^{0.55} Z.$$

Although the manner in which the Re_G number affects the heat transfer in the case where humid gases are cooled had already been revealed in [2], series of tests were performed in our study for further verifying this effect. It has been established here that, with all other conditions the same, $Nu = CRe_G^{0.55}$. Considering the slight deviation of the exponent of Re_G from its value in [2], a similar value was assumed in the overall evaluation of the test data.

The generalized curves in Fig. 4 represent Nu* as a function of the group $(1 + X_M^{0.55})/(1 + X_M^{0.55}Z)$ and they correspond to different values of X_m. They satisfy the equation

$$Nu^* = 0.102 Re_G^{0.52} Pr^{0.33} \frac{1+\xi X_M^{0.55}}{1+\xi X_M^{0.55} Z}. \quad (2)$$

NOTATION

a	is the thermal diffusivity, m/h;
d_e	is the equivalent diameter of the furnace hood, m;
G_w	is the flow rate of sprinkling water, 10^3 kg/h;
L	is the sprinkling intensity, 10^3 kg/m^2·h;
t''_m, t_{hum}, t_{wb}	are the temperature of the gases at the entrance to and at the exit from the humidification zone, and wet bulb temperature, °C;
w	is the velocity of the gases, m/sec;
x''	is the humidity of an equivalent saturated mixture, kg/m^3;
x_G	is the humidity of the gases, kg/m^3;
x''_L	is the saturation humidity at the temperature of the cooling water, kg/m^3;
$X_M(x''-x''_L)/\ln(x''/x''_L)$	is the logarithmic mean between the humidity of the equivalent saturated mixture and the saturation humidity at the water temperature, kg/m^3;
$Z = (x''-x_G)/(x''-x''_L)$	is the parameter characterizing the departure from saturation;
α	is the heat-transfer coefficient, kcal/m^2·°C·h;
α^*	is the modified heat-transfer coefficient, including the heat transfer due to condensation, kcal/m^2·°C·h;
ξ	is the coefficient depending on the temperature of the saturated mixture (at t = 30°C, ξ = 93; at t = 80°C, ξ = 150);
λ	is the thermal conductivity, kcal/m·°C·h;
ν	is the viscosity coefficient, m^2/h;
$Nu = \alpha d_e/\lambda$	is the Nusselt number for heat transfer;
$Nu^* = \alpha^* d_e/\lambda$	is the modified Nusselt number;
$Re_G = wd_e/\nu$	is the Reynolds number;
$Pr = \nu/\alpha$	is the Prandtl number.

LITERATURE CITED

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2. E. N. Bukharin, Inzh.-Fiz. Zh., 17, No. 5 (1969).