# AN ENTHALPY METHOD OF DESIGNING CONTACT-TYPE HEAT EXCHANGERS

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A graphical method of calculating the limiting water heating temperature and the thermal efficiency of contact and contact–surface heat exchangers is presented.

Heat exchangers in which the water is heated by direct contact with the gas ("contact water heaters") are becoming more and more widely used in industry in view of their considerable technical and economic advantages compared with the usual metal water heaters.

Besides their undoubted advantages, contact heat exchangers have a specific feature, namely, the temperature to which the water is heated in them is limited to the temperature of the wet thermometer, the value of which for gases at atmospheric pressure is less than 100°C, which is often insufficient for heat consumers. Hence, in such cases, the water, after heating in the contact heat exchanger, must be heated to the required temperature $t > t_M$ in an ordinary metal heat exchanger. A combined contact–surface heat exchanger is obtained. In view of the high efficiency of contact heating compared with surface heating, in the contact zone it is best to heat the liquid to the maximum possible temperature, i.e., to $t_M$. We will consider on what physical parameters $t_M$ depends. The quantity $t_M$ can be found from the expression for the balance of the flows of apparent heat from the gas to the liquid and the heat of vaporization from the liquid to the gas

$$\alpha (t_g - t_M) = \beta_x r (x''_M - x_g),$$

which after approximation of $x''_M$ by the expression $C_1 e^{c_2 t_M}$ reduces to the form

$$\frac{\alpha}{\beta_x} t_M + C_1 r e^{c_2 t_M} = \frac{\alpha}{\beta_x} t_g + r x_g. \tag{1}$$

The value $t_M$ can be found from Eq. (1) for specified $t_g$ and $x_g$, as a function of the ratio of the heat and mass transfer coefficients $\alpha/\beta_x$.

Another physical characteristic of the contact process between a liquid and a gas is the equilibrium temperature of the gas–liquid system established as a result of the completion of the heat and mass exchange process (the adiabatic saturation temperature t"). The value of t" can be found from the balance equation of the enthalpy of the initial and final state of the gases, which, after transformation, takes the form

$$I_1 = (c''_g + c_w x_1) t'' + C_1 r e^{c_2 t''} \tag{2}$$

It can be seen from (2) that t" is independent of the kinetic characteristics of the heat and mass transfer, and is determined uniquely by the value of the initial enthalpy of the gas. Hence, $t_M$ and t" have a quite different physical meaning.

In [1] it is stated that the position of the point $t_M$ on the saturation curve in the Id-diagram is indeterminate, and in general may not coincide with the point of intersection of the line $\tau$ = const and the saturation curve (i.e., the temperature t"). However, $t_M$ and t" are identified in the scientific literature. It is well-known [2] that $t_M$ and t" may be equal when there is an analogy between the heat and mass transfer during evaporation which corresponds to satisfaction of the ratio $\alpha/\beta_x \approx c$. It was previously assumed that in the gas temperature range up to 300-350°C during evaporation of water this analogy holds, which is the reason for the identification of $t_M$ and t". In earlier investigations [3] it was observed that during evaporation even at moderate gas temperatures (200-400°C) there is a disturbance of the analogy between the heat and mass transfer processes, and it is therefore possible for $t_M$ and t" to be different (this problem was not considered in [3]). Disagreement between $t_M$ and t" was more probable for contact between high-temperature gases and a liquid. In order to clarify this question, investigations of convective heat and mass

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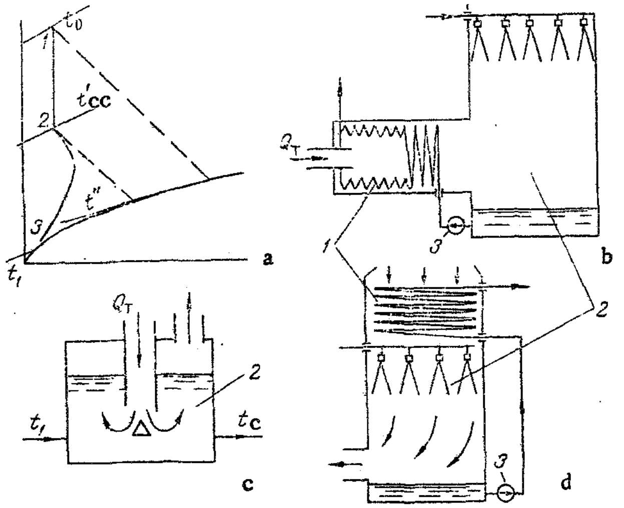

Fig. 1. Representation of the process in a heat exchanger on an Id-diagram (a); b, c, and d are sketches of the heat exchangers [b) combined with a counterflow sketch of the motion of the heat carrier; c) contact with a direct flow sketch of the motion of the heat carrier; d) combined with mixed counter flow—direct flow motion of the heat carrier]; 1) surface heat exchanger; 2) contact zone; 3) pump.

transfer between high-temperature gases (up to 1100°C) and water was investigated for adiabatic evaporation. Measures were taken in the experiments to eliminate the effect of radiation heat transfer. As a result of the experiments it was established that under these conditions $t_M$ and $t''$ are practically equal.

The use of this configuration enables one to determine $t_M$ from the more simple equation (2) as a function of the initial enthalpy of the gases $I_1$. The value of $I_1$ is usually known for a purely contact heat exchanger, but for the case of heating in a combined heat exchanger the gases, prior to entering the contact zone, are first cooled in the surface part. When using an Id-diagram to construct the processes taking place in a combined heat exchanger (Fig. 1b) the cooling of the gas in the surface zone will be represented by a vertical line 1—2, while the cooling in the contact zone will be represented by curve 2—3. The position of point 2 corresponds to the parameters of the gas on the boundary of the zones and is the value of the maximum temperature to which the water is heated in the contact zone $t_M$. Hence, for heat calculations of a combined heat exchanger it is first necessary to establish the position of point 2. The value of $t_M$ for combined heating (Fig. 1b) can be found from the equation

$$I'_{cc} = I_0 - Bc_W(t_c - t_M). \tag{3}$$

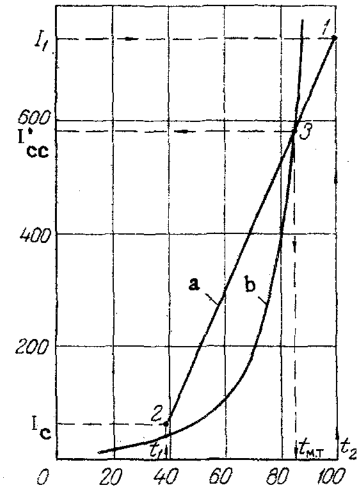

Fig. 2. Determination of the limiting temperature to which the water is heated in a combined heat exchanger: a) the working line, b) the saturation curve.

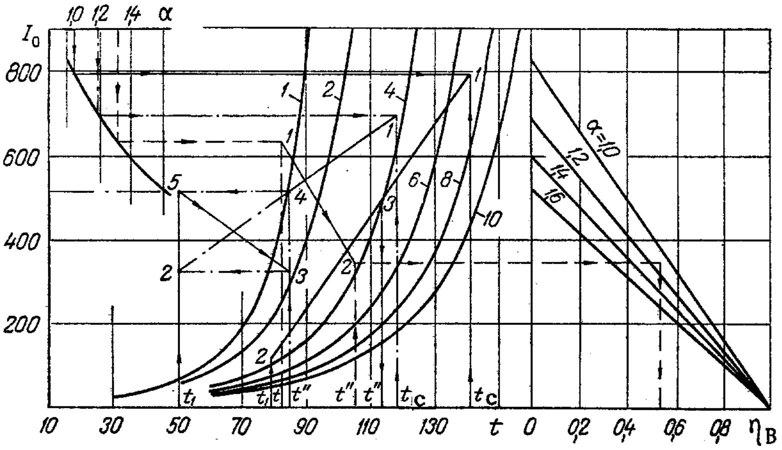

Fig. 3. Graphical construction of the heat-exchange processes in the It-nomogram: the continuous lines are combined counter flow for p = 4 bar, $\alpha = 1.05$, $t_c = 141°C$, and $t_1 = 78°C$; the dashed curves are for direct flow, p = 4 bar, $\alpha = 1.31$, t = 105°C, and $t_1 = 82°C$; the dash–dot lines are for a combined counter flow–direct flow exchanger with p = 2 bar, $\alpha = 1.19$, $t_c = 118°C$, $t_1 = 50°C$, and t" = 85°C.

Equating the required enthalpy of the gases at the input to the contact chamber $I'_{cc}$ to the adiabatic saturation enthalpy I", corresponding to $t_M$, we obtain

$$I'_{cc} = I'' = I_0 - Bc_w(t_c - t'') = c_g t'' + i'' \frac{0.622p''}{P - p''} . \tag{4}$$

Since p" is a unique function of t", Eq. (4) can be solved by the method of successive approximations. The graphical method presented below for determining the parameters of the heat carriers at the boundary of the zones is more convenient and more obvious. We will consider this method as it applies to the most widely used industrial heat exchangers shown in Figs. 1b, c, and d. Industrial models of contact water heaters with a thermal power of 4-25 GJ are constructed in accordance with the scheme shown in Fig. 1b [5,6], water heaters with immersion heaters operate as shown in the scheme of Fig. 1c [7], while the scheme shown in Fig. 1d is used in different thermal devices for utilizing the heat of exhaust gases [8].

Figure 2 shows two dependences on the water temperature: 1 is the enthalpy of the saturated mixture, and 2 is the enthalpy of the gases ("the working line"). The working line corresponds to the equation of the heat balance of the heat exchanger $I_0 - I_{ex} = Bc_w(t_c - t_1)$ and is constructed from the values of the initial and final enthalpy of the gas. The value of the initial enthalpy of the gas $I_0$ when using the combustion products of a burner depends on the value of the excess-air coefficient and is found from handbook data. The value of the final enthalpy of the gases $I_{ex}$, according to experimental investigations [4], can be taken as somewhat higher than the enthalpy of the saturated gases at the initial temperature of the water so that $I_{ex} = I'' + (15...20)$kJ/kg.

The point of intersection of the working line with the saturation curve (3) defines $I'_{cc}$ and $t_M$, which is the solution of Eq. (2).

The enthalpy method considered is universal. It can be employed both in the case of the counter-current phase and in the direct-current phase, and also for any pressures. Since the value of the enthalpy of gases is independent of the pressure, when the pressure changes the position of the working line remains unchanged, and only the position of the saturation curve changes in view of the fact that the enthalpy of the saturated mixture, according to the relation

$$I'' = c_g t'' + i'' \frac{0.622p''}{P - p''} , \tag{5}$$

for a given temperature, depends on the pressure of the process P (p" is a unique function of t").

To calculate the limiting temperatures to which the water can be heated in the contact zone and the thermal efficiency of the process for any pressures of the gas we designed the nomogram shown in Fig. 3. On the left side we plot a curve of the initial enthalpy of the gases $I_0$ as a function of the excess air coefficient $\alpha$. In the middle part we plot curves of the saturation enthalpy for different gas pressures constructed from Eq. (5).

On the right side we plot curves of the thermal efficiency of the heat exchanger $\eta_B$ as a function of the final enthalpy of the gas for different values of $\alpha$. Figure 3 shows an example of the construction of the process in a heat exchanger using the scheme of Fig. 1b for $p = 4$ bar, $\alpha = 1.05$, $t_1 = 78°C$, and $t_c = 141°C$. The procedure for constructing the curves is the same as that described above.

For direct-flow motion of the heat carrier (Fig. 1c) the procedure is somewhat different. In this case ($\alpha = 1.31$, $t_1 = 82°C$, and $t'' = t_c = 105°C$) the point of the final state of the gases is close to the saturation curve $p = 4$ bar at the final temperature of the water (exceeding it by the value of the final difference in the enthalpy of the heat carriers). The value of $t_c$ in this arrangement is specified in the calculation.

The most important difference between the direct-flow scheme compared with the counter-flow scheme is the presence of a direct relation between the thermal efficiency of the process and the value of the final temperature to which the water is heated $t_c$. When $t_c$ is increased the enthalpy of the exit gases $I''$ increases and the efficiency falls. For the conditions of the example, according to Fig. 3, $\eta_B = 0.53$.

The case of combined heat transfer with mixed counter-flow—direct-flow motion of the heat carrier (Fig. 1d) may be of interest in practice. The advantage of this arrangement compared with the arrangement shown in Fig. 1b is the possibility of attaining much higher gas velocities and intensification of the heat exchange in view of the fact that in direct flow there is no "flooding" of the contact chamber, characteristic of the counter-flow arrangement. Under certain conditions this advantage may compensate for the lower thermal efficiency of direct flow.

The thermal efficiency of a heat exchanger based on the arrangement shown in Fig. 1d, as in Fig. 1c, is determined by the assumed water temperature at the exit from the contact zone $t''$, the value of which mainly depends on the water flow rate through the heat exchanger. Hence, for thermal calculations, unlike the counter-flow arrangement shown in Fig. 1b, the value of $t''$ is known, and is chosen from technological and economic considerations; the problem is then to determine the parameters of the gas at the boundary of the zones. The value of the enthalpy of the gas before the contact zone $I'_{cc}$ can be found from the general and one of the two particular equations of the heat balance of the zones

$$I_0 - I''_{ex} = Bc_w(t_c - t_1) = I_0 - \left(c_{ex}t''_{ex} + i''\frac{0.622p''}{P - p''}\right), \tag{6}$$

$$I_0 - I'_{cc} = Bc_w(t_c - t''). \tag{7}$$

From (4) we have

$$B = \frac{I_0 - \left(c_{ex}t''_{ex} + i''\dfrac{0.622p''}{P - p''}\right)}{c_w(t_c - t_1)}, \tag{8}$$

and substituting (8) into (7) we obtain

$$I'_{cc} = I_0\left(1 - \frac{t_c - t''}{t_c - t_1}\right) + \left(c_{ex}t_{ex} + i''\frac{0.622p''}{P - p''}\right)\frac{t_c - t''}{t_c - t_1}, \tag{9}$$

where according to experimental data $t_{ex} = t'' + (2\ldots5)°C$. Equation (9), unlike Eq. (4), can be solved in explicit form, since $t''$ is specified. However, in this case also the graphical method is more convenient and clearer. In Fig. 3 we show an example of constructing the process for this arrangement under the following conditions: $p = 2$ bar, $\alpha = 1.19$, $t_1 = 50°C$, and $t_c = 118°C$. We choose the temperature to which the water is heated in the contact zone $t'' = 85°C$ and proceed as follows: 1) we plot the point 3 of the enthalpy of the exit gases somewhat above the saturation curve for $p = 2$ bar so that $I_{ex} = I'' + (10\ldots20)$ kJ/kg, 2) we transfer point 3 to the left along the horizontal until it intersects the vertical $t = t_1 = 50°C$ and obtain point 2 of the final state of the gases, 3) we plot the point 1 of the initial state of the gases corresponding to $\alpha = 1.19$ and $t_c = 118°C$, and we draw the working line 1–2, 4) at the intersection of the vertical $t = t'' = 85°C$ with the working line we obtain point 4, which indicates the required enthalpy of the gases at the boundary of the zones $I'_{c.c} = 515$ kcal/kg dry gas.

Completing the construction, we transfer point 4 to the left (point 5) and joint points 5 and 3. The section 1–4 represents the process for counter flow in the surface zone, while 5–3 represents forward flow in the contact zone.

Hence, the method described enables one to analyze rapidly and clearly the temperatures attained by the water when heated and the thermal relations for any pressures for all arrangements of practical interest: purely contact heat exchangers and their counter-flow and forward-flow conditions, and combined contact–surface heat exchangers for counter-flow and mixed counter-flow–forward-flow systems.

## NOTATION

$I_0 = Q_p^b/g_{dg}$, initial enthalpy of the gases, kJ/kg; $Q_p^b$, fuel-consumption heat, kJ/kg; $g_{dg}$, consumption of dry gas per unit of fuel, kg/kg; $I'_{cc}$, $I_{ex}$, enthalpies of the gases at the entrance and exit of the contact chamber, kJ/kg; $B = G_w/G_{dg}$, sprinkling coefficient; $G_w$, $G_{dg}$, flow rates of the water and gases, kg/h; $t_1$, $t_M$, and $t_c$, temperature of the water: initial, wet-bulb thermometer, and final, °C; $c_w$, $c_{ex}$, heat capacities of the water and exit gases, kJ/kg·°C; $t''$, $p''$, $I''$, temperature (°C), partial pressure of water vapor (bar), and enthalpy (kJ/kg) of the saturated vapor–gas mixture; $i''$, enthalpy of the vapor, kJ/kg; $t_{ex}$, temperature of the exit gases, °C; $\alpha$, $\beta_x$, heat and mass transfer coefficients, $kJ/m^2 \cdot h \cdot °C$ and $kg/m^2 \cdot h$; $c_g$, $x_g$, $t_g$, instantaneous heat capacity, kJ/kg·°C, moisture content, kg/kg, and temperature of the gases, °C; $x_1$ and $x''_M$, initial moisture content of the gas and the saturation moisture content, kg/kg.

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