Method of research on heat exchange for parallel plates with low thermal conductivity in short-time processes

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Abstract. Temperature of coolant was measured at the input and output of an alternating regenerator. Additionally, the temperature was measured both at several points on the surface of the parallel plates and along their geometrical axis. The package of the parallel plates was used as a nozzle of the regenerator. The method of determining heat transfer coefficients was described for the side and butt end surfaces of the plates in short-time processes. The mathematical models were created for the following two non-stationary processes. The first process is convective energy transfer in heat carrier flows. The second one is the process of thermal conductivity of two-dimensional body (plate) washed cyclically by cold and hot heat carrier flows. The coefficients of cyclic heat exchange with the cold and hot heat carrier flows were obtained for the side and butt end surfaces of the plates in short-time processes.

1. Introduction

Lots of thermal processes occurred at power plants are limited in time that is, they take place under non-stationary conditions. Such plants include regenerative air preheaters (RAPH) used by power steam generators, air cooling and air separation machines, gas turbine engines, metallurgical furnaces, ventilation and heating systems [1-3].

The unsteady nature of processes affects qualitatively and quantitatively the local heat transfer [3-7]. Therefore, the heat transfer coefficient averaged for the period of a nozzle becomes dependent on duration of the period and other factors [3, 4]. Thus, dependence of this coefficient on circumstances causes additional difficulties for RAPH developers.

In [8, 9] the description of an experimental alternating RAPH with nozzles in the form of parallel metal plates packages and the measurement results of the averaged heat transfer coefficients (Nusselt numbers) for the side surface are presented. However, applicability of these results to similar nozzles with plates of low thermal conductivity requires verification. This is necessary because using plastic materials such as polymethyl methacrylate (Plex) to produce nozzles of RAPH for heating and ventilation systems is profitable due to corrosion resistance, processability and low cost.

Another problem of RAPH that still hadn’t been studied is uncertainty of heat transfer of butt end surfaces of nozzle plates. According to geometry of these plates the entire square of their butt end surfaces is much less than square of side ones. Despite this fact the butt ends of nozzle plates due to their more intensive heat exchange have quite noticeable influence on the heat transfer process. The lack of data concerning the heat exchange coefficient of butt end surfaces is caused not only by...
underestimation of their influence on heat exchange of RAPH but also by limited abilities of traditional methods of heat exchange. Development of this method is essentially important to fill in this gap of knowledge.

The work presents the method of measurement of the heat transfer coefficient averaged for the period for the side and butt end plate surfaces of Plex.

2. Methods of the nozzle RAPH measuring heat transfer coefficient

The experiments were carried out by means of the test stand described in [8-10]. Experiments with switching periods from 2 to 40 s can be performed using this stand. A test with one of the shortest periods was studied and demonstrated in order to verify the method. The package of plates installed in parallel at distance \( h = 2 \text{ mm} \) from each other was used as the nozzle of the setup. The material of the plates is Plex, produced at LLC SafPlast, Kazan. Dimension of the package is determined by the sizes of the work section. The thickness of a separate plate was \( \delta = 2 \text{ mm} \), length \( l = 101 \text{ mm} \), width \( b = 50 \text{ mm} \), equivalent hydraulic diameter of inter-plate channels \( d_e = 3.32 \text{ mm} \). Thermocouples of chromel-copel wire with a diameter of \( d_w = 0.2 \text{ mm} \) were used to measure temperature of the central plate. The wires of thermocouples were welded by a resistance welding machine. The sensors were installed at the following distances \( z_w \) along the longitudinal axis in the middle of the plate from its cold end: 3.9, 25.6, 50.8, 77.8, and 95.5 mm on the lateral surface, and 5.7, 29.4, 50.8, and 96 mm on the axis. The thermocouples designed to measure temperature along the plate axis were laid in the grooves with the depth of 1 mm.

Measurement of temperature of the cold and hot air streams (heat carriers) was carried out by the same thermocouples. Three thermocouple sensors were installed both at the inlet and outlet of the RAPH [10]. The results of measuring temperature were digitized and recorded by a computer aid automatic measuring system (AMS). Figure 1 shows temperature variation of the nozzle and the air flows for one cycle. Duration \( \tau \) of the cold and hot periods is 2.06 and 2.42 s, respectively. The cycle is conventionally divided into four calculation periods indicated in Fig. 1. These periods are designated with roman numbers. The periods I and III are the working ones. In periods II and IV with duration of \( \Delta \tau_w = 0.34 \text{ s} \), switching from the cold period to the hot one and vice versa takes place.

As it is shown in Fig. 1, the most rapid changes of air flows temperatures take place at the beginning of the first and third periods, that is, right after the switching of the flows. An analytical description of the dependence of temperature flows at the input and output of the RAPH, taking into account corrections for inertia of thermocouples in each working period of the cycle, is described by the expression [10]:

\[
\theta_f = \theta_\infty \pm \exp(a_0 + a_1 t) + \sum_{n=0}^{k_f} c_n t^n.
\]

Where \( \theta_f = (T_f - T_{\text{min}})/T_\infty; \theta_\infty = (T_{\text{max}} - T_{\text{min}})/T_\infty; T_f \) is coolant temperature at the instant \( t \) from the beginning of the period, \( \theta_\infty \); \( T_\infty \) is coolant temperature at the end of the transition period, \( \theta_\infty; T_f = (T_{\text{max}} - T_{\text{min}})/2 \) is temperature scale, \( \theta_\infty; T_{\text{max}} \) and \( T_{\text{min}} \) are maximum and minimum temperatures of the hot and cold streams, respectively, \( \theta_\infty \); the signs "+" and "-" correspond to the cold and hot periods, respectively; \( t = \tau / \tau_\infty \); \( c_n \) are the coefficients of the polynomial, which determines the temperature of the flow at the inlet/outlet after the transition process; \( k_f \) is the order of the polynomial; \( a_0 \) and \( a_1 \) are coefficients determined from the joint solution of a system of two equations:

\[
a_0 = \ln(\theta_0 + b t_\infty);
\]
\[
\theta_0 t_s \left[ 1 - \exp \left( -1/t_s \right) \right] + \frac{\exp(a_0)}{1 + a_{1}t_s} \left\{ \exp(a_{1}) - \frac{1}{a_{1}} \right\} \left[ 1 - \exp \left( -1/t_s \right) \right] = \bar{\theta},
\]

where \( \theta_0 \) is the relative temperature of the stream at the initial moment of time; \( \bar{\theta} \) is the average for the period the relative temperature of the stream; \( t_s = \rho c V / (aF) \) is characteristic thermocouple time, s; \( \rho \) and \( c \) is density, kg/m\(^3\), and specific heat capacity of the thermocouple junction, J/(kg K); \( F \) and \( V \) is surface area, m\(^2\), and volume, m\(^3\), weld.

**Figure 1.** Cycle thermograms obtained by AMS for heat carriers (a) and the plate (b):
1, 2 – cold and hot heat carriers; 3–7 – on the surface of the plate; 8–11 – along the axis of the plate;
I–IV – periods: I and III – workers; II and IV – transitional.

Measured surface temperature of the plates was approximated by a double polynomial:

\[
\theta_w(Y,t) = \sum_{k=0}^{k_z} \sum_{n=0}^{k_t} b_{n,k} t^n
\]

where \( \theta_w = (T_w - T_{\text{min}})/T_s \); \( Y = y/l \); \( b_{n,k} \) are regression coefficients; \( T_w \) and \( T_{\text{min}} \) are measured and minimum temperatures, K; \( T_s \) is the temperature scale, K; \( y \) is the distance from the cold end, m; \( k_z \) and \( k_t \) are orders of polynomials.

Approximation of the obtained experimental results was made by means of expressions (1) and (2) with \( k_f = 2, k_z = 3, k_t = 4 \). Accuracy of this approximation can be assessed by analyzing Fig. 2 where the experimental data are shown by points, and the temperature values calculated by means of equations (1) and (2) are shown by lines.

The averaged along the surface heat transfer coefficients of the nozzle surfaces for each particular period were found by comparing the measured temperatures of the nozzle and coolants at the outlet of the RAPH with the calculated temperatures. Calculation of temperature fields in the air flow and the nozzle was carried out according to mathematical models of energy transfer in the flow and the nozzle.

3. Model energy transfer in flow

In [8–10] a linear law of temperature variation of the coolants along interdigital channels was assumed, which is admissible with a linear dependence of the surface temperature of the plates on the longitudinal coordinate. However, as it can be seen from Fig. 3, in experiments, a nonlinear dependence of the surface temperature is observed. Therefore, it became necessary to build a
mathematical model of heat exchange with the plate surface, which temperature varies according to law (2).

There is a known assumption that the heat transfer coefficient of the plate surface remains constant both in time and along the surface of the plate along each period. This assumption was utilized to build the mathematical model. In the coordinate system associated with the coolant flow $Z = Y_0 \pm Y \ (Y_0 = 0$ and "+" in the cold period; and "-" in the hot), the differential equation of the flow energy for the average coolant temperature in the cross section of the channel, taking into account the negligibly small longitudinal thermal conductivity of the flow compared to convective energy transfer, in relative variables, has the form [3]:

$$A_1 \frac{d\theta_f(Z,t)}{dt} + A_2 \frac{\partial \theta_f(Z,t)}{\partial t} + A_2 \left[ \theta_f(Z,t) - \theta_w(Z,t) \right] = 0,$$

(3)

where $A_1 = \frac{l}{w_z \tau_p}$; $A_2 = \alpha F_w / (G_f c_p)$; $w_z, F_w, G_f$ and $c_p$ are average flow rate, m/s, wetted body surface area, m$^2$, mass flow rate, kg/s, and specific isobaric heat capacity, J/(kg K), coolant in the j'th period. In the images of Laplace [11]

$$\theta_{f,\perp}(Z,s) = \int_0^\infty \exp(-s t) \theta_f(Z,t) dt \cdot \theta_{w,\perp}(Z,s) = \int_0^\infty \exp(-s t) \theta_w(Z,t) dt$$

(4)

equation (3) takes the form:

$$\frac{d\theta_{f,\perp}(Z,s)}{dz} + (A_2 + A_1 s) \theta_{f,\perp}(Z,s) = A_1 \theta_f(Z,0) + A_2 \theta_{w,\perp}(Z,s).$$

(4)

The general solution of equation (4) [12]:

$$\theta_{f,\perp}(Z,s) = \theta_{f,\perp}(0,s) \exp[-(A_2 + A_1 s)Z] + A_2 \int_0^Z \exp[(A_2 + A_1 s)(\eta - Z)] \theta_{w,\perp}(1,\eta,s) d\eta.$$

(5)

The Laplace temperature images on the right-hand side of (3) are based on relations (1) and (2) [11]:

Figure 2. Measured temperature variations of heat carriers (a) and the plate (b) during a cycle:
1, 2 – at the input and output of the working section; 3–7 – on the surface of the plate;
8–11 – along the longitudinal axis of the plate.
\[ \theta_{f,L}(0,s) = \frac{\theta_\infty}{s} + \frac{\exp(a_0)}{s - a_1} + \sum_{n=0}^{k_2} \frac{c_n}{n!} s^{|s|+1}, \quad \theta_{w,L}(Z,s) = \sum_{k=0}^{k_2} \left( Y_0 \pm Z \right)^k \sum_{n=0}^{k_1} \frac{b_{n,k}}{n!} \frac{n!}{s^{n+1}}. \]

After substituting the right-hand sides of the expressions for these quantities into equation (4) and subsequent integration, an expression is obtained for the image of the coolant temperature:

\[ \theta_{f,L}(Z,s) = G_{0,L}(Z,s) + G_{1,L}(Z,s), \tag{6} \]

where

\[ G_{0,L}(Z,s) = \left( \frac{\theta_\infty}{s} + \frac{\exp(a_0)}{s - a_1} + \sum_{n=0}^{k_2} \frac{c_n}{n!} s^{|s|+1} \right) e^{-(A_2 + A_3)Z}; \]

\[ G_{1,L}(Z,s) = A_2 \sum_{k=0}^{k_2} \frac{(-1)^k}{k_2} \sum_{n=0}^{k_2} \frac{n!}{n!} \left[ \sum_{p=k}^{n} \frac{p! b_{n,p} Z^{n-k}}{(p-k)!} \right] e^{-(A_2 + A_3)Z} . \]

The relative temperature original of the coolant is found by the inverse Laplace transformations of the expression (6) [11]:

\[ \theta_f(Z,t) = G_0(Z,t) + G_1(Z,t), \tag{7} \]

where

\[ G_0(Z,t) = \left[ \theta_\infty + e^{A_2(t-A_2Z)} \right] e^{A_2Z}; \]

\[ G_1(Z,t) = \sum_{k=0}^{k_2} \frac{(-1)^k}{A_2} \sum_{n=0}^{k_2} \frac{n!}{n!} \left( \sum_{p=k}^{n} \frac{p! b_{n,p} Z^{n-k}}{(p-k)!} \right) \frac{K_{k,n-r}}{n!} \left( \frac{A_1}{A_2} \right)^{n-r} \left[ t - k b_{n,k} (t - A_1Z) e^{-A_2Z} \right]; \]

\[ K_{k,n} = K_{k,n-1} + K_{k-1,n}; \quad K_{k,0} = 1; \quad k, n \geq 0. \]

**Figure 3.** The measured longitudinal temperature distribution of the surface of the plate; curve 1 is at the end of the cold period; curve 2 is the same hot; the points are measurement results; the lines the result of approximation by polynomial (2).

**Figure 4.** Longitudinal temperature distributions of heat carriers and plates: 1–4 – by expression (7); and 6 – nozzle axes according to equation (15); 1 and 2 – at the beginning of periods; 4–6 – at the end of periods; 1, 3 and 5 – in the cold period; 2, 4 and 6 – in the hot; icons – measured temperatures of the nozzle axis: 7 – at the end of the cold period; 8 – the same hot.
4. Plate energy transfer model

The model is a solution to the boundary value problem of thermal conductivity of the third kind on a plate under cyclic boundary conditions in a two-period cycle [3, 13]:

$$\frac{\partial \theta}{\partial t} = F_0 \left( \frac{\partial^2 \theta}{\partial X^2} + \frac{1}{L^2} \frac{\partial^2 \theta}{\partial Y^2} \right), \quad t \geq 0, \ 0 < X < 1, \ 0 < Y < 1;$$

$$\theta_j(X, Y, 0) = \theta_{j, -1}(X, Y, t_{p, -1});$$

$$\frac{\partial \theta_j(0, Y, t)}{\partial X} = 0;$$

$$\frac{\partial \theta_j(1, Y, t)}{\partial X} = -Bi_{x,j} \left[ \theta_j(1, Y, t) - \theta_{f, j}(Y, t) \right];$$

$$\frac{\partial \theta_j(X, 0, t)}{\partial Y} = Bi_{y, 0,j} \left[ \theta_j(X, 0, t) - \theta_{f, j}(0, t) \right];$$

$$\frac{\partial \theta_j(X, 1, t)}{\partial Y} = -Bi_{y, 1,j} \left[ \theta_j(X, 1, t) - \theta_{f, j}(1, t) \right].$$

Here \( X = 2x/\delta_w; \) \( x \) is the transverse coordinate of the point, \( m; \) \( L = 2l/\delta_w; \) \( F_0_{p,j} = 4\alpha_{w,j} \tau_p/\delta_w^2 \) is limiting Fourier number in the \( j' \)th period \( (j = 0, 1); \) \( \alpha_j = \lambda_j / \rho c_p \) is coefficient of thermal diffusivity, \( m^2/s; \) \( Bi_{x,j} = \alpha_{x,j} \delta_w / (2\lambda_j); \) \( Bi_{y,0,j} = \alpha_{x,0,j} l / \lambda_j; \) \( Bi_{y,1,j} = \alpha_{x,1,j} l / \lambda_j; \) \( \alpha_{x,j}, \alpha_{y,0,j}, \alpha_{y,1,j} \) are the averaged along the surface heat transfer coefficients of the side surface, the front and rear ends of the plate in the \( j' \)th period.

The coolant temperature for the period varies according to the law:

$$\theta_{f, j}(Y, t) = \theta_{w, j} + e^{a_{0,j}} t + \sum_{k=0}^{n_x} \sum_{l=0}^{n_y} g_{k,l,j} t^l,$$

in which the coefficients \( g_{k,l,j} \) are found by approximating the solution (7) for the \( j' \)th period.

The solution of problem (8) - (14) is double Fourier series [13]:

$$\theta_{w,j}(X, Y, t) = \sum_{n=1}^{\infty} A_{n,j} \cos(\mu_{n,j} X) \sum_{m=1}^{\infty} A_{m,j} K_{j, y} \left( r_{m,j} Y \right) \times \theta_{L,j} \left( \mu_{n,j}, \gamma_{m,j}, 0 \right) \exp(-r_{n,m,j}^2 t) + \Psi_j(t),$$

where

$$\Psi_j(t) = F_0_{p,j} \left[ a_{n,j} \theta_{n,j} \frac{1 - \exp(-r_{n,m,j}^2 t)}{r_{n,m,j}^2} \exp(a_{0,j}) - \sum_{i=0}^{n_x} d_i S_{f, i}(t) \right].$$

Expressions for \( A_n, A_m, K_j(\gamma_{m,y}), \mu_n, \gamma_m, \theta_{L,j}(\mu_n, \gamma_m, 0), r_{n,m,j}^2, \omega_0, d_i \) are given in [13].

5. Results and discussion

From the boundary conditions (11) - (13) it can be seen that the temperature fields on the plate are influenced by both heat transfer from the side surface, that is \( Bi_{j,1}(Y, t) \) and heat transfer from the end surfaces, that is, \( Bi_{y,0,j}(0, t) \) and \( Bi_{y,1,j}(1, t) \). The system of two equations (7) and (15) was solved by the method of successive approximations, sequentially changing the coefficients \( A_n, n_x \), \( A_{y,0} \) and \( A_{y,1} \) in the following equations:

$$\text{Nu}_x = A_x \text{Re}^{n_x} \text{Pr}^{0.4} \left( \text{Pr}_f / \text{Pr}_w \right)^{0.4} C_1,$$

$$\text{Nu}_{y,0} = A_{y,0} 0.068 \text{Re}^{0.73} \text{Pr}^{0.4}$$

$$\text{Nu}_{y,1} = A_{y,1} 0.4 \text{Re}^{0.5} \text{Pr}^{0.4}.$$
Here $\text{Nu} = \alpha d / \lambda_f$ is the Nusselt number; $\text{Re} = w_f d / \nu_f$ is Reynolds number; $w_f$ is average coolant velocity, m/s; $\text{Pr} = \nu_f / \alpha_f$ is Prandtl number; $C_l = 1.906(d_f / l)^{0.173}$ is correction for the channel length in the laminar flow regime; $\lambda_f$, $\nu_f$ and $\alpha_f$ are thermal conductivity coefficients, W/(m K), kinematic viscosity and thermal diffusivity, m$^2$/s, coolant.

The optimal values of the coefficients $A_x$, $n_x$, $A_y$ and $A_y$ corresponded to the minimum of the sum of two functionals:

$$\phi_j = \frac{1}{3} \sum_{m=1}^{4} \left[ \theta_{w,j}(0,Y_m,t) - \bar{\theta}_{w,j}(Y_m,t) \right]^2 + \left[ \theta_{f,j}(1,t) - \bar{\theta}_{f,j} \right]^2,$$

where $\theta_{w,j}(0,Y_m,t)$ and $\bar{\theta}_{w,j}(Y_m,t)$ are calculated according to expressions (7) and (15), the values of the relative temperatures of the plate on its axis and the corresponding coolant at the outlet of RAPH, respectively; $\bar{\theta}_{w,j}(Y_m,t)$ and $\bar{\theta}_{f,j}$ are the relative values of the measured temperatures of the nozzle and the coolant in the appropriate places.

The sum of the functionals $\phi_j$ was minimized by the simplex method [14].

The results of calculations with cold and hot flow rates of $G_f$ 32.6 and 32.4 g/s, for air flow speeds in the interplates channels about 24 and 28 m/s and durations of periods $\tau_p$ of 2.06 and 2.42, respectively, are shown in Figs. 4-6 and in the table.

The longitudinal temperature distribution of coolants, calculated by expression (7) and shown in Fig. 4, changes its character from nonlinear at the beginning of each period to linear at the end of it. The theoretical longitudinal distribution of the plate axial temperature satisfactorily agrees with the measured temperatures.

![Figure 5](image1.png)

**Figure 5.** Changes in the cycle temperature of coolants: 1, 2, 5, 6 – at the entrance to the RAH; 3, 4, 7–12 – at the exit from RAPH; 5–10 – expression (7); 9–12 – the average for the period values; 1–4, 11 and 12 – measurement results.

The temperatures of the RAPH coolants at the outlet (Fig. 5) also coincide with satisfactory accuracy with the measured corresponding values, taking into account the above corrections for the inertia of thermocouples. The discrepancies between the mean temperatures of 9 and 11, 10 and 12 are less than 3 K.

The calculated plate temperatures at the locations of thermocouple junctions on the axis, shown in Fig. 6, differ somewhat from the measured values. These discrepancies, especially near the hot end of
the plate, are probably due to the displacement of thermocouple junctions in thickness relative to the plate axis. Near the cold end, satisfactory, both in qualitative and quantitative terms, the nature of theoretical dependence of temperature on time and behavior of the measured temperature is observed.

### Table. Calculation results.

| Period   | \( Q \), W | \( \alpha_x \), W/(m²K) | \( \alpha_{x0} \), W/(m²K) | \( \alpha_{x1} \), W/(m²K) | Re | Foₚ | Nuₓ |
|----------|-------------|--------------------------|--------------------------|--------------------------|----|-----|-----|
| Cold     | 358         | 320                      | 411                      | 392                      | 4593 | 0,2337 | 38,3 |
| Hot      | 352         | 326                      | 419                      | 408                      | 4239 | 0,2742 | 37,1 |

The values of heat load given in the table in certain periods indicate compliance with the heat balance.

The values of the Nusselt numbers for the side surface, given in the table, are comparable to the calculated values using the similarity equation recommended for a package of similarity numbers for the metal plates in the range: \( 450 \leq \text{Re} \leq 8460, 16,6 \leq \text{Fo}_p \leq 21760 \) [15]:

\[
\text{Nu} = 1,06 \left( \frac{\text{Re}}{10^3} \right)^{0,14} \left( \frac{\text{Fo}_p}{10^3} \right)^{-0,069} \text{Nu}_\text{st},
\]

where \( \text{Nu}_\text{st} \) is the Nusselt number for stationary heat exchange conditions:

\[
\text{Nu}_\text{st} = \begin{cases} 
\text{Nu}_i & \text{at } \text{Re} < 2300, \\
\text{Nu}_i (1 - \gamma) + \gamma \text{Nu}_j & \text{at } \text{Re} > 2300;
\end{cases}
\]

\[
\text{Nu}_i = \begin{cases} 
\text{Nu}_{\text{min}} & \text{at } \text{RePrh}/l < 100, \\
\text{Nu}_{\text{min}} & \text{at } \text{RePrh}/l > 100;
\end{cases}
\]

\[
\text{Nu}_{\text{min}} = 8,24 - 16,5 \frac{h}{b} + 20,7 \left( \frac{h}{b} \right)^2 - 8,8 \left( \frac{h}{b} \right)^3;
\]

\[
\text{Nu}_{\text{lam}} = 1,55 \left( \frac{\text{Re} d_s}{l} \right)^{0,4} \text{Pr}^{1/3} \left( \frac{\text{Pr}_f}{\text{Pr}_w} \right)^{1/4} \text{C}_t;
\]

\[
\text{Nu}_f = 0,021 \text{ Re}^{0,8} \text{ Pr}^{0,43} \left( \frac{\text{Pr}_f}{\text{Pr}_w} \right)^{1/4} \text{ C}_t;
\]

\[
\text{C}_s = \exp \left[ 7,41 + 12,893 \ln \left( \frac{d_s}{l} \right) \right] \text{ is correction for the channel length in the turbulent flow regime.}
\]

The values obtained according to equation (16) \( \text{Nu} = 26.3 \) and 24.7 are about in third less than the values given in the table. Such discrepancies can be explained by the fact that the Fourier \( \text{Fo}_p \) numbers for plastic plates are beyond the limits of applicability of equation (16).

The described method determines not only the heat transfer coefficients of the plate side surfaces, but also the heat transfer coefficients of the butt end surfaces. As it is shown in the table, the heat transfer at the ends is more intensive than on the side surface.

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Conclusion
1. The method of measuring the heat transfer coefficients for the side and butt end surfaces of the low thermal conductivity plates in short-time processes is proposed.
2. Satisfactory agreement of the theoretical temperature fields in the nozzle and coolant flows with the proper experimental values was obtained.
3. The value of the heat transfer coefficient for the butt end of the plate was measured the first time ever.

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