CONTENTS AUTHORS

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DYNAMIC ANALYSIS AND MODELING OF DIRECT HEAT EXCHANGERS

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ABSTRACT
In this paper is taken for analysis the heat exchanger through which circulates the fluid with definite flow and temperature. For the model is taken the heat exchanger with additional electric heater placed inside it. Specifically is analyzed and modified the heat exchanger by cases where the heat exchanger walls are isolated and not thermally isolated, and when the accumulation of heat can also take place into the heat exchanger walls. As a result of the paper are given the mathematical models that describe the thermal balances for the mentioned cases and computation of the dynamics of fluid temperature into the heat exchanger. Obtained diagrams represent the change of temperature in function of time and other thermal-physical parameters.

Key words: heat direct exchangers, thermal isolation, temperature, flow.

1. INTRODUCTION
Heat exchangers in the heating and cooling technique can be said that have the main role and heat exchange in most cases done by convection [1]. Heat exchangers are different types and materials and are classified depending on the direction of fluid flow. For intensive heat exchange from water to the wall and from the wall to the secondary part of fluid [2], should be used materials which have large thermal conduction and resistance to high pressures (e.g. materials as iron, copper, aluminum and its alloys, etc.).

2. DIRECT HEAT EXCHANGERS

2.1 Direct heat exchangers thermally insulated
In fig. 1 is shown a heat exchanger where heat is transported directly from the thermal mixture mass inflow and additional heat through electric heater (steam or any other heater) [3]. In bowl is set a mixing tool to mix fluid creating homogeneous thermal field, i.e. the same temperature [4]. As the model is taken the dynamic change of temperature at the exit from the exchanger, with the following assumptions: mass flow by entrance and exit are constant: $\dot{m}_h = \dot{m}_d = \dot{m}$; thermally insulated exchanger: $\dot{Q}_T = 0$; ideal mixing in the exchanger: $t_b = t_d$; constant specific heat of fluid [5]: $c_p = konst.$, for fluid overlooked member $p \cdot v$ in the expression of specific internal heat $u = i - p \cdot v$, ie $u = i = c_p \cdot v$.

Heat conservation equation has the usual form:

$$\dot{Q}_h + P_{el} - \dot{Q}_d = \frac{dE}{d\tau} \tag{1}$$

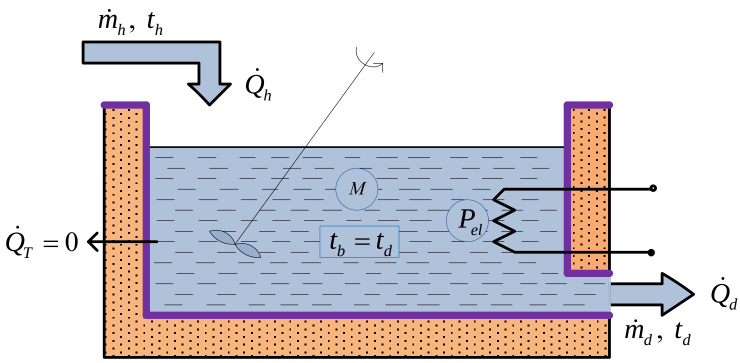

Figure 1. Isolated directly Heat exchangers

$$\dot{Q}_h = \dot{m} \cdot c_p \cdot t_h \,;\; \dot{Q}_d = \dot{m} \cdot c_p \cdot t_d \,;\; E = M \cdot c_p \cdot t_d \tag{2}$$

Where: *E, J* – heat energy; P_{el} *, W* – electric heater; $\dot{Q}_h$*, W* – incoming heat flux; $\dot{Q}_d$*, W* – outgoing heat flux; t_h*,* t_d*,* ^0C – temperatures at the entrance and exit; c_p*, J/(kgK)* – fluid specific heat; *M, kg* – accumulated mass of water. With the recent regulation of expression emerges:

$$T\frac{dt_d}{d\tau} + t_d = t_h + \frac{1}{\dot{m}c_p} \cdot P_{el} \tag{3}$$

Time constant is $T = M / \dot{m}$. From the above assumptions derived EDZ (ordinary differential equations) standard linear proportional order for members to view. If not fulfilled $\dot{m} = konst.$, equation may not be linear as products $\dot{m} \cdot t_d \cdot c_p$ and $\dot{m} \cdot t_h \cdot c_p$ expression no longer linear. Equation (3), in the form of Laplace transform is:

$$t_d(s) = \frac{1}{Ts+1} t_h(s) + \frac{1}{\dot{m} \cdot c_p} \frac{1}{Ts+1} \cdot P_{el}(s) \tag{4}$$

Solution of the equation is:

$$t_d = t_h + \frac{1}{\dot{m} \cdot c_p} \cdot P_{el} + e^{-\frac{\tau}{T}} \left[t_0 - \left(t_h + \frac{1}{\dot{m} \cdot c_p} \cdot P_{el} \right) \right] \tag{5}$$

2.2 Directly heat exchangers without thermal insulation

The following is presented the mathematical model which does not differ much from the model above except: exchanger isn't thermally insulated, $\dot{Q}_T \neq 0$; no heat accumulation on the walls of the exchanger (exchanger with thin walls), the coefficient of thermal conductivity through the walls is infinitely large: $\lambda = \infty$.

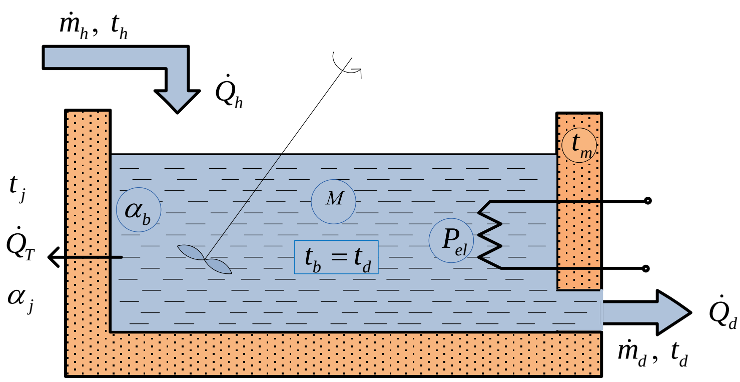

Figure 2. No isolated directly heat exchangers

Heat conservation equation now has the following form:

$$\dot{Q}_h + P_{el} - \dot{Q}_{\alpha_j} - \dot{Q}_d = Mc_p \frac{dt_d}{d\tau} \quad \text{or} \quad \dot{m}c_p \cdot t_h + P_{el} - \alpha_j \cdot A_j (t_d - t_j) - \dot{m}c_p \cdot t_d = Mc_p \frac{dt_d}{d\tau} \tag{6}$$

After adjustment of the equation (6) is obtained:

$$\dot{m}c_p \cdot t_h - t_d(\dot{m}c_p + \alpha_j \cdot A_j) + P_{el} + \alpha_j \cdot A_j \cdot t_j = Mc_p \frac{dt_d}{d\tau} \tag{7}$$

if equation (7) divided by ($\dot{m}c_p + \alpha_j \cdot A_j$), then we will have:

$$\underbrace{\frac{\dot{m} \cdot c_p}{\dot{m} \cdot c_p + \alpha_j \cdot A_j}}_{k_1} \cdot t_h + \underbrace{\frac{1}{\dot{m} \cdot c_p + \alpha_j \cdot A_j}}_{k_2} \cdot P_{el} + \underbrace{\frac{\alpha_j \cdot A_j}{\dot{m} \cdot c_p + \alpha_j \cdot A_j}}_{k_3} \cdot t_0 = \underbrace{\frac{M \cdot c_p}{\dot{m} \cdot c_p + \alpha_j \cdot A_j}}_{T_1} \cdot \frac{dt_d}{d\tau} + t_d \tag{8}$$

After adjustment will have:

$$k_1 \cdot t_h + k_2 \cdot P_{el} + k_3 \cdot t_j - t_d(\tau) = T_1 \cdot \frac{dt_d(\tau)}{d\tau} \tag{9}$$

And after converting into Laplace:

$$t_d(s) = \frac{k_1}{T_1 s + 1} t_h(s) + \frac{k_2}{Ts + 1} \cdot P_{el}(s) + \frac{k_2}{Ts + 1} t_j(s) \tag{10}$$

Solution of the equation is:

$$t_d = k_1 \cdot t_h + k_2 \cdot P_{el} + k_3 \cdot t_j + e^{-\frac{\tau}{T_1}} \left[t_0 - \left(k_1 \cdot t_h + k_2 \cdot P_{el} + k_3 \cdot t_j \right) \right] \tag{11}$$

Where $T_1 = \dfrac{M \cdot c_p}{\dot{m} \cdot c_p + \alpha_j \cdot A_j}$, s – time constant.

2.3. Directly heat exchangers with intensive mixing and without thermal insulation

This model is based on the assumption that the mixing of fluid flow is very intensive, which provided high coefficient of heat convection α_b on the inner side of the wall. Then reached $t_m = t_d$, and near $\lambda = \infty$ have thermal equilibrium equation:

$$\dot{m}c_p \cdot t_h + P_{el} - \dot{m}c_p \cdot t_d = (Mc_p + M_m c_m) \frac{dt_d}{d\tau} \tag{12}$$

After Laplace transformations obtained forms:

$$t_d(s) = \frac{1}{T_3 s + 1} t_h(s) + \frac{1}{\dot{m} \cdot c_p} \frac{1}{T_3 s + 1} P_{el}(s) \tag{13}$$

Solution of the equation is:

$$t_d = t_h + \frac{1}{\dot{m} \cdot c_p} \cdot P_{el} + e^{-\frac{\tau}{T_3}} \left[t_0 - \left(t_h + \frac{1}{\dot{m} \cdot c_p} \cdot P_{el} \right) \right] \tag{14}$$

Where $T_3 = \dfrac{Mc_p + M_m c_m}{\dot{m}c_p}$, s – constant time.

3. RESULTS OBTAINED BY DYNAMIC ANALYSIS OF MODELS OF HEAT EXCHANGERS

Judging from previous cases we will see graphically the exit temperature changes depending on the time and inflow. General Information: electrical heat capacity P_{el}= 12000 *W*, fluid accumulated mass M=17 kg, wall accumulated mass M_m=50 kg, the outer surface of the exchangers A_j = 0.5 m^2, the inflow mass $\dot{m} = 0.07 \div 0.14$ kg/s, entrance temperature t_h=12 ^{0}C, wall temperature t_m = 15 ^{0}C, for time τ = 0÷3600 s, specific heat of water c_p=4187 J/(kgK), Initial water temperature t_0 = 10 ^{0}C. Heat transmission coefficient k = 0 W/m^2K

Case 1

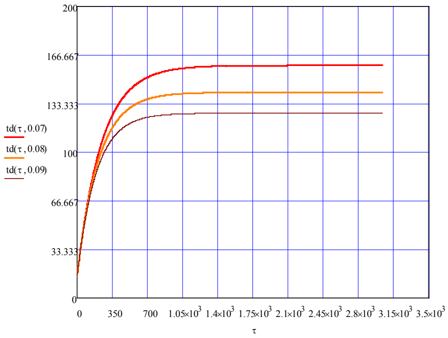

Figure 3. Temperature at the exit of fluid depending on the time and mass inflow

Case 2

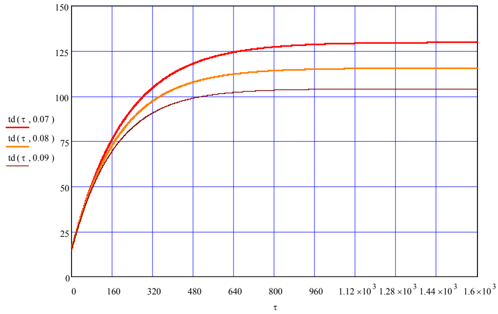

Figure 4. Temperature at the exit of fluid depending on the time and the mass inflow

Mass flow $\dot{m}$=0.07 kg/s, $k = 1/(\frac{1}{\alpha_{air}} + 0 + \frac{1}{\alpha_{water}})$, $W/(m^2K)$, $\alpha_{water} = 1000$ W/(m^2K),

$\alpha_{air} = 25$ W/(m^2K)

Case 3

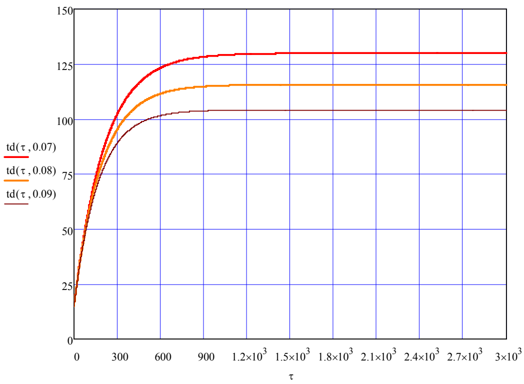

Figure 5. Temperature at the exit of fluid depending on the time and the mass inflow

$\lambda = \infty$; $\alpha_{water} = \infty$; $k = \alpha_{air} = \alpha_j = 25$ W/(m^2K)

4. CONCLUSION

Based on the done analysis we see that the maximum temperature achieved at the exit for flow $\dot{m} = 0.07$ kg is t_d =159.402^{0}C, which reach for the time τ = 2712,196 s, while for mass inflow greater $\dot{m} = 0.08$ kg / s, the maximum temperature reached at exit is t_d = 140.97 ^{0}C, for time τ = 2088,718 s. While the flow $\dot{m} = 0.09$ kg / s, maximum temperature is t_d = 126.6 ^{0}C and time τ = 1472.16 s. On this basis we see that with increasing mass flow decreases the maximum temperature at the exit and the time for its achievement. The same applies to other cases. From the above figures we see that thermal processes in the physical sense are dependent on the heat accumulation and flow processes and that presented no action inertia, and in fact there are periodic actions.

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