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HEAT EXCHANGER BETWEEN NEWTONIAN VISCOUS FLUIDS, WITH FLAT SURFACES, WITHOUT PHASE CHANGE, IN LAMINAR FLOW REGIME

Radu TARULESCU[1], Liviu COSTIUC[2], Victor BENCHE[3]
[1]TRANSILVANIA University of Brasov, radu.tarulescu@unitbv.ro
[2]TRANSILVANIA University of Brasov, lcostiuc@unitbv.ro
[3]TRANSILVANIA University of Brasov, victor.benche@unitbv.ro

***Abstract*—** The paper analyzes systems of heat and mass transfer, with heat exchangers of flat surfaces, rectangular, with two liquid thermal agents. The liquids are viscous, slow, and the flows are laminar, type Hagen - Poiseuille or Bingham.

It is studied, theoretical and applied, a heat exchanger "in plates" used as a cooler (radiator) or a recover system, main thermal agent circulating in a closed system with recirculation, assisted by thermo-siphoning.

***Keywords*—** flow, heat, laminar, resistance

I. INTRODUCTION

THE scheme of a heat transfer system between two viscous fluids, the first with closed circuit recirculation is shown in Fig. 1.

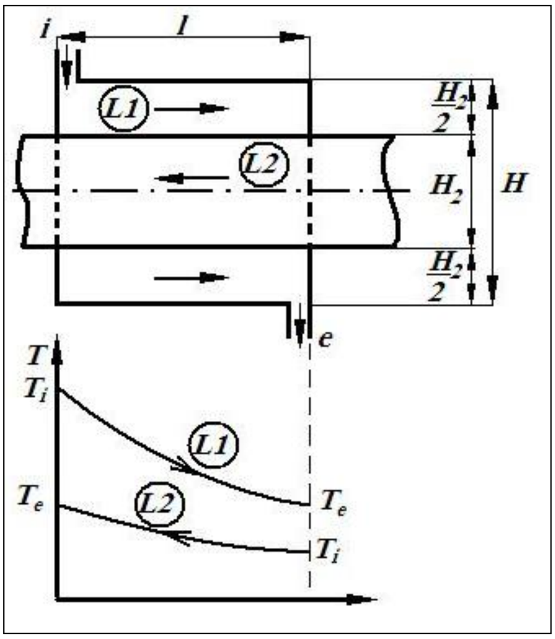

Fig. 1. The heat transfer system between two viscous fluids

"Warm body" A is a generator type source of heat and represents thermal agent heater L_1, with the flow of heat Q. The beneficiary B represents the heat consumer, being received in the form of hot water and hot air. The thermal hydraulic system of fluid in the closed circuit L_1 may work through thermo-siphoning and/or override, using a pump P and a fan V, the surface heat exchanger R being a heat radiator for the liquid L_1, and a heater for the liquid L_2 and air, as also an economizer for the beneficiary.

The consumer radiator heat transfer circuit is open, without recirculation, as short, with thermally insulated pipes. The radiator R has n double concentrically pipes, with rectangular crossing sections and flat surfaces [1].

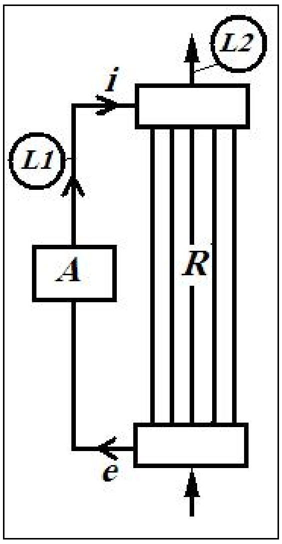

Fig. 2. The heat transfer system without a cooling fan

Through inner pipe circulates liquid L_2, through the space between it and the outer rectangular pipe (in contact with the air) circulate the fluid L_1, in countercurrent, with cooling from downward movement

[2].

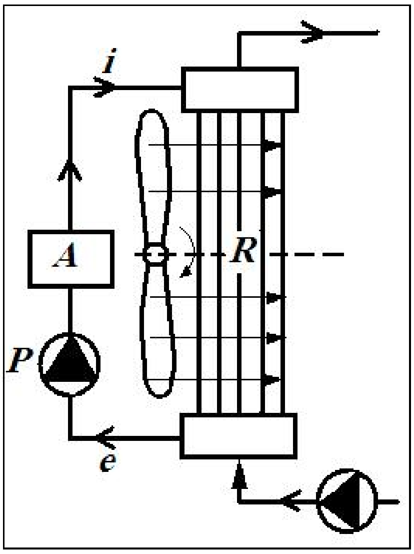

Fig. 3. The heat transfer system with a cooling fan

The flow of air delivered by the fan circulates axially in cross-flow like in Fig. 3.

II. SPEED DISTRIBUTION IN A FLUID LOCATED BETWEEN TWO PARALLEL FLAT PLATES

The distribution of speed in a viscous fluid located between the two parallel flat plates, when the plates are fixed and only fluid moves, is shown in Fig. 4.

Choosing the origin of axes at distance *a/2*, the boundary conditions are expressed as follows:

$$\begin{cases} \mathbf{z = \pm \frac{a}{2}} \\ \mathbf{v = 0} \\ \mathbf{\frac{\partial p}{\partial z} = const} \end{cases} \tag{1}$$

The general equation of motion for the incompressible fluid, laminar, is reduced to [3]:

$$\mathbf{\frac{d^2v}{dz^2} = \frac{1}{\eta} \cdot \frac{\partial p}{\partial x}}, \tag{2}$$

and by integration will result:

$$\mathbf{v = \frac{1}{\eta} \cdot \frac{\partial p}{\partial x} \cdot \frac{z^2}{2} = C_1 \cdot z + C_2}, \tag{3}$$

where:
p = pressure;
v = speed;
η = the coefficient of dynamic viscosity.

By putting the boundary conditions results:

$$\mathbf{C_1 = 0,\ C_2 = -\frac{1}{\eta} \cdot \frac{\partial p}{\partial x} \cdot \frac{a^2}{4}}, \tag{4}$$

and the speed will have the expression:

$$\mathbf{v = -\frac{1}{2 \cdot \eta} \cdot \frac{\partial p}{\partial x} \cdot \left(\frac{a^2}{4} - z^2 \right)}. \tag{5}$$

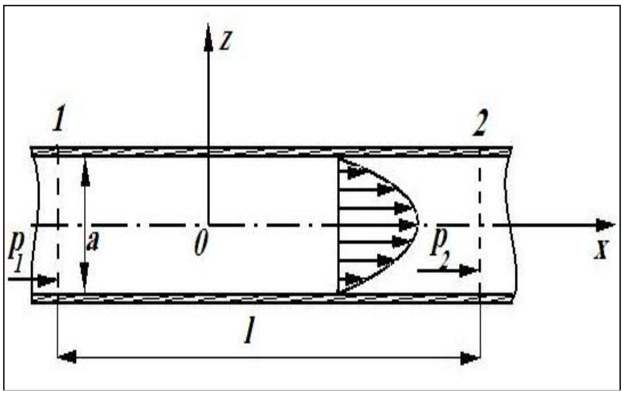

Fig. 4. The distribution of speed in a viscous fluid

Taking into account the fact that:

$$\mathbf{-\frac{\partial p}{\partial x} = -\frac{p_2 - p_1}{l} = \frac{p_1 - p_2}{l}}, \tag{6}$$

we can write the relation:

$$\mathbf{v = \frac{p_1 - p_2}{2 \cdot \eta \cdot l} \cdot \left(\frac{a^2}{4} - z^2 \right)} \tag{7}$$

And results:

$$\mathbf{v_{max} = \frac{p_1 - p_2}{8 \cdot \eta \cdot l} \cdot a^2}. \tag{8}$$

The flow rate of fluid which flows between the two plates across the width b is obtained as [4]:

$$\mathbf{\dot{V} = b \cdot \int_{-\frac{a}{2}}^{+\frac{a}{2}} v \cdot dz = \frac{p_1 - p_2}{12 \cdot \eta \cdot l} \cdot a^3 \cdot b}. \tag{9}$$

Average speed in the cross section $A = a \cdot b$ is:

$$\mathbf{v_m = \frac{\dot{V}}{A} = \frac{p_1 - p_2}{12 \cdot \eta \cdot l} \cdot a^2} \tag{10}$$

Differential pressure across the length l is given by:

$$\mathbf{\Delta p = p_1 - p_2 = \frac{12 \cdot \dot{V} \cdot \eta \cdot l}{a^3 \cdot b} = \frac{12 \cdot \eta \cdot v_m \cdot l}{a^2}} \tag{11}$$

III. NUMERICAL APPLICATION

In the first case (Fig. 4), thermal agent L_1 is a viscous liquid, non-Newtonian with laminar motion, having physical properties: density $\rho_m = 1086.5 \frac{kg}{m^3}$ and dynamic viscosity $\eta_m = 8.39 \cdot 10^{-3} \frac{kg}{m \cdot s}$. Adherent and laminar thickness is: $\delta_s = 0.45mm$.

Tangential effort of viscous friction to the wall, the threshold stress, is [5]:

$$\tau_0 = \rho_m \cdot g \cdot \delta_s . \tag{12}$$

Making substitutions will result: $\tau_0 = 4.78Pa$.

The stopper radius was calculated with the equation:

$$r_0 = \frac{2 \cdot \tau_0}{\frac{\Delta p}{l_T}} \tag{13}$$

Physical length of the pipe is l = 4.8 m, and the equivalent total length l_T=$5.75m$. It is obtained $r_0 = 0.45mm < \frac{H}{2}$, which is a condition of flow.

It was considered that the local hydraulic resistance represents 20% of linear resistance [6]. Linear section is $A = B \cdot H_2$. From the calculation results the average speed values ($v_m = 1.57 \frac{m}{s}$) and maximum speed ($v_{max} = 2.36 \frac{m}{s}$), flow rate ($\dot{V} = 7.85 \cdot 10^{-4} \frac{m^3}{s}$) and pressure drop ($\Delta p = 3.05 \cdot 10^4 Pa$).

Equivalent hydraulic radius of the transition section is:

$$r_h = \frac{A}{P} = \frac{H_2 \cdot B}{2(B + H_2)}, \tag{14}$$

where P is the "wetting" perimeter. Making substitutions will result $r_h = 2.38mm$.

It is calculated the Reynolds number:

$$R_e = \frac{v_m \cdot 4 \cdot r_h}{\nu_m} = \frac{\rho_m \cdot v_m \cdot 4 \cdot r_h}{\eta_m}, \tag{15}$$

where ν_m is the coefficient of kinematic viscosity:

$$\nu_m = \frac{\eta_m}{\rho_m} . \tag{16}$$

Result from replacement: $R_e = 2050 < 2320$ (laminar motion).

Fluid mass flow rate is: $\dot{m} = \rho_m \cdot \dot{V} = 0.86 \frac{kg}{s}$.

Density of the fluid based on the temperature will be: $\rho_{input} = 1064.9 \frac{kg}{m^3}$ ($T = 95^\circ C$), $\rho_{output} = 1108 \frac{kg}{m^3}$ ($T = 30^\circ C$), $\rho_{medium} = 1086.5 \frac{kg}{m^3}$.

The heat transferred through the heat exchanger will be:

$$Q = \dot{m} \cdot c_m \cdot \Delta T = \dot{m} \cdot c_m \cdot (T_i - T_e), \tag{17}$$

where T_i and T_e are input and output temperatures, and c_m is the average specific heat of the water.

Replacing values into the formula: $c_m = 0.581 \frac{kcal}{kg \cdot grd}$, $\Delta T = 95 - 30 = 65^\circ C$, results: $Q = 33.2 \frac{kcal}{s} = 137W$.

In the second case, the thermal agent L_2 has $\rho = 1100 \frac{kg}{m^3}$. After the calculations, the values are: $\delta_s = 0.3mm$; $\tau_0 = 3.24Pa$; $\nu = 10st$.

It requires that: $v_m = 0.15 \frac{m}{s}$, $v_{max} = 0.2 \frac{m}{s}$, $A = 5 \cdot 10^{-4} m^2$, $r_h = 4.55mm$ and obtained value of Reynolds number is $R_e = 36.4$, specific to a very slow movement, laminar. The resulting volumetric flow rate is $\dot{V} = 4.5 \frac{l}{min} = 0.75 \cdot 10^{-4} \frac{m^3}{s} = 0.27 \frac{m^3}{h}$. The pressure drop $\Delta p = 2.68 \cdot 10^5 Pa = 2.68bar$, must be done by a displacement pump, developing power of $P = \dot{V} \cdot \Delta p = 0.75 \cdot 10^{-4} \cdot 2.68 \cdot 10^5 = 20W$.

In the third case, thermal agent L_1, by cooling, provides a significant temperature drop. Because of the fluid density variation with temperature, appears the thermo-siphoning phenomenon that helps pump for forced circulation in closed circuit with recirculation [7]. The pressure drop through thermo-siphoning can be calculated by the relation:

$$\Delta h = h_i - h_e = \frac{p_{at}}{g} \left(\frac{1}{\rho_i} - \frac{1}{\rho_e} \right), \tag{18}$$

where ρ_i and ρ_e are density of the liquid L_1 on entry and outlet, h is the height gauge pressure, and g is the gravitational acceleration.

For atmospheric pressure was adopted value $p_{at} = 1bar = 10^5 Pa$.

The pressure drop is:

$$\mathbf{p_{at}} = \rho_\mathbf{m} \cdot \mathbf{g} \cdot \Delta \mathbf{h}, \qquad (19)$$

where $\Delta h = \frac{10^5}{9.81}\left(\frac{1}{1064.9} - \frac{1}{1108}\right) = 0.377 m.col.liquid$.

Results $\Delta p = 0.377 \cdot 1086.5 \cdot 9.81 = 4000 Pa$, which represents 13.1% from the pump discharge pressure.

IV. Transfer Of The Heat Flow

The heat flow $Q = 33.2 \frac{kcal}{s} = 1.23 \cdot 10^5 \frac{kcal}{h}$ transfer shall be made from the liquid thermal agent L_1, to:

- Liquid L_2 that flows through the inner tube, through a heat exchange surface $A_{12} = 0.96 m^2$ (in a proportion of 60%, therefore $Q_2 = 0.738 \cdot 10^5 \frac{kcal}{h}$), with an overall heat transfer coefficient $k_{12} = 7.7 \cdot 10^4 \frac{kcal}{m^2 h}$;
- External cooling air driven by a fan through a heat exchange surface $A_{aer} = 0.96 m^2$ (in a proportion of 40%, therefore $Q_{aer} = 0.492 \cdot 10^5 \frac{kcal}{h}$), with an overall heat transfer coefficient $k_{aer} = 5.1 \cdot 10^4 \frac{kcal}{m^2 h}$.

To intensify the heat exchange can take constructive steps: providing the inner tube of the agent L_2 (in countercurrent with agent L_1) with turbulence promoters. On the outside of the tube where flows agent L_1, will provide cooling fins for cooling the heat exchange surface. Movement of fluid may be "forced" by increasing the flow rate of liquid and air.

If the goal is not only the heat exchanger cooling of agent L_1 (radiator) but also the heat transfer to the agent L_2 (recuperative), then double pipe outer wall, in contact with air, must be better insulated and dropped the fan.

V. Conclusion

Heat exchanger with plan partitions offers large areas of heat transfer and therefore are effective having small dimensions and weights. When using low flow rates and velocities of liquid viscous fluids, flows are generally slow, laminar, of Hagen-Poiseuille type for Newtonian fluids and viscous plastic (Bingham), for non-Newtonian fluids (with threshold or with stopper). Hydraulics for these fluids, especially through the ducted pipe with annular or rectangular sections, as in this case, is the most complex, unusual and little studied.

This paper, theoretical and applied pleads for equipment with surface heat exchangers between viscous fluids, with rectangular passing sections, and overcome the obstacles of difficult hydraulic.

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