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HORIZONTAL MICRO-CARRIER OF HEAT AND MASS

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Abstract **—** In this paper it is presented the structural scheme and analytical background of a hydrothermal micro-carrier of heat and mass, horizontal, opened, by thermo-aeration. This system transfers heat and mass with fluid flow rate conserving and can be implemented on the cooling system of internal combustion engines. The fluid has natural circulation, but in case when that solution is short of satisfactory, it can be insert a forced cooling with a pump or with an air blower. As an application it is analyzed the cooling system of an engine with natural circulation by thermo-aeration.

Keywords **—** heat, pipe, pump, water.

I. INTRODUCTION

IN Fig. 1. is showed the scheme of a thermo-hydraulic circuit (opened to atmosphere), witch transfers water and heat from a source A, in horizontal plane, to a heat consumer B without pump, on natural way, by thermo-aeration.

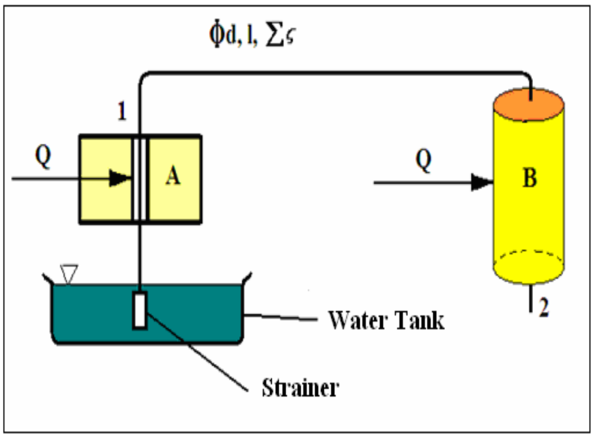

Fig. 1. The functioning scheme of the micro-carrier

The heat source A can be a secondary industrial source, like exhausted gases, steam or hot water. Also can be an assembly or an engine that must be cooled or a natural source like thermal water, lava, hot sand and stones or steam.

The could object B can be a heat consumer (hot water, hot air, air conditioner), heat exchanger or cooler.

II. THEORETICAL CONSIDERATIONS

The pressure difference witch ensure the water circulation and heat transfer, flux Q, from A to B is the result of mass density difference between cold and hot water. Because of heating, the water mass density decreases: the flow rate $\dot{V}$, or weight rate $\dot{m} = \rho \cdot \dot{V}$, is driven through a pipe with diameter d and length l and cooled through natural thermal transfer. For heat flux conservation, on driving route, the pipe must be isolated, the process being isotherm.

The water circulation is made between temperatures $t_1 = 95(^\circ C)$ and $t_2 = 30(^\circ C)$.

The water lifting from the water source through the strainer (filter and one way valve) and the heat source A it's compensated with the dropping through object B, resulting the level difference being near to zero.

The local lost head in pipes are approaching near to 20% from linear lost head. The cinematic viscosity has the medium value $\upsilon = 0{,}478 = const.$ (for temperature 60ºC).

The manometric head variation of pressure (suction) is [1]:

$$\frac{p_1}{\rho_1 \cdot g} - \frac{p_2}{\rho_2 \cdot g} = \sum h_{r1,2} \quad (1)$$

Where: p_1 and p_2 are pressures according to points 1 and 2 on the route, $g = 9{,}81\left(\frac{m}{s^2}\right)$ is gravity acceleration and $\sum h_{r1,2}$ is the sum of lost head through resistance to the flow.

$$\sum h_{r1,2} = 1{,}2 \cdot \lambda \cdot \frac{l}{d} \cdot \frac{v^2}{2g} \quad (2)$$

The water speed through the pipe is:

$$\mathbf{v}=\frac{\dot{\mathbf{V}}}{\mathbf{A}}=\frac{\dot{\mathbf{V}}}{\frac{\pi\cdot \mathbf{d}^2}{4}} \quad (3)$$

Where: λ is the linear lost head coefficient through viscous friction, stated in function of inner walls of pipe quality and the flowing regime given by the Reynolds number [2]:

$$\mathbf{R_e}=\frac{\mathbf{v}\cdot\mathbf{d}}{\upsilon} \quad (4)$$

For streamline regime ($R_e < 2320$):

$$\lambda=\frac{64}{\mathbf{R_e}} \quad (5)$$

And for turbulent regime it is recommended the Blasius equation:

$$\lambda=\frac{0{,}3164}{\mathbf{R_e^{0,25}}} \quad (6)$$

With assumption $p_1 = p_2 = p_{at} = 10^5 (Pa)$, is obtaining the effective pressure equal to the lost head sum [3]:

$$\frac{\mathbf{p_{at}}}{\mathbf{g}}\cdot\left(\frac{1}{\rho_1}-\frac{1}{\rho_2}\right)=1{,}2\cdot\lambda\cdot\frac{\mathbf{l}}{\mathbf{d}}\cdot\frac{\mathbf{v}^2}{2\mathbf{g}} \quad (7)$$

The mass density at water temperature t_1 and t_2 is $\rho_1 = 960\left(\frac{kg}{m^3}\right)$ and $\rho_2 = 992\left(\frac{kg}{m^3}\right)$.

From equations (6) and (7), assuming the streamline regime, the pipe diameter is:

$$\mathbf{d}=\sqrt[4]{\frac{2\pi\mathbf{g}\cdot\frac{\mathbf{p_{at}}}{\mathbf{g}}\cdot\left(\frac{1}{\rho_1}-\frac{1}{\rho_2}\right)}{1{,}2\cdot 64\cdot 4\cdot\upsilon\cdot\mathbf{l}\cdot\dot{\mathbf{V}}}} \quad (8)$$

It can be define a flow rate coefficient, dimensionless:

$$\mu=\frac{1}{\sqrt{\alpha+\lambda\cdot\frac{\mathbf{l}}{\mathbf{d}}+\sum\varsigma}} \quad (9)$$

Where α is Coriolis coefficient, witch can be assumed in this application equal to 1.

III. Application

For pipe's length $l = 109\ (m)$ and the fluid speed $v=0{,}117\ (m/s)$, are obtaining: $R_e = 2450$; $\lambda = 0{,}0452$ (with Blasius equation); $d = 0{,}01\ (m)$;

$$\dot{V} = 0{,}117\cdot\frac{\pi}{4}\cdot 10^{-4} = 0{,}915\cdot 10^{-5}\left(\frac{m^3}{s}\right) = 0{,}55\left(\frac{l}{min}\right).$$

The transferred heat flux it was computed with equation:

$$\mathbf{Q}=\dot{\mathbf{m}}\cdot\mathbf{c_m}\cdot\Delta\mathbf{t}=\rho_{\mathbf{m}}\cdot\dot{\mathbf{V}}\cdot\mathbf{c_m}\cdot(\mathbf{t_1}-\mathbf{t_2}) \quad (10)$$

Where: medium mass density of water is $\rho_m = 975\left(\frac{kg}{m^3}\right)$, the difference of temperature $\Delta t = (95-30) = 65(^\circ C)$ and the medium thermal capacity $c_m \cong 1\left(\frac{kcal}{kg\cdot grd}\right)$.

For a small length of pipe, l = 55 (m) and speed $v=0{,}163\ (m/s)$ it is obtaining: $\dot{V} = 0{,}765\left(\frac{l}{min}\right) = 46\left(\frac{l}{h}\right)$.

Over the natural circulation, in case when that is short of satisfactory, it can be insert a pump circulation of cooling water or a forced cooling with an air blower (like the cooling system of the internal combustion engines) [4].

Through the presence of that helping elements it can be admitted a lower value for temperature t_2, a highest speed in the driving pipe, a higher flow rate, a higher heat flux, a higher transport length, the possibility to secure level differences different to zero, $\Delta z \neq 0$ and lower weights for heat exchangers.

IV. Heat and Mass Transfer with Flow Rate Conservation

A system that can be used is a hydro-thermal system of heat and mass transfer with flow rate conservation. The scheme for this system is showed in Fig. 2.

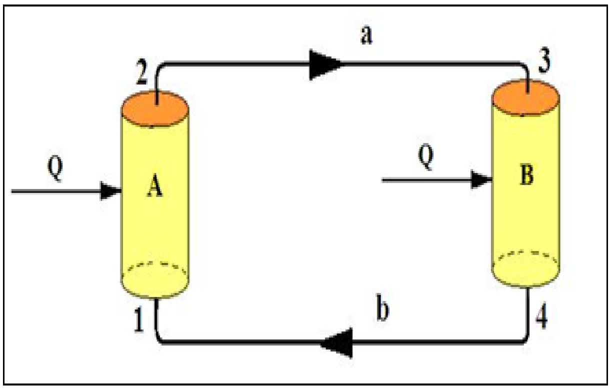

Fig. 2. Hydro-thermal system of heat and mass transfer with flow rate conservation

Where the notations are:
A – warm object, the heater of hydrothermal fluid;
B – cold object, the cooler, heat consummator;
a – high pressure pipe, warm, isothermal;
b - low pressure pipe, cold, isothermal.

The hydrothermal system transfers heat flux from object A to object B. The missing of a conventional pump in the circuit reduces the performances obtained through thermo-aeration. The hydrothermal agent can be water or water with ethyl glycol. It will be assumed the following parameters:

The manometer height:

$$\mathbf{h} = \frac{\mathbf{p}}{\rho \cdot \mathbf{g}} \tag{11}$$

The variation $\Delta h = h_2 - h_1\,(m.col.liquid)$, when the mass density decreases in object A because of the heating [5]:

$$\Delta \mathbf{h} = \frac{\mathbf{p}_2}{\rho_2 \cdot \mathbf{g}} - \frac{\mathbf{p}_1}{\rho_1 \cdot \mathbf{g}} > 0 \tag{12}$$

In object B, through worming, the mass density increase and $\Delta h < 0$.

The fluid speed is:

$$\mathbf{v} = \sqrt{2\mathbf{g} \cdot \Delta \mathbf{h}} = \sqrt{2\left(\frac{\mathbf{p}_2}{\rho_2} - \frac{\mathbf{p}_1}{\rho_1}\right)} \tag{13}$$

The flow rate in pipe:

$$\dot{\mathbf{V}} = \mathbf{A} \cdot \mathbf{V} = \frac{\pi \mathbf{d}^2}{4} \cdot \sqrt{2 \cdot \left(\frac{\mathbf{p}_2}{\rho_2} - \frac{\mathbf{p}_1}{\rho_1}\right)} \tag{14}$$

And the pressure:

$$\mathbf{p}_2 = \left(\lambda_a \cdot \frac{\mathbf{l}_a}{\mathbf{d}} + \xi_a + \xi_B\right) \cdot \frac{\rho_2 \cdot \mathbf{v}^2}{2} + \left(\lambda_b \cdot \frac{\mathbf{l}_b}{\mathbf{d}} + \xi_b + \xi_A\right) \cdot \frac{\rho_1 \cdot \mathbf{v}^2}{2} \tag{15}$$

Where: λ_a şi λ_b are the linear lost head coefficients in function with pipe lengths l_a şi l_b. The local lost head coefficients are ξ_a and ξ_b.

At the entering in abject A, the pressure is:

$$\mathbf{p}_1 = \xi_A \cdot \frac{\rho_1 \cdot \mathbf{v}^2}{2} \tag{16}$$

The pressure loss in object B:

$$\Delta \mathbf{p}_B = \xi_B \cdot \frac{\rho_2 \cdot \mathbf{v}^2}{2} \tag{17}$$

The pressure loss in object A:

$$\Delta \mathbf{p}_A = \xi_A \cdot \frac{\rho_1 \cdot \mathbf{v}^2}{2} \tag{18}$$

As an application it will analyze the cooling system of an engine with natural circulation by thermo-aeration [6].

If the cooling flow rate of the engine is $\dot{V} = 14\left(\frac{l}{min}\right)$, is necessary to compute the value of the pipe diameter and the radiators number of pipes. The fluid circulation is under temperatures $t_1 = 95(^\circ C)$ and $t_2 = 40(^\circ C)$.

The medium speed of water in radiator's pipes is $v_m = 0{,}15\left(\frac{m}{s}\right)$ and pipe's length is $l = 0{,}4(m)$. Also the speeds $v_1 = v_2$ and heights $z_1 \approx z_2$:

$$\frac{\mathbf{p}_1}{\rho_1 \cdot \mathbf{g}} - \frac{\mathbf{p}_2}{\rho_2 \cdot \mathbf{g}} = \sum \mathbf{h}_{r1,2} \tag{19}$$

For $p_1 = p_2 = p_{at} = 10^5(Pa)$ (absolute pressures), it is obtaining the real overpressure:

$$\frac{\mathbf{p}}{\mathbf{g}} \cdot \left(\frac{1}{\rho_1} - \frac{1}{\rho_2}\right) = 1{,}2 \cdot \mathbf{n} \cdot \lambda \cdot \frac{\mathbf{l}}{\mathbf{d}} \cdot \frac{\mathbf{v}^2}{2\mathbf{g}} \tag{20}$$

The water mass density at t_1 and t_2 is $\rho_1 = 960\left(\frac{kg}{m^3}\right)$ and $\rho_2 = 992\left(\frac{kg}{m^3}\right)$. The continuity equation is:

$$\dot{\mathbf{V}} = \mathbf{n} \cdot \frac{\pi \mathbf{d}^2}{\mathbf{v}} \cdot \mathbf{v} \tag{21}$$

For the following parameters it will be used the values: $R_e = 1{,}565 \cdot 10^3$, $\lambda = 0{,}041$, $p_1 = 2{,}04 \cdot 10^5(Pa)$, $p_2 = 1{,}04 \cdot 10^5(Pa)$, $\lambda \cdot \frac{l}{d} = 263\,(m.\,H_2O)$, $l = 32(m)$, $\sum \xi = 52{,}6$, $\xi_a = \xi_b = 6{,}3$, $\xi_A = \xi_B = 20$.

The calculated diameter *d* of radiator pipe *d* is 5 mm, and the pipes number $n = 80$.

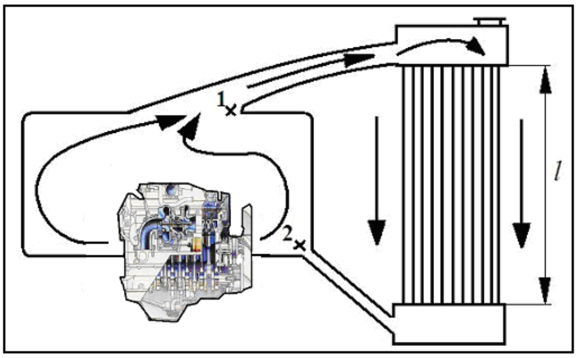

Fig. 3. Cooling system for the internal combustion engine

For a forced cooling with an air blower, the temperature $t_2 = 30(^\circ C)$, and for using a pump, the water speed is $v = 1\left(\frac{m}{s}\right)$. The number of pipes is lower, n = 12 pipes, the radiator being much smaller.

The obtained heat flux is:

$$\mathbf{Q = 960 \cdot 2{,}33 \cdot 10^{-4} \cdot 0{,}9973 \cdot (95-40) = }$$
$$\mathbf{= 12{,}3\left[\frac{kcal}{s}\right] = 4{,}43 \cdot 10^4} \quad (22)$$

Where the water thermal capacity at *40 (ºC)* is $c_m = 0{,}9973\left(\frac{kcal}{kg \cdot grad}\right)$, water mass density at *95 (ºC)* is $960\left(\frac{kg}{m^3}\right)$, the flow rate $2{,}33 \cdot 10^{-4}\left(\frac{m^3}{s}\right)$ and the difference of temperature in radiator is $95 - 40(^\circ C)$. The exhausting power in point 2 (Fig. 3) is *0,88(W).*

V. CONCLUSION

The opened transport system, without recirculation, analyzed and computed in this study, can transfer a heat flux, $Q = 0{,}582\left(\frac{kcal}{s}\right)$, that shows , with *1 (kcal) = 4,187 (kJ), 1(J) = 1 (Nm),* a small thermal power of *2,42 (kW),* on a thermal isolated pipe, with length *l = 109 (m)* and diameter *d = 0,01(m)*, with the level difference $\Delta z \approx 0$.

In case of thermo-aeration engine cooling, l is the total length of pipes from the cooling radiator, having *d = 5 (mm)*, with radiator height of 0,4 (m), results, after the calculations 80 pipes with a cooling water flow rate $\dot{V} = 14\left(\frac{l}{min}\right)$, with temperatures $t_1 = 95(^\circ C)$ and $t_2 = 40(^\circ C)$, with recirculation.

In Fig. 4 is showed a cooling system in closed circuit for an internal combustion engine. Over the natural circulation it is insert a forced circulation of cooling water with an axial air blower [7].

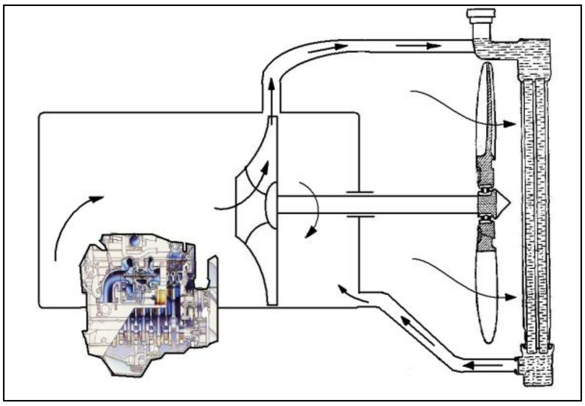

Fig. 4. Cooling system in closed circuit for an internal combustion engine

Conserving the water flow rate, the pipes diameter and lengths, bringing down the temperature, $t_2 = 30(^\circ C)$, it was obtain a much lower number of pipes (12 pipes).

In case of forced cooling, the radiator for heat transfer it's smallest over the radiator which running with natural circulation of cooling liquid (for the same flow rate) [8].

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