CONVECTIVE HEAT TRANSFER THROUGH COMPACT PLATE AND FINS HEAT EXCHANGERS

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Rezumat. *Această lucrare prezintă analiza transferului termic convectiv prin zece schimbătoare de căldură compacte cu aripioare ondulate, sinusoidale, de diferite tipodimensiuni, care au fost testate în condiţii reale de funcţionare. Se urmăreşte influenţa lungimii canalelor de aer şi a pasului transversal al aripioarelor ondulate sinusoidal asupra transferului termic convectiv. De asemenea, sunt redate ecuaţiile criteriale deduse de către autor pentru invarianţii Nusselt şi Colburn specifici acestor aparate* .
Cuvinte cheie. *Schimbătoare de căldură cu plăci, aripioare ondulate sinusoidal, convecţie forţată, coeficient de convecţie, randamentul nervurii, criteriul Nusselt, invariantul Colburn.*

1. INTRODUCTION

Compact plate and fins heat exchangers are made by a mixture between two layers composed of profiled wings separated by aluminium plates (fig.1), which permits a cross flow for the thermodynamic agents in two ways:

- through two consecutive plate tables closed by bars;
- through the channels formed by two plate tables and a fin that covers the entire surface (fig. 2).

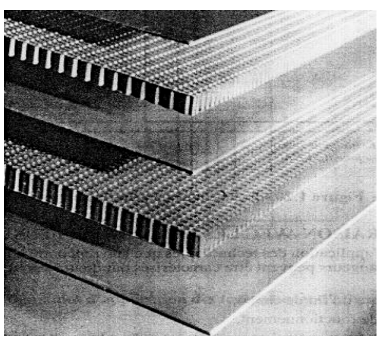

Fig. 1. The principle of the plate and fins compact heat exchangers **[1]**.

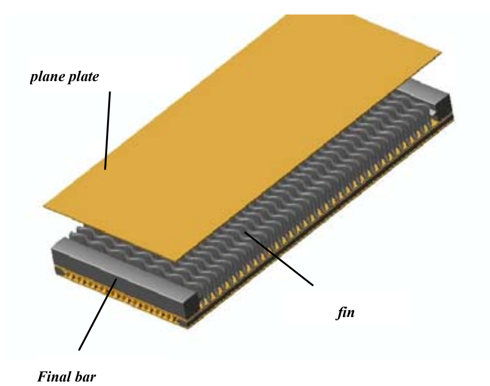

Fig. 2. The construction of a plate and fins heat exchanger **[2].**

The results presented in this paper are based upon the values obtained after the research made on 10 compact aluminium plate and fins heat exchangers with different lengths of the air channel L_{air}. L takes values between 30, 45, 65, 95 and 115 mm. The cross pitch of the fins is p_n=4 and p_n= 6,5mm.

2. CRITERIA EQUATIONS FOR CONVECTIVE HEAT EXCHANGE

The convective air coefficients were found using a mathematical model developed by the author based on Engineering Equation Solver software.

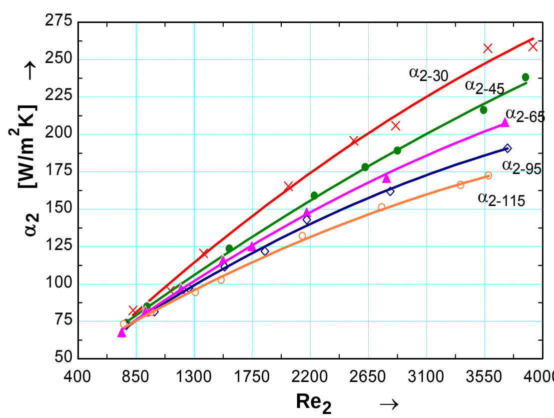

Fig. 3. Air convective coefficients dependent of Reynolds number for p_n = 4 mm and different L_{air} [3].

The values of (α_2) are almost the same in laminar flow (Re_{2c} <1000) for all the pitches and all the length of the air channel. This phenomena can be explain by the low values of the thermal conductive resistance in comparison with the thermal convective resistance

For transition flow (1000<Re_{2c}<2000), the convective coefficients are continuously growing as Re_{2c} grow. For Re_{2c}>2000 convective coefficients are growing and the importance of the convective resistance is less important (fig.3). For p_n = 4 mm and

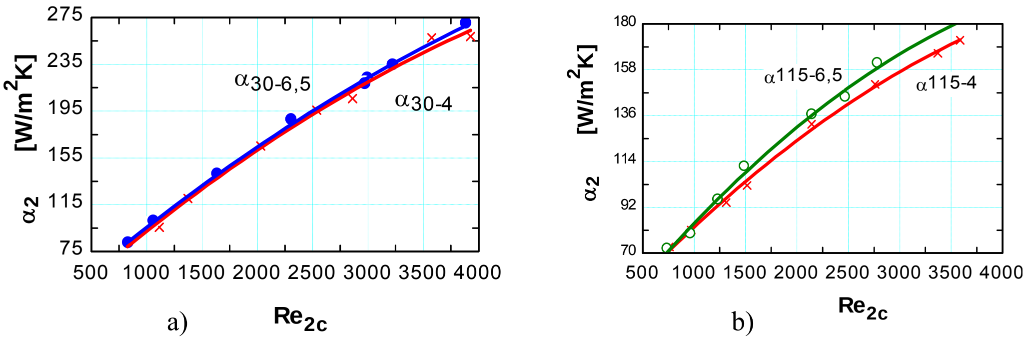

Fig. 4. Air convective coefficients dependent of Reynolds number for:
$a - L_{aer}$ = 30 mm and $b - L_{aer}$ = 115 mm [3].

also for p_n = 6,5 mm the convective coefficient (α_2) is decreasing as the length of the air channel is increasing. For p_n = 4 mm the air convective coefficient takes values between (82 ÷ 259) W/m^{2}K, for L_{aer} = 30 mm and (73 ÷ 172) W/m^{2}K for L_{air} = 115 mm.

The flow can be considered to be two-dimensional, having the axes: $x = L_{air}$ and $y = p_n$. The velocity vector has also two components, one on x direction and the other on y direction, which gives different velocities for the fluid layers. The fluid jets interact on the main directions and generate vortexes. High turbulence and the change of the velocity distribution in the layers caused by the sinusoidal shape of the channel create good condition for a strong convective heat exchange. As the channel length is increasing the main jets velocity and the vortexes are decreasing which makes the convection to decrease as L_{air} increases.

In order to study the influence of the pitch p_n upon the air convective coefficient apply to the same dimension L_{air} apparatus the diagrams from figure 4 were established. The minimum and maximum values for L_{air} were used in order to make a comparison with the same exchangers.

The difference between the corresponding p_n = 6,5 mm and p_n = 4 mm coefficient for high L_{air} is high and decrease as the values of L_{air} are decreasing. Rising the values for the pitch has no influence when Re < 1000.

The values of the efficiency η_2 of the studied heat exchangers, having narrow surfaces are (0,927 ÷ 0,9854), whatever the fin pitch is. This explained the fact that the fins surfaces intensify the heat exchange (fig. 5). These values are smoothly rising as the air channel length is also rising.

The maximum values of the narrow surface efficiency (η_2) and for fin efficiency (η_{n2}) correspond to minimum values of the air flow through the channel and to minimum convective coefficients, while the minimum values correspond to the maximum velocities. This aspect is caused by the temperature variation along and also across the fin.

The dependence between the fin efficiency η_{n2} on $\left(m \cdot h_{cr}\right)$, where m is the fin thermal charge and h_{cr} is the critical fin height shows that it matches with the dependences given by the literature.

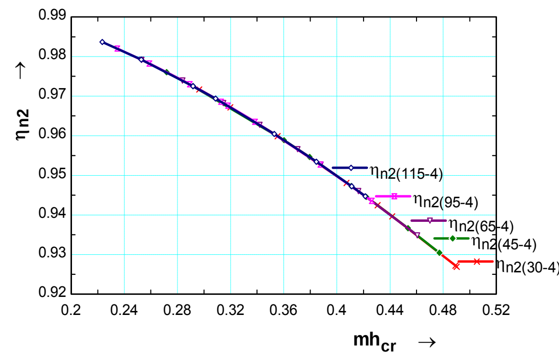

Fig. 5. Fin efficiency variation η_{n2} function of the thermal charge (m).

The calculus of the efficiency coefficient of the thermal convective exchange can be made only knowing the values of the convective coefficient, Nusselt. For all types of heat exchangers studied, it can be seen in figure 6 that Nu grow almost linear as Reynolds increases, due to thermal convective flux (α_2).

For low Reynolds numbers, in laminar flow there is an influence of the resistance on the thermal transfer upon the choose length (d_{2ech}), which makes Nu_2 values to be pretty the same to all heat exchangers, whatever the fin pitch is. The influence of the resistance becomes very important when Re>1000. For a given L_{air} Nusselt is under the influence of the convective flux (α_2). In this case it has a similar behaviour, increasing as Reynolds increases. When Reynolds is constant and different lengths L_{air}, Nusselt criteria decreases as the air length is increasing, whatever the pitch value p_n is (fig. 6 a şi b).

The dependence between Nusselt and Reynolds makes the authors tried to find criteria relations that express the variation between Nu_2 and Re_2 on geometrical dimensions of the heat exchangers [2]. Based on experimental researches the equations determined by the authors are:

– p_n = 4 mm,

$$\mathrm{Nu}_{2c} = C \cdot \mathrm{Re}_{2c}^{x} \cdot \mathrm{Pr}^{1/3} \cdot \left(\frac{d_{2ech}}{L_{d-aer}}\right)^{y} \cdot \left(\frac{p_n - g_n}{h_{cr}}\right)^{-0.32} \quad (1)$$

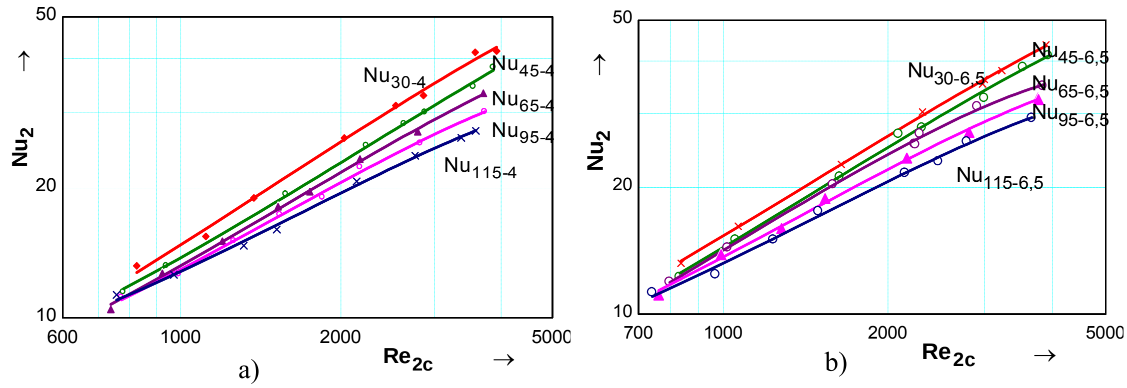

Fig. 6. The variation of Nusselt criteria funcion of Reynolds for different pitches:
$a - p_n = 4$ mm, $b - p_n = 6{,}5$ mm [3].

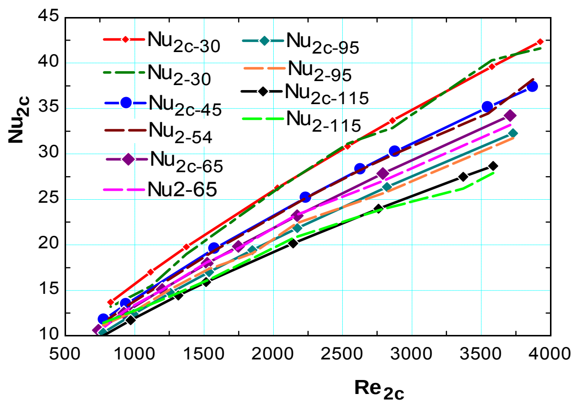

Fig. 7. Nusselt criteria for $p_n = 4$ mm [3].

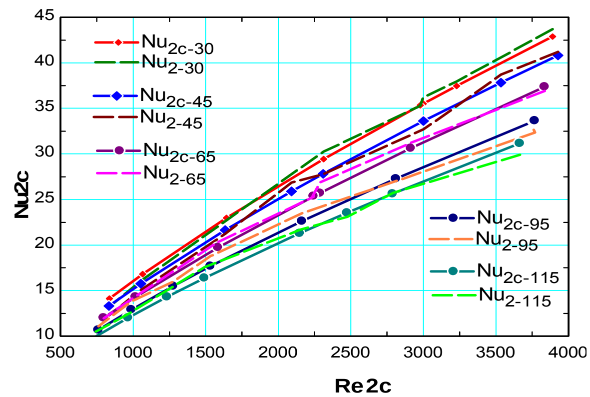

Fig. 8. Nusselt criteria for $p_n = 6{,}5$ mm [3].

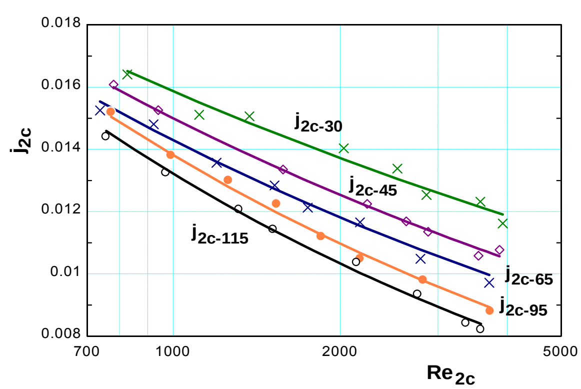

Fig. 9. Colburn function for $p_n = 4$ mm

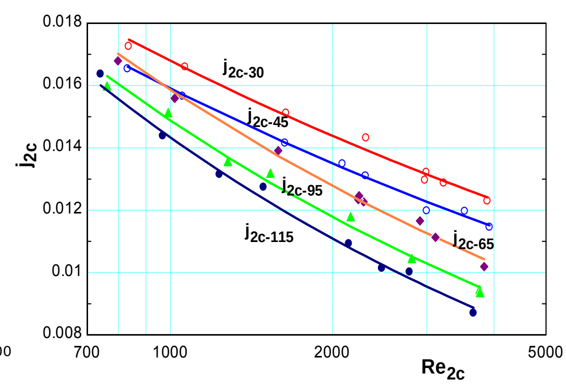

Fig. 10. Colburn function for $p_n = 6{,}5$ mm [3].

Turn

$- p_n = 6{,}5$ mm,

$$\mathrm{Nu}_{2c} = C \cdot \mathrm{Re}_{2c}^{x} \cdot \mathrm{Pr}^{1/3} \cdot \left(\frac{d_{2ech}}{L_{d-aer}}\right)^{y} \cdot \left(\frac{p_n - g_n}{h_{cr}}\right)^{-0.66} \qquad (2)$$

where C, x şi y are constant and have values depending of the heat exchanger type with an relative error under 3,5 %. The curves obtained using equation (1) and (2), are presented in figure 7 (for a 4 mm pitch) and in figure 8 (for a 6.5 mm pitch).

The literature presents different methods of comparison of the thermal performances of the surfaces using Colburn factor, which gives the measure of the intensity of the thermal transfer using forced convection.

In figure 9 and 10 the dependences between Colburn and Reynolds criteria for a 4mm pitch (fig.9), respectively for a 6,5 mm pitch (fig.10) are presented. The increasing of the velocity drives to decreasing $\left(\frac{\mathrm{Nu}_2}{\mathrm{Re}_2}\right)$, because the small values of the equivalent diameters and kinematical viscosity.

Stanton function depends directly proportional on this ratio and reverse proportional with Prandtl number, which has small variations for (0,76...0,8). This drives to the decrease of the Stanton number for (0,02…0,01), as the velocity increases. This tendency is valid even for ($Pr_2^{2/3}$), in the same velocity conditions. From mathematical point of view Colburn function depends directly proportional with $\left(St \cdot Pr^{2/3}\right)$. Almost all the heat exchangers having extended surfaces are designed based on experimental researches and simple equations. The authors developed a equation valid for the heat exchangers tested:

$$j_{2c} = a \cdot Re_{2c}^{b} \qquad (3)$$

where ***a*** and ***b*** are correlation coefficients and their values are presented in table 4, for a constant pitch p_n and different L_{air}.

Table 4

Constants a and b for p_n = 4 mm [3]

	L30-4	L45-4	L65-4	L95-4	L115-4
a	0,067522	0,0895681	0,0955	0,137	0,154411
b	-0,2097	-0,2587	-0,275	-0,425	-0,355826
Er_j	< 2,7 %	< 2,2 %	< 2,8 %	< 2,3 %	< 2,9 %

The influence of the narrow fins cross pitch upon the thermal transfer coefficient **j** ($j_2 = f(Re_{2c})$) is presented in figure 11. It is shown that the heat exchangers having 6.5 mm pitch have superior j value than those with 4 mm pitch.

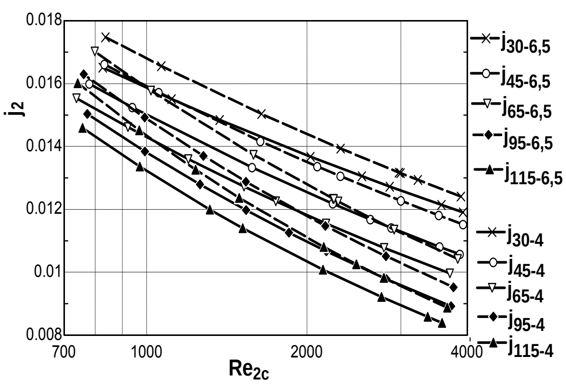

Fig. 11. Variation of Colburn function for different p_n.

This means that in heat exchangers with 6.5 mm pitch the convective heat transfer is better. Using the equation (3) the comparison between the thermal performances for different types of fins can be easily made.

3. CONCLUSIONS

The results obtained by using the software application developed by the authors drive to the following conclusions:

- Convectiv phenomena is decreasing as the length of the channel increases (for L30_4 type, $\alpha_2 = (82 \div 259)$ W/m²K; for L115_4 type $\alpha_2 = (73 \div 172)$ W/m²K);
- For high values of L_{air}, the difference between convective coefficient corresponding to p_n=6,5mm and those corresponding to p_n=4mm is higher than for low values of L_{air}, (for laminar flow Re<1000 the value of the pitch has no influence);
- The efficiency of the narrow surface is η_2 = (0,927 ÷ 0,9854) and do not depend on the fin pitch, it increases as the air channel length is increasing;
- The efficiency of the fin (**η_{n2}**), takes values between (0,92 ÷ 0,98);
- The efficiency factor of the convective thermal exchange (Nu_2) has values almost the same for all types of heat exchangers, whatever the pitch value is (a better heat transfer was found for 6.5 mm fin than for 4 mm fin).

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