Analytical and numerical investigation of heat exchange in fin-tube refrigerating heat exchanger

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Abstract. Intensification by tube fin is intended to increase the heat exchange surface between the heat exchange surface and surrounding fluid. This method of intensification is applied to the heat exchangers type air coolers (refrigeration heat exchanger with indirect cooling). The interaction of cooled air in refrigeration equipment is most often a natural convection (quiet cooling). For the purposes of this work, a finned tube heat exchanger with a coolant a brine solution of CaCl₂ was viewed. The numerical studies for ribbed cross sections at three different heights were done. On the basis of the results obtained for heat flow in a rib and throughout the exchanger, the effectiveness of the apparatus was determined. The model studies in Flow Simulation were carried out. For the purpose of the simulations a geometric models of two elements from cooling fin - tubes at three different heights (h) of the ribs (10 mm, 20 mm and 30 mm) were made. Numerical studies to investigate the velocity and temperature distribution at different rib height were carried out.

1. Introduction
The intensification of heat transfer without phase transition has been directed to the main thermal resistance along the path of the heat flow - the boundary layer. Methods for intensifying convective heat exchange have been divided into two groups: intensive and extensive. Intensive methods refer to the possibilities for influence on the thickness and character of the flow in the boundary layer. The second group of methods have been applied to these cases and constructive solutions associated with the increase of the heat exchange surface by ribbing. Ribbing to intensify heat exchange on the side with higher thermal resistance (with a lower heat transfer coefficient) has been intended. Accurate demarcation between the two groups of intensification methods cannot be done. Therefore, the application of a specific principle of intensification has been required knowledge of heat transfer and hydraulic or aerodynamic technical features of the interaction of the fluid flows with the heat exchange surface.

In many cases, fluids streamlined the heat exchange surface have very different heat-physical properties, have under different pressure or have a different physical condition.

This is the reason to observe very different values of the two coefficients of convective heat exchange.

Thus, if one surface of the heat exchange wall has streamlined from a gas stream, the coefficient of heat transfer has usually not more than 100 W/(m²·K), while on the other hand, the liquid refrigerant having a low viscosity or in the condensation of water vapor coefficient of convection can to have one or even two orders of dimension higher [7, 8].
It has been proven that the ribbing has to be done on the side of the low heat transfer coefficient. That's why has been developed of a wide variety of ribbed surfaces - pipes with different cross-section of the pipe, flat or with other geometry.

The method of ribbing has been also different in terms of the geometry of the ribs and in the manufacturing methods.

Cooling fin - tubes may be by direct cooling (via evaporating refrigerant) and with indirect cooling (using a salt solution, etc.). They have been made of smoothness or ribbed pipes with a horizontal or vertical position. Smoothness pipes for cooling fin - tubes have been hardly used, as the thermal load of one linear meter of the pipe has been very low. This led to an increase in the length of the tubes and the amount of refrigerant required for their filling. Application has been mainly for cooling fin-tubes, in which the cost of the metal (tubes) has been decreased about three times and their dimensions have been considerably - small [3].

Ribs of heat exchange surfaces can be of a different profile (rectangular, round, triangular, etc.). Depending on their purpose, their design has been subject to different conditions: effective use of the material, maximum heat transfer, maximum mass, minimum dimensions, etc. The main characteristics that determine the efficiency of the rib and ribbed wall have been the distribution of the temperature at height and the heat flow. They have been depending on the profile, the shape of the cross section, the material, the dimensions and the boundary conditions of the heat exchange [3, 7, 8].

In the present work have been used analytical and model studies of heat exchanger with applied intensification method. As a particular type of the heat exchanger, cooling fin - tubes have been used to cool the air in cold-storage rooms (chambers) with free convection. They have been used mainly in the chambers for the storage of frozen foods at temperatures $t_f \leq -20 \, ^\circ C$ as auxiliary equipment. They can also be combined with air coolers [3].

2. Materials and methods

The investigated cooling fin-tubes are with following geometrical performance: $\varnothing$ 38x2; rib thickness – $\delta = 2$ mm; step between ribs – $s = 10$ mm; heights of the ribs ($h$) – 10 mm; 20 mm; 30 mm (figure 1). The pipe and the ribs have been made of the same material 304L(1.4307), with thermal conductivity $\lambda = 17$ W/(m·K) [10].

![Figure 1. 3D model in h=30mm.](image)

Water solution of CaCl$_2$ has been used for liquid refrigerant at temperature ($t_{f1} = -25 \, ^\circ C$) [5]. Numerical investigation were conducted at chamber temperature $t_{f2} = -20 \, ^\circ C$. The fin-tube
refrigerating heat exchanger has been used as an auxiliary element in a cooling chamber and therefore a small temperature difference was used. The temperature at the base of the ribs \( t_0 \) via applying a heat balance has been determined from:

\[
t_0 = \frac{\alpha_2 t_{f2} + \alpha_1 t_{f1}}{\alpha_1 + \alpha_2}
\]

where \( \alpha_i \) has been the heat transfer coefficient between liquid refrigerant and the inner side of the pipe, \( \alpha_1 = 1100 \, \text{W/(m}^2\cdot\text{K}) \) [3]; \( \alpha_2 \) has been the heat transfer coefficient between the ribs and the air in the refrigeration chamber, \( \alpha_2 = 3.6 \, \text{W/(m}^2\cdot\text{K}) \) [3]. Solving equation (1) has been received \( t_0 = -24.983^\circ \text{C} \).

The efficiency coefficient \( E \) has been used for comparison of the ribs. The same has been the ratio of the heat flow has been taken away from real rib \( Q_R \) and the heat flow \( Q_0 \) has been taken away from the sample rib under the same conditions and dimension.

\[
E = \frac{Q_R}{Q_0}
\]  

The heat flow of a real round rib \( Q_R \) has been determined by the following method:

- An existing rib (conventional rib on a plane surface) has been considered with length \( l = 1 \, \text{m} \), thickness \( \delta \), adjusted height \( h' \).

\[
h' = (r_2 - r_1) + 0.5\delta
\]

where: \( r_2 \) has been an outer radius of the rib, mm; \( r_1 \) - inner radius of the rib, mm.

The heat flow \( Q_{adj} \) of this conventional rib has been calculated according to the following equation:

\[
\dot{q}_{adj} = \frac{Q_{adj}}{F_{adj}} \approx \frac{\lambda \theta_0 m h (m h')}{2h'}
\]

where: \( f \) has been the fore-part surface of the rib, \( \text{m}^2 \); \( F_{adj} \) - the around surface of the conventional rib, \( \text{m}^2 \); \( \theta_0 \) - the temperature difference between the temperature of the fluid and the temperature in the base of the rib, K; \( m \) has been the parameter of the rib, \( \text{m}^{-1} \). The same have been calculated by following equations:

\[
\theta_0 = t_0 - t_{f2}
\]

\[
m = \sqrt{\frac{2 \alpha_2}{\lambda \delta}}
\]

The heat flow has been calculated for real round rib with the dependence:

\[
\dot{Q}_R = \varepsilon_r F_r \dot{q}_{adj}
\]

where \( \varepsilon_r \) has been the correction coefficient accounting the influence of the curvedness of the rib; \( F_r \), \( \text{m}^2 \) has been calculated the around surface for the real round rib with the dependence:

\[
F_r = 2\pi \left[ (r_2 + \delta)^2 - r_1^2 \right]
\]

The correction factor has been accounting by graphic dependencies via figure 2 [7], as a function of relations \( \frac{\theta_h}{\theta_0} \) and \( \frac{r_2}{r_1} \), \( \theta_h \) - the temperature difference between the temperature at the edge of the rib and the surrounding fluid

\[
\theta_h = t_h - t_{f2}
\]
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Figure 2. Correction coefficient accounting the influence of the curvedness of the rib.

The heat flow of the conventional rib $\dot{Q}_0$ has been determined according to the Newton equation:

$$ \dot{Q}_0 = \alpha_2 F_r \theta_0 $$

(10)

The rib is conventional, where the temperature on its entire surface is equal to and equal to that at the base.

Numerical studies to investigate the velocity and temperature distribution at different rib height were carried out. For the purpose of the simulations a geometric models of two elements from cooling fin-tubes at three different heights of the ribs (10 mm, 20 mm and 30 mm) were made (figure 1). To create the 3D models a software product SolidWorks was used. It has two CFD tools - FloXpress and FlowSimulation. There are studies of heat exchangers in FloXpress were carry out [4], but due to a number of limitations regarding the initial and boundary conditions of this tool, the numerical studies in the current work in FlowSimulation were carry out. For working fluid, a water solution of CaCl$_2$ was used. CaCl$_2$ is missing in the program library and therefore additionally was added (manually all thermophysical characteristics required for calculation were imported). A square mesh was used, as the specific locations a local mesh with a lesser density was created (Figure 3).

Figure 3. Mesh preview.
Studies of the solution’s independence by the density of the mesh were done. Two mesh densities (233173 and 451413 cell number) were studied, and the results obtained concluded that the optimal model for the study was that of 451413 cells. For the verification, the analytically calculated temperature at the base of the rib and that calculated by the program was adopted:

\[ d\dot{Q} = \alpha_2(t - t_f)udx = \alpha_2 \theta udx \]  

(11)

The resulting difference with the optimal mesh was 0.03 %. To close the system of partial differential equations for motion [1] the \(k-\varepsilon\) turbulence model was used [2, 6, 9]. The simulations on a computer with processor Intel Core I7 and 16 GB RAM were carried out. Performing a simulation with shown mesh takes approximately 1 hour.

3. Results and discussion

Analytical results were obtained for the thermal flow of the real rib \(\dot{Q}_R\) and the heat flow \(\dot{Q}_0\) take away from the conventional rib and the efficiency of the round rib.

Table 1. Efficiency of a round rib at different rib heights.

| \(\lambda\) | \(\alpha_1\) | \(\alpha_2\) | \(r_2\) | \(h^*\) | \(mh\) | \(\partial\theta/\partial x\) | \(\dot{Q}_{adj}\) | \(\dot{Q}_0\) | \(\dot{Q}_R\) | \(E\) |
|-----------|------------|-------------|--------|-------|-------|----------------|-------------|---------|---------|-----|
| W/m²K     | W/m²K      | W/m²K       | mm     | mm    |       | W          | W           | W       | W       | %   |
| 29        | 1100       | 3.6         | 29     | 11    | 0.160 | 0.987      | 0.391       | 0.610   | 0.051   | 97.2|
| 49        | 31         | 0.451       | 49     | 31    | 0.305 | 0.955      | 0.731       | 0.134   | 0.133   | 95.1|
| 17        | 1100       | 3.6         | 39     | 21    | 0.305 | 0.955      | 0.731       | 0.134   | 0.133   | 95.1|

On the basis of analytical studies were obtained graphic dependences of efficiency and heat flow from the heights of the ribs (Figure 4).

![Figure 4. Dependences of efficiency and heat flow from the heights of the ribs.](image)

The results were present that increasing the height of the rib increases the heat flow but reduces its efficiency.

Figures (5, 6, 7) shows the distribution of velocity at the three rib heights.
Figure 5. Distribution of velocity in \( h = 10 \) mm.

Figure 6. Distribution of velocity in \( h = 20 \) mm.

Figure 7. Distribution of velocity in \( h = 30 \) mm.
It is seen that with increasing height of the ribs, airflow velocity around the ribs between the two elements decreases, it can be argued that almost no natural circulation of air. Figures 8, 9 and 10 shows the temperature distribution of the first and last rib.

**Figure 8.** Distribution of temperature in $h = 10$ mm.

**Figure 9.** Distribution of temperature in $h = 20$ mm.

**Figure 10.** Distribution of temperature in $h = 30$ mm.

At $h = 30$ mm it is noticed that the temperature of a part of the surface of the ribs is close to that of the environment.
Figures 12, 13 and 14 shows the temperature distribution in length - L1 (Figure 11) for one element of the cooling fin - tubes.

**Figure 11.** Length (L1) preview.

**Figure 12.** Distribution of temperature for length (L1) in $h = 10$ mm.

**Figure 13.** Distribution of temperature for length (L1) in $h = 20$ mm.
Figure 14. Distribution of temperature for length (L1) in $h = 30$ mm.

Well known that at $h = 30$ mm the temperature of the air between the ribs is almost equal to their surface temperature, which is related to the lack of good natural convection also is associated with (figure 7).

4. Conclusion

Under the given conditions, the optimal height of the rib is $h = 20$ mm;

On the basis of the simulations it can be argued that the use of ribs with height $h = 30$ is inappropriate, which is also confirmed by the analytical results.

Modeling the transmission processes for this type of cooling fin - tubes can be successfully done using FLOW SIMULATION from the SOLID WORKS software.

The obtained results can be successfully used in the design, optimization and constructing of this type cooling fin - tubes, as well as in the educational process.

References

[1] Angelov M and Raynov P 2013 Model studies of the hydrodynamics of flow in corrugated tubes, *Journal of food and packaging Science Technique and Technologies*. vol 2

[2] Angelov M, Raynov P and Stoева D 2012 Possibilities for numerical modeling of the exchanged processes in a double-pass heat, (in Bulg.), *Science conference: Food science, engineering and technologies*. Plovdiv, 19-20 October, Scientific works, vol LIX, pp 768-773

[3] Dichev S 1993 Refrigeration machines, (in Bulg.). Sofia, Technics

[4] Hodževa Z and Rasheva V 2014 Analysis of fluid dynamics in shell and tube heat exchanger with floating head, (in Bulg.), *Scientific research of the Union of Scientists in Bulgaria-Plovdiv, series C. Technics and Technologies*, vol XII, Union of Scientists, Session 31 October – 1 November 2014

[5] Kimenov G 1995 Thermodynamic and Thermophysical Properties of Substances - Guidebook, (in Bulg.), Technics

[6] Launder B E and Spalding D B 1974 The numerical computation of turbulent flow. *Comp. Mech. In Appl. Mech. Engng*. 3, pp 269-289

[7] Raichkov G 2011 Heat and masstransfer, (in Bulg.), *Academic Publishing House of UFT*, Plovdiv

[8] Sendov S and Yordanov P 1981 Intensification of heat exchange, (in Bulg.), Sofia, Technics

[9] Stoeva D 2014 Intensification of the heat transfer through corrugated wall, *Journal of Faculty of Food Engineering*. vol XIII, Issue 1, pp 14 – 22

[10] http://www.matweb.com.