Effectiveness of fins formed by dimples in the form of ball segments

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Abstract. One of the famous ways to improve efficiency of a heat exchanger is associated with the topography of the surfaces being in contact with coolants. So, use of hemispherical dimples leads to progressive growth of the relative heat transfer coefficient compared to increase of the relative resistance coefficient. Usually a plate having the spherical dimple intensifiers for heat transfer is considered as a flat one with embedded cavities. However, such a plate can be also considered as the plate with inbuilt fins which are formed by dimples in the form of ball segments. Given that for the flow of fluid (gas) from left to right, the minimum local heat transfer enhancement occurs in the first (left) half of the dimples, and the maximum falls on the edge of the second (right) half, we obtained an analytical solution describing the temperature distribution along the height of the fin. In the solution we used the Harper-Brown approach. Presented are the results of the calculation of the efficiency of the surface on the parameters of the considered fin and on a known value of the average heat transfer coefficient corresponding to the stage of the fluid flow steady state.

1. Introduction

In the design of heat exchange equipment, various methods of heat transfer intensification are used: application of special devices for providing turbulence and rotated flows; application of rough surfaces, short channels and discontinuous surfaces [1].

Among the well-known intensifiers of heat exchange the spiral recesses (ledges) and hemispherical dimples have advantages due to their better thermal and hydrodynamic properties. The called intensifiers ensure rapid growth of the relative heat transfer coefficient compared to an increase in the relative resistance coefficient.

Studying of heat exchange surfaces of shaped dimples, was engaged in: Kiknadze, G. I. and Leontiev, A. I., Ligrani P. M. and Khalatov A. A., Gortyshov F. Yu. and Isaev, S. A., and Sergievsky, E. D. and Popov, I. A., Belenky, M. Ya. etc. Analysis of their works enabled to detect the following differences in the numerical and experimental investigations:

- use of different turbulence models, numerical methods, mesh types and grid generators;
- use of various means for supplying heat to surfaces having dimples (Fig. 1);
- use of schemes with different arrangement of dimples (corridor-shaped, chess-shaped);
- use of dimples with different parameters - depth, spot diameter (projection of the dimple on a plate surface), radius of curvature, sharp or smooth edges, density of the location;
- methods of embedding dimples - punching and milling.

![Image](image1.png)

Figure 1. Ways of supplying heat to the surfaces with dimples.

The main objective of these works was to study processes related to heat exchange and hydrodynamics on surfaces with embedded dimples and obtain empirical formulas for determination of heat transfer coefficient. The authors considered the test surface of the heat transfer from the point of view of the presence of dimples but fins formed by the spherical segments (see Fig. 2). Therefore, further in the present paper the test surface of the heat transfer is investigated from the perspective of just fins.

![Image](image2.png)

Figure 2. View of test surface with dimples (fins).

It is known that fins differ from each other in efficiency [6]. But in scientific literature there are only data on efficiency of the fin whose profile is described by a common function:

\[ f(z) = \frac{\delta_0}{2} \left( \frac{z}{h} \right)^{(1-2n)/(1-n)} \]

where \( \delta_0 \) - thickness of the fin at the base, \( h \) - fin height, \( n \) - parameter of the fin profile (rectangular, triangular, trapezoidal, parabolic, cylindrical and prismatic). There are analytical solutions describing the temperature fields and determining the optimal parameters of fins for different target functions. However, there is no information about the fins formed by dimples in the form of spherical segments. Therefore a real interest is formulation and solution of the tasks related to determination of efficiency of such fins.

2. Statement of the problem

Analysis of experimental works showed that the minimum local heat transfer enhancement occurs in the first (left) half of the dimples, and the maximum falls on the edge of the second (right) half, we obtained an analytical solution describing the temperature distribution along the height of the fin [2-3,7]. In addition, the temperature distribution and also distribution of the heat transfer coefficient in the dimple and its surroundings are symmetrical with respect to the direction of movement of liquid or gas (Fig.3).
Figure 3. Thermograms of the surfaces with dimples (a) [2-3] and the heat transfer coefficient (b) [7] on the surface with dimples (the movement of the liquid from top to bottom).

On the given surface we can select a repeating element which is similar in the geometric and thermo-physical sense to the same elements forming the heat exchange surface regardless of density of the dimples location. In this case, efficiency of the surface can be estimated from the efficiency of the repeating element, i.e. fin, formed by dimples in the form of ball segments. Obviously, such a procedure is valid only at the steady-state flow regime and heat transfer. The average heat transfer coefficient of this surface is equal to the average heat transfer coefficient of the specified fin and is defined by the formula:

\[
\bar{\alpha} = \frac{\alpha_1 \cdot F_1 + \alpha_2 \cdot F_2}{F_1 + F_2}
\]  

where: \(\alpha_1\) and \(\alpha_2\) – coefficients of heat transfer on the lateral surface of the fin (i.e. in the dimple) and in its top; \(F_1\) and \(F_2\) – area surfaces of the fins in contact with liquid (gas).

To determine distribution of the temperature field in such the fin we used the Harper-brown approach. For that reason the real fin was replaced with a virtual fin, the top of which is thermo-insulated. With that the lateral surface of the virtual fin was taken to be equal to the surface of the real fin participating in the heat exchange with the liquid. In addition, it is assumed that the heat transfer coefficients for both fins are identical and equal to the average heat transfer coefficient of the plate with dimples.

3. Mathematical model and results of calculations

The differential equation describing the temperature field of the fin over its height has the following form:

\[
\frac{d^2 \theta(Z)}{dZ^2} + \frac{2 \cdot \pi \cdot Z}{L_x \cdot L_y + \pi \cdot (Z^2 - 1)} \cdot \frac{d\theta(Z)}{dZ} - \frac{\pi \cdot B_i}{L_x \cdot L_y + \pi \cdot (Z^2 - 1)} \cdot \theta(Z) = 0
\]  

where: \(\theta(Z) = (t(z) - t_\infty)/(t_0 - t_\infty)\) - relative fin temperature;
\( t_0 \) и \( t_{\infty} \) - temperature of fin base and flow of gas or fluid accordingly, °C;
\( Q_z, Q_{z+dz} \) - heat flows on the surfaces of the fin elementary layer \( dz \) limited by surfaces with coordinates \( z \) and \( z+dz \), W;
\( F = f / R^2 = L_x \cdot L_y + \pi \cdot (Z^2 - 1) \) – relative area of the surface through which the heat flow \( Q_z \) passes;
\( dS = ds / R^2 = 2 \cdot \pi \cdot dZ \) – relative elementary surface which is involved in the convective heat exchange with the fluid in the dimple;
\( L_x = l_x / R \) – relative step between dimple centers in the direction of movement of the liquid (gas) flow;
\( L_y = l_y / R \) – relative step between dimple centers in the direction being perpendicular to the movement of the liquid (gas) flow;
\( Z = z / R \) – relative spatial coordinate;
\( Bi = R \cdot (\alpha_1 + \alpha_2) / \lambda \) – the Bio number for a dimple;
\( \alpha_1, \alpha_2 \) – average heat transfer coefficients for left and right halves of the dimple with fluid flow from the left to the right, W/(m²·K);
\( \lambda \) – coefficient of thermal conductivity of plate material, W/(m·K);
\( \Delta = 1 - h / R \) – relative coordinate of the top of the fin formed by the dimples.

Fig. 4 shows designations of the used values and also the repeating element of the surface.

The solution of equation (2) in Maple program is:

\[
\theta(Z) = _c1LegendreP\left(\frac{1}{2} \cdot \sqrt{1 + 4 \cdot Bi - \frac{1}{2} \cdot \frac{\sqrt{\pi} \cdot Z}{\sqrt{\pi - S_1 S_2}}}\right) + _c2LegendreQ\left(\frac{1}{2} \cdot \sqrt{1 + 4 \cdot Bi - \frac{1}{2} \cdot \frac{\sqrt{\pi} \cdot Z}{\sqrt{\pi - S_1 S_2}}}\right)
\]

Fig. 5 shows temperature curves obtained by solving equation (3) with the following boundary conditions:

\[
\left. \frac{d}{dz} \theta(Z) \right|_{Z=1-\Delta} = 0 ,
\]
\[
\theta(Z)|_{Z=1} = 1 ,
\]
Figure 5. Diagram of the temperature distribution along the height of the considered fin with thermo-insulated top.

The calculation of efficiency of the fin $\eta$ was formed by the known formula (6), where numerator is the heat flow from the real fin, and denominator is the heat flow from an ideal fin.

$$\eta = \frac{Q_{\text{real}}}{Q_{\text{ideal}}} = \frac{\lambda \frac{d^2}{dx^2}(x)|_{x=R} \gamma^2}{\alpha(t_0-t_\infty)2\pi Rh}$$  \hspace{1cm} (6)

where: $s = s_1 = s_2$; heat transfer coefficient $\alpha$ was calculated by the empirical formula (7) obtained by Amirkhanov R. D. [8] for the case of $7 \cdot 10^3 \leq Re_{d_{3}} \leq 5 \cdot 10^3$, $0.1 \leq h_{l}/d_{3} \leq 0.5$, $0.1 \leq h_{k}/d_{3} \leq 0.4$ :

$$\alpha = Nu_{d_{3}} \cdot (\lambda_{\infty}/d_{3}) = 0.033 \cdot Re_{d_{3}}^{0.8} \cdot (h_{l}/d_{3})^{0.42} \cdot (h_{k}/d_{3})^{-0.46(h_{l}/d_{3})^{-0.3}} \cdot (\lambda_{\infty}/d_{3})$$  \hspace{1cm} (7)

Table 1 presents the thermo-technical characteristics of the plates differing from each by depths of the dimples at the same density of their location.

| $h_{l}/R_{3}$ | $d_{3}/R_{3}$ | $s/R_{3}$ | $f_{3}$ | $h_{l}/R_{3}$ | Nu | $\eta$ |
|--------------|--------------|----------|--------|--------------|----|-------|
| **0.63**     | 1.85         | 1.98     | 0.69   | 0.696        | 51.798 | 0.497 |
| **0.72**     | 1.92         | 2.05     | 0.69   | 0.720        | 55.065 | 0.496 |
| **0.78**     | 1.95         | 2.08     | 0.69   | 0.732        | 57.220 | 0.495 |
Table 1 shows that in the studied depth range of dimples (heights of fins), at the density of their arrangement \( f \), equal to 0.69, the efficiency of the surface \( \eta \) remains practically unchanged and equal to 0.5.

4. Conclusions

- Representation of the surface with dimples as the surface with thermally thin fins created by the ball segments grooves, and also the sequence of the performed calculations allowed to quantify not only efficiency of the fin but also efficiency of the surface as a whole.
- Developed by the authors algorithm is the basis of the technology for determining the thermal efficiency of the surface with the studied type of heat exchange intensifier.

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