Promising dimple technologies of vortex heat and mass transfer enhancement in energy and microelectronics

S A Isaev¹, ², I A Popov³, N I Mikheev³, S V Guvernyuk⁴, D V Nikushchenko⁵ and A G Sudakov¹

¹ Saint-Petersburg State University of Civil Aviation, Saint-Petersburg, Russia
² A. N. Tupolev Kazan National Research Technical University (Kazan Aviation Institute), Kazan, Russia
³ Kazan Scientific Center of the Russian Academy of Science, Kazan, Russia
⁴ Moscow State University, Moscow, Russia
⁵ Saint-Petersburg State Marine Technical University, Saint-Petersburg, Russia

E-mail: isaev3612@yandex.ru

Abstract. Turbulent separated air flow and heat transfer at the starting hydrodynamic length of the longitudinal section of the dimpled narrow channel (1 in height, 4 in width, and 84 in length) with inclined one-row oval-trench dimples (OTDs) are calculated. Packages of 20 and 31 dimples are compared. Each dimple has a length of 4.5, a width of 1, a depth of 0.2, an edge rounding radius of 0.2, and an inclination angle of 45°. Re=6000 and is determined in terms of the flow velocity at the channel entrance. The effect of abnormal separated flow intensification and heat transfer enhancement in OTDs due to the dimple density is confirmed. \( \frac{f_x}{f_{xpl}} \)_{min}=−4 and \( \frac{Nu}{Nu_{pl}} \)_{max}=7.2. Relative heat transfer grows by a factor of 1.9 and relative hydraulic losses – by a factor of 1.6.

1. Introduction

Heat transfer enhancement in the turbulent flow between the fins of the air-cooled condenser [1] is an important scientific and practical task. This task is solved using surface vortex generators [2]. The most efficient generators are the oval-trench dimples (OTDs) inclined to the flow [3-17]. The two halves of a spherical dimple with rounded edges are connected by a cylindrical insert – trench. Such a dimple increases heat transfer from a wall more quickly than hydraulic losses grow, i.e. the thermal and hydraulic performance of dimpled channels increases. The analysis of single inclined OTDs at the narrow channel wall using the computer analog of V.I. Terekhov’s test stand [3-7] shows that the secondary flow velocities appearing in such dimples are compared with the bulk turbulent flow velocity. The increase in the length of the dimple with a constant spot area appears to be an important factor affecting the separated flow structure change and intensification in the inclined OTD. The dimple depth based on trench width, the inclination angle of the dimple, and the Reynolds number also have a significant influence on flow and heat transfer intensification in the channel. Most of the studies are made at an OTD inclination angle of 45° close to an optimal one.

Works [8-12] consider the inclined one-row oval dimples of moderate lengthening placed at the narrow channel walls for water, air and oil flows. The density of dimples plays a significant role in heat transfer enhancement. The starting hydrodynamic phase of the dimpled channel goes into the stabilization flow phase. For the heat transfer stabilization length, the applicability of the model for the...
periodic section of the channel with one inclined oval dimple is justified when the periodic boundary conditions are set at the outflow boundaries.

Works [13-15] analyze laminar heat transfer enhancement in the air flow at the stabilization length of the narrow channel with inclined OTDs. It is revealed that the maximum flow velocity in the dimpled narrow channel increases one and a half times. Heat transfer from the channel section with an inclined OTD increases by 80% at a 25% growth of hydraulic losses in comparison with the flat plate-parallel channel.

Works [16-18] are devoted to the modeling of the phenomenon of abnormal intensification of turbulent separated flow and heat transfer at the stabilization length of the dimpled narrow channel in inclined one-row OTDs. It is shown that the flow core velocity increases up to 1.4 times at a dense packing of dimples. It is found that absolute friction in the backflow zone in the dimple increases by a factor of about 5.5 in comparison with friction at the smooth channel wall. The reason for such a phenomenon is as follows. This phenomenon is associated with forming a large pressure drop between the closely spaced zones of stagnation and low pressure at a place where a spiral vortex is generated on the entrance portion of the semi-spherical segment of the dimple.

![Figure 1. Segment of the dimpled channel at the beginning and the end of the control section, as well as at the removal of the top plane wall (a) and on the multiblock computational grids covering the starting length of the channel (b).](image)

The present study focuses on turbulent (Re=6000) heat transfer enhancement at the starting hydrodynamic length of the narrow channel with one-row OTDs inclined at an angle of 45° when their density is varied.

2. Problem statement, solution method, grids
This section considers the turbulent air flow acceleration (Pr=0.7) and heat transfer enhancement (Re=6000) on the long section of the narrow channel that imitates some space between the fins of the air-cooled condenser. The narrow channel is 84 in length, 4 in width, and 1 in height. The 80-long control section has packages of 20 and 31 one-row oval-trench dimples inclined at an angle of 45°. The dimple width is 1, its length is 4.5, its depth is 0.2, and its edge rounding radius is 0.2
The length of the plate-parallel channel section at the entrance and the exit in front of and behind the dimpled control section of the channel is set to 2. The distances between the centers of the dimples are equal to 4 and 2.53. The density of the dimples on the control section is varied from small (0.26) to moderate (0.51) values in comparison with the recommended density of spherical dimples [2]. Symmetry conditions are assigned at the side wall of the channel, uniform laminar air flow is set at the channel entrance, soft boundary conditions are set at the channel exit, and the no-slip condition is set at the channel walls. The system of the Cartesian $x,y,z$ coordinates centered at the beginning of the control section of the channel are introduced as follows: the $x$-axis is along the channel, the $y$-axis is directed upward, and the $z$-axis is across the channel. All linear sizes are selected in relation to channel height. Cartesian velocity components and the Re number are defined in terms of the bulk flow velocity in the channel. The room temperature $T=293$ K is chosen as the characteristic temperature. The channel walls are isothermal and the temperature drop is insignificant: at the bottom wall with dimples, $T = 1.034$ and at the top wall, $T = 1$.

The system of the steady Reynolds-averaged Navier–Stokes equations for incompressible liquid and the energy equation are solved using the second-order approximation factorized finite-volume method on the multiblock different-scale grid overlapping the computational grid [4-18]. The SST model [19] modified within the framework of the Rodi–Leszhiner–Isaev approach is used to close the momentum equations. The multiblock computational technique (MCT) is based on solving the governing equations for sub-domains covered with structured grids matching the dimpled surface of the narrow channel. The linear interpolation procedure, as in [6,18], is applicable in the zones where dimples overlap. Leonard’s quadratic upwind scheme [20] is adopted to present the convective terms on the implicit hand-side of the linearized equations and the upwind scheme with one-sided differences – the implicit terms on the explicit hand-side of the equations. The diffusion terms are approximated using the central differences. The used method of solving the algebraic equations is the BiCGSTAB preconditioner [21] with an AMG preconditioner from Demidov’s library (amgcl) [22] for pressure correction and ILU0 for other variables. The VP2/3 (velocity-pressure, 2D/3D) code using the MCT is selected as a base code in this numerical study.

A multiblock different-scale grid is designed by overlapping particular different-topology grids (Figure 1, b). A Cartesian channel grid is condensed to the top plane wall and to the bottom curvilinear wall. Longitudinal and transverse steps of this grid are set to 0.07 and the near-wall step is assigned equal to $5 \times 10^{-4}$. This grid covers a curvilinear non-orthogonal grid with the height of 0.175. It matches the dimpled bottom wall of the channel and is more detailed than the base grid. Longitudinal and transverse steps of the grid are of the order of 0.05. The near-wall step is assigned to be $5 \times 10^{-4}$. The total number of computational cells is 11 million.

Figure 2, b depicts the channel section $z=1$ that passes through the fitted midsections connecting the entrance portion of the spherical segments of the dimples and the cylindrical trenches. The idea to highlight the above section is to show the evolution of separated flow parameters in dimples as the flow develops in the dimpled channel.

![Figure 2](image_url)  
**Figure 2.** Configuration of the bottom wall of the channel with 20 (a) and 31 (b) OTDs.

### 3. Discussion of the obtained results

Figures 3–5 illustrate some of the results of the present study.

The distribution of the static pressure in the dimpled channel section (Figure 3, a), which passes through the separated flow zone in the OTD, looks like a pulsating pressure decrease corresponding to a linear pressure decrease in the plate-parallel channel. Local maximum pressures are realized at the
edges of the dimples at the places where they interact with the external flow. Local minimum pressures are observed in the separated flow zones.

The analysis of the distributions $\frac{f_x}{f_{xpl}(x)}$ in the channel section (Figure 3, b) shows that the values of local maximum and local minimum pressures increase cyclically from dimple to dimple along the channel length. The influence of flow past the upstream dimples on the dimples in the wake behind them significantly intensifies the separated flow in the dimples and increases friction in the spaces between the one-row dimples in the package. Minimum relative friction in the separated flow zones approaches a value of -1.5 for the package of 20 inclined loosely packed OTDs. Approximately from the midsection of the channel, it is within the range -2 - (-2.5) for the package of 31 closely packed OTDs. This is evident of the fact that abnormal separated flow intensification in the inclined OTD [15] is also seen at the starting hydrodynamic length. Maximum relative friction reaches a value of 6 in the package of 20 loosely packed OTDs. In the package of 31 closely packed OTDs, maximum friction approaches a value of 8 in the first part of the dimpled channel; then it decreases and reaches the friction value observed in the package of 20 loosely packed OTDs. However at a dense packing of OTDs, the level of relative friction in the spaces between the dimples significantly grows; and local minimum friction reaches a value of 2 in these spaces.

![Figure 3](image)

**Figure 3.** Distributions of $p$ (a), $\frac{f_x}{f_{xpl}}$ (b) and $\frac{Nu}{Nu_{pl}}$ (c) at $z=1$ for the channels with 20 (curves 1) and 31 (curves 2) OTDs. Curve 3 corresponds to the plate-parallel channel.

![Figure 4](image)

**Figure 4.** Distributions of $\frac{Nu_{m}}{Nu_{mpl}}$ integrated over transverse (b) and longitudinal (c) coordinates for the channels with 20 (curves 1) and 31 (curves 2) OTDs.

The distribution of the relative Nusselt numbers (Figure 3, c) is also cyclic and sharply decreases in the separated flow zones and increases at the dimple edges and in the spaces between the dimples. For
the package of one-row OTDs, the maximum and minimum values of heat transfer monotonically increase. For the package of closely packed OTDs, the value of heat transfer increases in the first part of the channel (its maximum value is 6.5) and it somewhat decreases in the second part of the channel (its local maximum value is 6). For the package of 20 loosely packed OTDs, maximum heat transfer is seen in the vicinity of the channel exit and its value is close to 5.5. Minimum relative heat transfer in the separated flow zones of OTDs appears to be below 1.

Figure 5. Distributions of $\frac{Nu}{Nu_{pl}}$ at the wall of the channel with 20 (a) and 31 (b) OTDs.
The inclined OTDs located at the heated wall of the narrow channel significantly enhance the integral heat transfer characteristics of the device. As shown in Fig. 3, increasing the density of the dimples with the same depth of 0.2 significantly enhances heat transfer. For the package of closely packed OTDs, relative heat transfer averaged over the transverse coordinate grows with a high rate in the first part of the channel, reaching a local maximum value of 2.6. Further, with increasing $x$, the relative Nusselt number slowly decreases, having a value above 2.4 (Figure 4, a). The analysis of the distributions of the relative Nusselt numbers averaged over the longitudinal coordinate (Figure 4, b) shows that the left part of the channel has an area of decreased relative heat load (less than 1). The right side of the channel is evident of heat transfer enhancement, reaching a 3-fold increase in relative integral heat load for the package of closely packed OTDs. For the package of loosely packed OTDs, $\frac{Nu_m}{Nu_{mpl}}$ approaches a value of 2.2.

It should be noted that the package of loosely packed dimples is characterized by a monotonic increase of heat transfer that tends to reach its value at the convective heat transfer stabilization length. Fig. 5 illustrates the distribution of heat transfer at the channel wall at the end of the dimpled section. $\frac{Nu}{Nu_{pl}}$ achieves a value of 7.2. In the spaces between the dimples, $\frac{Nu}{Nu_{pl}}$ exceeds a value of 2. A significant heat transfer growth of order 2 is seen in the inclined OTDs.

Conclusions
Numerical simulation of turbulent heat transfer enhancement in the space between the dimpled fins of the air-cooled condenser was performed. The narrow channel with a package of inclined one-row oval-trench dimples (OTDs) at $Re = 6000$ was considered. The channels with loosely packed (20) and closely packed (31) OTDs on the 80-long control section of the 4-wide channel were compared. The case of 31 OTDs with a dimple depth of 0.2 is preferred. The growth of hydraulic losses does not exceed 60% at an almost 2-fold increase in heat transfer in comparison to the plate-parallel channel. The turbulent flow at the dimpled channel exit accelerates by a factor of 1.23 in comparison to the plate-parallel channel.

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