Thermal-hydrodynamic design of energy-efficient surfaces with inclined oval-trench vortex generators

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Abstract. In the present study it is found that enhancement of air turbulent flow and heat transfer at Re=10⁴ is influenced by a set of one-row oval-trench dimples of depth 0.25 inclined at an angle of 65° to the incoming flow and located at the heated wall of the rectangular (9×1) narrow channel when the dimple step H is varied. The periodic section of the dimpled channel 8 in length is considered. The dimple step H is varied from 8 to 2. Abnormal enhancement of flow and heat transfer at H=2 is followed by both a 4-fold decrease in relative negative friction and a 6.5-fold increase in relative heat transfer in the separated flow zone. The phenomenon of turbulent flow acceleration with a 1.39-fold increase in the maximum flow velocity (at H=2) in the dimpled narrow channel in comparison to the plane-parallel channel is discovered.

1. Introduction
Surface vortex generators have proven to be effective tools for heat transfer enhancement [1]. Of particular interest, among them are sets of dimples, which are characterized by the fact that heat transfer grows faster than hydraulic losses. However, it appears that inclined oval-trench dimples are preferred to spherical ones and close to symmetrical dimples as effective vortex generators. Comprehensive studies [2-4] deal with single, periodic, one-row dimples of constant spot area in a narrow channel and analyze the influence of Reynolds number and dimple depth. Works [5,6] are concerned with abnormal enhancement of separated flow and heat transfer in inclined oval-trench dimples, as well as with medium acceleration in the core of laminar and turbulent flows in the channel. The influence of the angle of inclination of the oval-trench dimple on the limiting characteristics of flow and heat transfer in a dimpled narrow channel is investigated.

2. Problem statement
Low-velocity turbulent air flow and heat transfer in the dimpled rectangular narrow channel at the Reynolds number, Re=10⁴, is calculated by solving the steady Reynolds-averaged Navier–Stokes equations (RANS) for incompressible viscous liquid and the energy equation. To close these equations, Menter’s shear stress transport model (SST model) is used; the eddy viscosity is determined using the strain rate tensor modulus [8]. In [9], it is shown that when calculating separated flows such an SST model, unlike the original one [10], finds false maximum eddy viscosity values in the cores of large vortices. To eliminate errors, the SST model was modified with the consideration of the streamline curvature within the framework of the Rodi–Leschziner–Isaev (RLI) approach. The RLI approach corrects eddy viscosity by multiplying it by the correction function fc = 1/(1+Cc×Ri.t). Here, Cc=0.02 is the semi-empirical constant determined from the conditions of the best agreement between numerical predictions and experimental data [11], and Ri.t is the turbulent Richardson number.
Figure 1. Periodic sections of the channel with inclined one-row oval-trench dimples ($\theta=65^\circ$) at the lower wall (the upper wall removed) for different dimple steps. $a - H=8$; $b - 4$; $c - 2.667$; $d - 2$.

The flow stabilization length is considered in the form of the periodic section of the channel with inclined one-row OTDs arranged at the heated wall. Periodic boundary conditions (Figure 1) are assigned at the through-flow input (A) and output (B) boundaries of the periodic section. No-slip conditions are satisfied at the walls. The dimpled lower wall and the upper plane wall are isothermal. The side walls are thermally insulated. The lower wall is kept at a temperature of 303 K and the upper wall – at a temperature of 293 K taken as characteristic temperature. The channel height is selected as linear scale. The channel width is equal to 9 and the periodic section length is equal to 8. OTDs of depth 0.25, inclined at an angle of 65°, are 1.05 in width and 7.05 in length. The rounding radius of edges is 0.21. The sizes and the inclination of dimples are selected from the studies of heat transfer in turbulent flow past single and one-row dimples [6,7]. A step $H$ in the one-row dimple is varied from 2 to 8 (Figure 1). The origin of the Cartesian $x,y,z$ coordinates is in the center of the lower wall surface of the periodic section. The lower wall has an $8 \times 8$ control area where integral characteristics, including the total Nusselt number $Nu_{mm}$, are determined.

3. Computational Methodology
The RANS equations and the energy equation are solved by the multiblock computational technique (MCT) with the use of a specialized VP2/3 code (Velocity-Pressure 2D/3D) [12]. The MCT is created on the basis of different simple-topology grids O and H, including overlapping ones. Local grids reflect both the geometric features of the task stated and the specific different-scale structural elements such as boundary and shear layer, and separated flow zones. The factorized finite-volume method was developed using the SIMPLER-algorithm [13] applied to the original equations written in increments.
Rhie–Chow’s monotone approximation for determination of the pressure correction [14] is modernized and is used in computations on centered grids [15].

The convective terms on the explicit hand-side of the transport equations are presented, using both Leonard’s one-dimensional quadratic upwind scheme [16] and the TVD scheme [17]. The method for solution of discretized algebraic equations is the BiCGSHAB preconditioner [18] with an AMG preconditioner from Demidov’s library (amgl) [19] for pressure correction and ILU0 for other variables. The linear interpolation is used for grid by grid computation in near-wall cells [12]. The VP2/3 code similar to the OpenFOAM code is used to solve the stated tasks with the implication of multicore computational codes.

The procedure of pressure and bulk temperature [12] is applied to the flow in the periodic section of the channel.

![Figure 2](image)

**Figure 2.** Distributions of relative friction components $f_s (a,b)$ in the longitudinal section of the oval-trench dimple inclined at $\theta=65^\circ$ on normal (a) and enlarged (b) scales for different $H$.

1 – $H=8$; 2 – 4; 3 – 3.3; 4 – 2.667; 5 – 2.

**Conclusion**

1. A good quantitative agreement of the predictions of extreme averaged characteristics of turbulent flow, heat transfer, and hydraulic losses, as well as of the friction and Nusselt number distributions in the midsection of the inclined oval-trench dimple in the 8-long periodic section of the rectangular ($9\times1$) plane-parallel channel is obtained on computational grids of both different topology (multiblock different-scale overlapping structured and monoblock matched to the curvilinear channel wall) and different packing of dimples. The OTD inclined at an angle of 45° to the incoming flow is located in the center of the heated wall. The dimple length is 7.05, its width is 1.05, its depth is 0.25, and its rounding radius of edges is 0.21. The predictions are obtained using the Menter 2003 SST model with the streamline curvature modified within the Rodi–Leschziner–Isaev approach in order to close the Reynolds-averaged Navier–Stokes equations.

2. As in the case of laminar flow acceleration in the narrow channel with inclined OTDs, at a dense packing of OTDs inclined at an angle of 65° turbulent flow acceleration is obtained when the maximum flow core velocity is increased by a factor of 1.39 at $H=2$ in comparison to the plane-parallel channel.

3. A dense packing of one-row dimples substantially increases abnormal enhancement of separated turbulent flow and heat transfer on the entrance portion of the OTD inclined at an angle of 65° at the heated wall of the dimpled channel. At $H=2$, this enhancement is characterized by a 4-fold increase in a maximum absolute value of $f_s/f_{pl}$ (figure 2) of the OTD and by an almost 6.5-fold increase (figure 3) of relative heat transfer (in relation to the parameters in the plane-parallel channel). A maximum absolute value of the secondary (transverse) flow velocity exceeds by approx. 10% a maximum value of the flow velocity in the plane-parallel channel. A maximum absolute value of the recirculation flow velocity in
the narrow channel with inclined OTDs nearly three times exceeds the same recirculation flow velocity in the spherical dimple, reaching a value of 0.89 of bulk velocity in the plane-parallel channel.

4. Transverse strip-integrated relative heat transfer in the separated flow zone on the entrance portion of the OTD inclined at an angle of 65° to the incoming flow at \( H=2-2.667 \) grows about 5 times in comparison to the plane-parallel channel. In this case, the thermal performance of the control surface of the periodic section of the narrow channel does not exceed 2, whereas the thermal-hydraulic performance, estimated with account of the hydraulic loss coefficient raised to the power of -1/3, is equal to a value of 1.6 (figure 4).

**Figure 3.** Distributions of the relative Nusselt numbers \((\text{Nu}/\text{Nu}_{\text{pl}})\) in longitudinal \((a,b)\) midsection of the oval-trench dimple inclined at the angle \( \theta=65°\) on normal \((a)\) and enlarged \((b)\) scales for different \( H \). 1 – \( H=8 \); 2 – 4; 3 – 3.3; 4 – 2.667; 5 – 2.

**Figure 4.** Total relative heat transfer \( \text{Nu}_{\text{mm}}/\text{Nu}_{\text{mmpl}} \) over the surface of the periodic section of the dimpled channel (1) and over the surface bounded by the dimple (2), relative hydraulic losses \( \zeta/\zeta_{\text{pl}} \) (3) and thermal-hydraulic performance (THP) determined as \( (\text{Nu}_{\text{mm}}/\text{Nu}_{\text{mmpl}})/( \zeta/\zeta_{\text{pl}}) \) (4) and as \( (\text{Nu}_{\text{mm}}/\text{Nu}_{\text{mmpl}})/( \zeta/\zeta_{\text{pl}})^{-1/3} \) (5) versus dimple step \( H \).

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