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# Vortex intensification of laminar heat transfer in a stabilized section of a channel with two-row inclined grooves when pumping coolants with different Prandtl numbers

© S.A. Isaev[1,2], E.A. Nikushchenko[1], D.V. Nikushchenko[1], N.S. Dushin[3], A.G. Sudakov[2]

[1] State Marine Technical University, St. Petersburg, Russia
[2] Novikov St. Petersburg State University of Civil Aviation, St. Petersburg, Russia
[3] Kazan Scientific Center RAS, Kazan, Russia
E-mail: isaev3612@yandex.ru

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With increasing Prandtl numbers, thinning of the near-wall temperature layers near the channel wall with double-row inclined grooves in a stabilized section of the coolant movement makes it possible to significantly intensify heat transfer for oil media. An increase in relative heat transfer for M20 oil by 44 times and for TM by 17 times was discovered, compared to almost 4 times and one and a half times the increase for water and air.

**Keywords:** Separated flow, heat transfer, narrow channel, inclined groove, intensification, numerical modeling, inert gas, air, water, transformer oil, M20 oil.

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Dimpled surface technology, which is aimed at vortex enhancement of heat transfer in channels with moderate growth of hydraulic losses, has a wide range of applications in thermal power engineering [1,2]. There is also some interest in it in the field of microelectronics [3–5], and even fewer studies have been published on the topic of influence of the Prandtl number on enhancement of laminar heat transfer. Specifically, studies [6,7], where the flow of transformer oil in a minichannel with single-row spherical and oval dimples on a heated wall was discussed, were mentioned in review [2]. It was found that the positioning of spherical dimples of moderate depth (0.2) with a pitch of 1.5 on a wall of a nine-section minichannel heated to 30°C (with a width of 2 and a height of 0.5 in units of channel height) with a low-velocity (Reynolds number $\mathrm{Re} = 308$) flow of transformer oil promotes vortex enhancement of heat transfer and augments the heat transfer from this wall by a factor of approximately 2.5 compared to the case of a smooth channel with a 7% reduction in hydraulic losses. Dimples of an oval shape with the same spot area (a width of 0.55 and a length of 1.5 in units of diameter of the base dimple spot) and the same depth provide a further 3.4-fold enhancement of heat transfer (i.e., a 8.5-fold enhancement in total) with a 2.1% reduction in hydraulic losses compared to the case of a smooth channel. Local acceleration of laminar air flow in a stabilized section of a channel with sparse single-row inclined grooves was discovered in [8]. The maximum longitudinal velocity in the forming shear flow is almost 1.5 times greater than the maximum flow velocity in a plane-parallel channel. It was established somewhat later that the thermal efficiency, which is specified by the relative total Nusselt number averaged over the washed surface of a section with an inclined groove, has a maximum of approximately 1.8 at a depth of 0.3125 (in units of channel height) and the maximum thermohydraulic efficiency of 1.3 is achieved at a depth of 0.25. Local acceleration of turbulent air flow was observed in [9] for a dense arrangement of inclined grooves. The genesis of anomalous enhancement of separated flow and heat transfer in inclined grooves on channel walls and plates was analyzed in review [10]. Specifically, the measured velocity fields of laminar and turbulent air flow in a channel with 26 double-row inclined grooves were compared with numerical predictions obtained using a digital counterpart of the experimental stand at the Kazan Research Center of the Russian Academy of Sciences. In the present study, the thermal efficiency and hydraulic losses of the specified channels with double-row inclined grooves and different coolants (M20 oil, transformer oil, water, air, inert gas) with different Prandtl numbers pumped through them at low Reynolds numbers $\mathrm{Re} = 308$ (as in [6,7]) are compared.

The digital counterpart of the experimental stand at the Kazan Research Center with a narrow plane-parallel channel is examined. Two rows of closely spaced grooves with a depth of 0.25 inclined by ±45° to the laminar flow are positioned on the lower wall of this channel [10]. The channel height and the mass-average flow velocity in the channel are chosen as the characteristic scales for normalization. The dimensionless width and length of the periodic section of the channel are 10 and 5.06. The origin of Cartesian coordinates (longitudinal $x$, vertical $y$, and transverse $z$) is located in the middle of the inlet cross section of the grooved section. The corresponding Cartesian components of laminar flow velocity are denoted as $U$, $V$, and $W$.

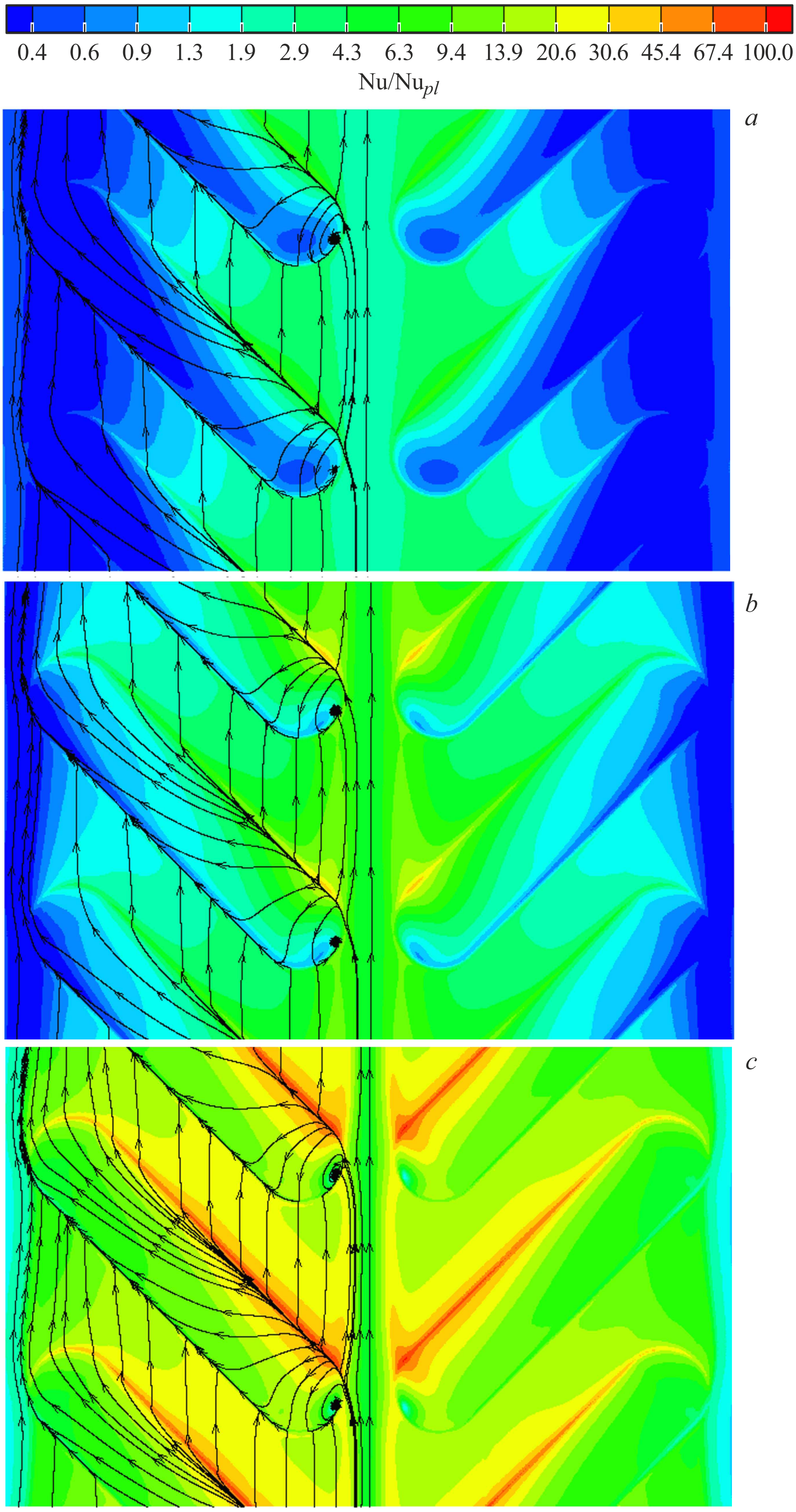


**Figure 1.** Comparison of relative Nusselt numbers Nu/Nu$_{pl}$ and spreading patterns on the heated surface of a channel with a dense arrangement of double-row grooves in a stabilized section for air ($a$), water ($b$), and oil ($c$) coolants.

Body-fixed coordinates $s$, $y$, and $t$ oriented in the median longitudinal section, vertical direction, and transverse median section are introduced within each groove. Double-row grooves with a width of 1, which consist of two halves of a spherical dimple with a depth of 0.25 and a connecting cylindrical trench with a length of 3.5, are oriented in such a way (inclined by $\pm 45^\circ$) that their inlet spherical segments are positioned in the middle part of the channel. The distance between the centers of grooves in a dense arrangement is set to 2.53 (in accordance with the data from [9]), and the edge radius of grooves is taken to be equal to 0.025.

A system of steady-state Navier–Stokes equations is used to characterize the steady-state laminar flow of a viscous fluid at $\mathrm{Re} = 308$ in the channel with a structured wall. Convective heat transfer is characterized by the energy equation. Periodic boundary conditions are set for all dependent characteristics at the flow boundaries of a channel section. The no-slip condition is satisfied at the channel walls. The upper flat wall of the channel is kept at room temperature (20°C, 293 K), which is chosen to be the characteristic one (dimensionless $T = 1$). The temperature of the lower structured wall is taken to be isothermal and equal to 30°C (303 K); i.e., the dimensionless temperature is $T = 1.034$. The side walls of the channel are thermally insulated.

In media with constant physical properties (with their viscosity and thermal conductivity being independent of temperature), the thermal problem is solved separately from the dynamic problem with the use of velocity fields calculated earlier. The physical properties of a heterogeneous medium such as oil (transformer oil or grade M20 oil) are specified in tabular form and are determined in the flow field via interpolation (as was done in [11]). The energy equation is solved simultaneously with the equations of motion of the medium in this case, and the influence of the temperature field on the flow parameters is taken into account. The Prandtl number is assumed to be equal to 7, 0.7, and 0.2 for water, air, and inert gas (helium–xenon mixture), respectively, and varies widely from several hundred to tens of thousands when the oil coolant is pumped.

A multi-block grid is constructed on the basis of multi-scale fragments of structured grids [12]. The spacing at the wall is $2.5 \cdot 10^{-4}$. The grid contains 5.7 million computational cells.

Numerical modeling of separated flow of viscous media and convective heat transfer is performed using multi-block computational technologies [12]. The calculation methodology is verified by solving the problem of convective heat transfer in a bundle of round pipes with oil coolant pumped through it [11]. Numerical predictions of integral thermohydraulic characteristics agree closely with the experimental data reported by Žhukaukas [13].

The problem is solved iteratively. At each iteration step, the pressure correction equation is solved and the fields of Cartesian components of velocity, pressure, and temperature are calculated. The computation ends when the maximum errors of dependent variables reach the level of $10^{-5}$ and the extreme local and integral parameters, including hydraulic loss coefficient $\zeta$ and total heat transfer $\mathrm{Nu}_{mm}$ in the section with inclined grooves, are stabilized. Local characteristics of flow and heat transfer in a stabilized section of the channel with double-row inclined grooves are reported in the present study. These include the fields of relative heat transfer $\mathrm{Nu}/\mathrm{Nu}_{pl}$ and relative Nusselt numbers $\mathrm{Nu}_m/\mathrm{Nu}_{m\,pl}$ integrated over transverse strips. Subscript *pl* denotes the parameters of a plane-parallel channel (including those on a heated wall). Hydraulic loss coefficient $\zeta$ in a channel (the scenario with double-row inclined grooves on a heated wall included) is determined using the method for calculating hydraulic losses in a channel with a spherical dimple [12].

Some of the most significant results obtained in calculations are presented in the table and in Figs 1–3. Relative heat transfer $\mathrm{Nu}_{mm}/\mathrm{Nu}_{mm\,pl}$ from the control channel section with a length of 5.06 and a width of 8 and heat transfer $\mathrm{Nu}_{mmd}/\mathrm{Nu}_{mmd\,pl}$ from the rectangular section of the curved surface within which the outline of an inclined groove is located are the focus of our study. It was found that the coolant type has a fundamental effect on the indicated characteristics; notably, the relative heat transfer of oil media in the mentioned sections (approximately 44–45 for M20 oil and 17–18 for transformer oil) exceeds the one for media with constant Pr by almost an order of magnitude: the values for water, air, and a mixture of inert gases are 3.56, 3.12; 1.35, 0.98; and 0.96, 0.67, respectively. It should also be emphasized that the relative heat transfer for oils from the surface of the section within the groove outline is slightly higher than the one in the control section of the channel with inclined grooves. In the case of water, air, and inert gases, the relative heat transfer from the surface within a groove is significantly lower than the heat transfer

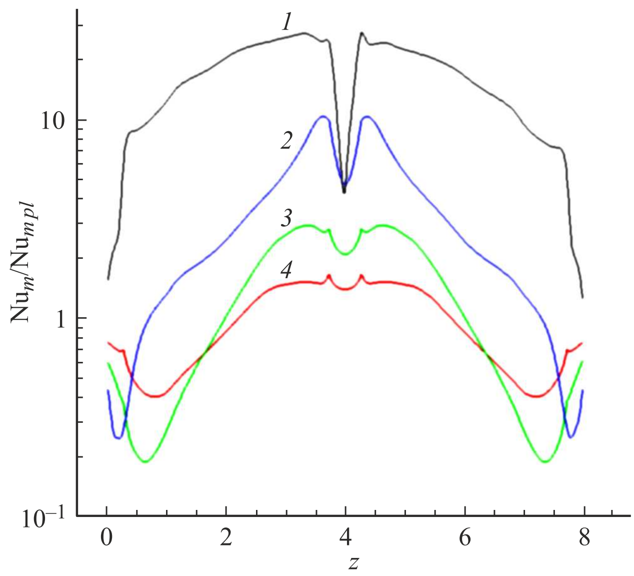


**Figure 2.** Comparison of the distributions of relative Nusselt numbers $\mathrm{Nu}_m/\mathrm{Nu}_{m\,pl}(z)$ for transformer oil (*1*), water (*2*), air (*3*), and inert gas (*4*) integrated over longitudinal strips of the channel section with inclined grooves.

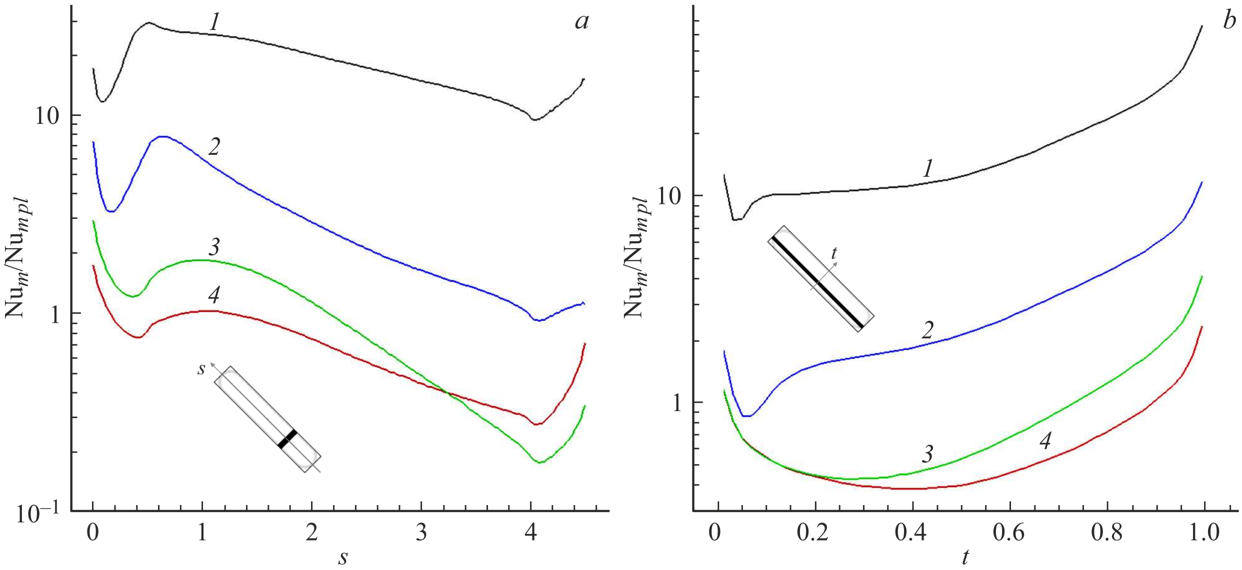


**Figure 3.** Comparison of relative Nusselt numbers $\mathrm{Nu}_m/\mathrm{Nu}_{m,pl}$ for transformer oil (*1*), water (*2*), air (*3*), and inert gas (*4*) integrated within the rectangular region bounded by the outline of the left groove spot over transverse strips in longitudinal $s$ ($a$) and transverse $t$ ($b$) directions.

Influence of the coolant medium on the relative heat transfer and hydraulic losses in a stabilized section of the channel with double-row inclined grooves and a rectangular section bounded by the outline of an inclined groove

| Medium | $\mathrm{Nu}_{mm}/\mathrm{Nu}_{mm\,pl}$ | $\xi/\xi_{pl}$ | $\mathrm{Nu}_{mmd}/\mathrm{Nu}_{mmd\,pl}$ | $\xi_d/\xi_{d\,pl}$ |
|---|---|---|---|---|
| M20 oil | 44.4 | 1.35 | 45.1 | 1.41 |
| Transformer oil | 17.2 | 1.26 | 18.1 | 1.31 |
| Water<br>Air<br>Inert gas | 3.56<br>1.35<br>0.96 | 1.12 | 3.12<br>0.98<br>0.67 | 1.17 |

from the control section. The relative hydraulic losses in the control section of the channel are approximately 5% lower than those in the channel section with an inclined groove. The $\xi/\xi_{pl}$ values are close to 1.3–1.4 for oils and fall within the 1.1–1.2 range for the examined media with Pr = const. As was noted in [3], the mechanism of heat transfer enhancement in inhomogeneous media with high Prandtl numbers is associated with profound thinning of near-wall temperature layers.

The $\mathrm{Nu}/\mathrm{Nu}_{pl}$ fields for transformer oil shown in Fig. 1 differ significantly from similar $\mathrm{Nu}/\mathrm{Nu}_{pl}$ fields for air and water, although the structures of separated flow are similar. It should be noted that the indicated spreading patterns were obtained by computer visualization of trajectories of liquid particles in the surface layer located on the curved heated channel wall at a distance of half the near-wall spacing ($10^{-4}$). The particle trajectories in the layer are calculated based on the fields of Cartesian velocity components $U$ and $W$.

The maximum heat transfer is observed on the structured wall in the vicinity of the channel symmetry plane (Fig. 2) near the inlet sections of inclined grooves. The $\mathrm{Nu}_m/\mathrm{Nu}_{m\,pl}(z)$ distributions averaged over longitudinal strips of the control section demonstrate that the maximum values for transformer oil and the mixture of inert gases are 12 and on the order of 1, respectively. The relative heat transfer decreases significantly toward the side walls, and regions with inhibited heat transfer emerge for Pr = const.

Inside the grooves, the heat transfer intensity decreases from the inlet to the end part (Fig. 3, $a$); however, $\mathrm{Nu}_m/\mathrm{Nu}_{m\,pl}(s)$ exceeds 10 for transformer oil and turns out to be less than 1 in the end part for air and inert gases. The upstream sides of grooves are heat-stressed (Fig. 3, $b$) with the maximum $\mathrm{Nu}_m/\mathrm{Nu}_{m\,pl}(t) = 16$ corresponding to transformer oil.

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### Conflict of interest

The authors declare that they have no conflict of interest.

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