Numerical and Physical Simulation of Heat Transfer Enhancement Using Vortex Generators

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Abstract. Vortex generation and flow disruption in fluid ducts by means of surface modification is a widely used passive heat transfer augmentation technique. The present paper contains the results of numerical and experimental studies of the flow fields and heat transfer enhancement in the ducts with oval-trench and oval-arc shaped dimples applied to the heat transfer surface. The results of the numerical study of the flow over the dimpled surface with periodic boundaries are verified by experimental visualization of the flow and temperature fields. The influence of characteristic geometrical parameters of the dimples on the heat transfer, friction factor, and flow structure in heat-exchange ducts has been obtained. Preliminary results on the artificial neural networks application on heat transfer and friction factor assessment for the flow over dimpled surfaces are presented.

1. Introduction

The major research of heat transfer augmentation by dimple systems today is the determination of the rational form of the dimple which provides the best thermal performance of the whole dimpled surface. The desired dimple shape is the one for which the heat transfer augmentation is ahead or comparable to the rise of the hydraulic losses. Resolving this challenging issue by the researchers will be decisive for heat-exchange equipment production for manufacturers and customers. A fairly full overview of the experimental investigations on fluid flow and heat transfer in the ducts with dimpled surfaces for various technical applications is presented in [1].

Numerical simulations have sharply increased the research effectiveness, as it has reduced the costs associated with the establishment of thermophysical field experiments. It allows estimating in detail the flow characteristics at various conditions (flow regimes, boundary conditions, thermal and physical properties of coolants, etc.) and different geometrical parameters (size, shape, and arrangement of dimples, etc.). However, numerical simulations findings have to be proved and verified with the results obtained experimentally. The paper discusses the outcomes of numerical simulations review of the fluid flow over the surfaces with implemented dimples of various shapes aimed to determine the optimal dimple geometrical parameters. The considered results were compared with the experimental results of friction factor and heat transfer coefficient determination in the ducts with the dimpled surfaces versus the dimple arrangements.
2. Review of numerical simulation
The recent study has shown a high thermal performance for the oval-trench shaped dimples inclined by an angle of attack \( \phi \) to the main flow. The oval-trench dimple consists of two halves of hemispherical dimple with the depth \( h \) spaced on the distance \( l \) and connected by a trench (Figure 1).

![Figure 1. A characteristic scheme of the oval-trench shaped dimple:](image)

 Isaev S. A. et. al in [3, 4] have presented numerical results of turbulent water flow over the oval-trench dimple inclined to the main flow by the angle of \( \varphi=45^\circ \). The value of the Reynolds number in the simulations remained constant \( Re=10^4 \) and was based not on the value of mean flow velocity and the diameter of the baseline spherical dimple. The study of the turbulent water flow over the oval-trench dimple oriented at the attack angle \( \varphi=30-60^\circ \) to the main flow [5] has shown that the best thermal properties are found for the value of the relative elongation of the dimple \( l_d/b =6 \) and for the angle of attack of \( \varphi=45^\circ \). The laminar flow (\( Re=10^3 \)) in the narrow duct with one row of oval-trench dimples oriented at the attack angle of \( \varphi=45^\circ \) to the main flow was studied in [6, 7] and varying the relative depth of the dimple \( h/b \) in the range \( h/b=0.0625-0.375 \). The numerical results presented by Isaev [9] were obtained for the periodic duct section with the row of oval-trench dimples inclined to the main flow. The other parameters of the simulation were as follows: the length of the periodic domain of \( 8h_{ch} \) and the width of the duct of \( 9h_{ch} \), where the \( h_{ch} \) is the height of the domain. The oval trench dimples had a length of \( 7.05b \), width \( b=h_{ch} \), depth of \( 0.25 h_{ch} \). The study on the attack angle influence (\( \varphi \) was varied in the range \( 1-89^\circ \)) has shown that optimal value of the attack angle of \( \varphi=45^\circ \). The extended results of [9] were presented in the work [10]. The value of the relative notching step \( S/h \) was varied in the range \( S/h_{ch}=2-8 \). The value \( S/h_{ch}=8 \) decreased to the value of \( S/h_{ch}=2 \), the attack angle value at which the extreme local velocities were observed was found to be \( \varphi=60^\circ \).

Based on the results noted above the recommendations on the values of the rational geometrical parameters of the oval-trench dimples have been formulated and patented. The elongation of such dimple varies in the range of \( l/b=5.57-6.78 \) and relative depth of \( h/b=0.18-0.37 \) (\( h/h_{ch}=0.18-0.37 \), \( h_{ch} \) – height of the duct), while the value of the attack angle to the mean flow is fixed \( \varphi=45-60^\circ \) [8].

A more thorough investigation of the influence of the relative periodic section length on thermohydraulic properties of the airflow in the plain-parallel duct was presented in [11]. It was shown, that the magnitude of a heat transfer performance could be increased from \( Nu/Nu_0 =1.18 \) to \( 2.27 \), by reducing the notching step \( S=8 h_{ch} \) to \( 2 \cdot h_{ch} \). The corresponding friction factor growth was from \( \xi/\xi_0 =1.18 \) to \( 2.23 \). Thus, the value of thermal-hydraulic efficiency \( E=1.1 \) could be achieved. However, these findings of the numerical modeling should be verified and compared to the results obtained experimentally.
Experimental studies on friction factor and heat transfer in ducts with the dimpled surfaces where the dimples had oval or oval-trench shape were carried out in a limited number of works. However, in contrast to the numerical studies cited above the dimples on the heat transfer surface were manufactured differently. The rounding of the dimple edge, as well as the dimple contours, considerably differed from the model used in the numerical studies. The analysis of streamlines and values of local friction factors for the flow over the oval-trench dimple (Figure 6 a, b) presented in [3–11] revealed the velocity stagnation areas in the vicinity of the front dimple edge. This has resulted in the local decrease of the heat transfer coefficient magnitudes. The elongation of the hemispherical dimple to the oval and oval-trench shape inevitably accompanies the noted problem.

The oval-arched dimple shape (Figure 7) was proposed to settle the described problem [12]. The tangent line to the centerline for such dimple has to be 45 and 0 degrees at the begging and at the end of the dimple, respectively. Thus, the attack angle with respect to the main flow varies within the indicated range along the dimple. The other geometrical parameters of the dimple have to be settled as follows: $l_d/b=5.57-6.78$; $h/b=0.18-0.37$; $r=0.025\cdot b$.

3. Experimental methodology
The experimental investigation was carried out on the experimental air stand presented in Figure 4. The volume flow rate was adjusted by the choice of the compressor 1-3 with the required capacity,
incorporated by the electromagnetic two-position valves 4, 5, 6, operated by means of the control panel 21. The air was filtrated by the filtration module 7 and dehumanized by dehumidifier 8. The volume flow rate was measured by the ultrasonic flow meter 16. The working area was heated up by an electrical resistance metal film substrate.

Figure 4. Scheme of the experimental stand: 1, 2, 3 – compressors (capacity 2400 l/min, 1400 l/min, 880 l/min); 4, 5, 6, 13 – solenoid valve with two-position pilot action; 7 – filtration module for cleaning compressed air; 8 – dehumidifier; 9, 11 – valves; 10, 12 – receivers; 14 – electrically operated ball valve; 15 – electrically operated ball valve for bypass; 16 – ultrasonic flowmeter-gas meter; 17 – heater; 18 – energy source for heater; 19 – working area; 20 – controller; 21 – valve control panel 4, 5, 6, 13

The working area had the shape of a narrow rectangular duct with the replaceable dimpled surface. The flow stabilization ducts with the length of 50 equivalent diameters were added before and after the working area. The static pressure taps were installed in the lid of the working area in order to evaluate the friction factor value (equation 1). The local heat transfer coefficients (equation 2) were calculated by evaluating the values of the chromel-type thermocouples installed under the heat transfer surfaces.

\[
\xi = \frac{2 \Delta P \cdot d_e}{\rho \cdot \overline{V} \cdot L}
\]

\[
\alpha = \frac{Q}{\Delta T \cdot A}
\]

where \( \rho \) is the air density, kg/m\(^3\), \( \overline{V} = \dot{m}/(\rho \cdot A_{ch}) \) is the mean airflow velocity, m/s; \( A_{ch}, P \) are the area and perimeter of the cross-section, m\(^2\), L is the length of the working area, m; \( d_e = 4 \cdot A_{ch}/P \) is the equivalent diameter of the duct, m; \( \Delta T = T_w - T_f \) is the surface to the mean temperature difference of the flow; \( Q \) is the heat flux supplied from the surface, Watt, determined by colorimetric method and controlled by the magnitude of the electric current supplied to the heating film, taking into account heat energy losses; and \( A \) is the heat exchange surface, calculated without taking into account the surface extend by the dimples (relative to a smooth surface). The equivalent diameter of the duct \( d_e \) was used as the characteristic dimension, and the average temperature of the heat carrier in the duct was used as the characteristic temperature.

The expected uncertainty was 6% for the friction factor value calculation and 12% for the heat transfer coefficient value. The test measurement data for the flow in a duct with a smooth surface were compared to the values calculated by the widely used Blasius friction factor correlation: \( (\xi = 0.3164/Re^{0.25}) \), and Mikheev heat transfer correlation: \( (Nu = 0.018 \cdot Re^{0.8} (T_f/T_w)^{0.5}) \) for the turbulent air flow in the duct; \( T_f \) and \( T_w \) are the flow and the wall temperatures, K. Determining temperature is the air temperature at the entrance to the channel, determining velocity is the average mass velocity in the channel, and linear size is the hydraulic diameter of the channel. The maximal data deviation of 10-15% was assumed to be acceptable for further work.

3.1. Experimental results Hydraulic resistance

The determination of the overall friction factor values for the flow in a duct with a single row and multirow dimple arrangement on the bottom surface of the duct was carried out on the experimental
The working area had the length \( L = 198 \text{ mm} \), the width \( B = 98 \text{ mm} \), and the height \( h_{ch} = 1.5 \text{ mm} \) for the multirow dimple arrangement and \( L = 270 \text{ mm} \), \( B = 21 \text{ mm} \), \( h_{ch} = 3 \text{ mm} \) for the single row dimple arrangement.

**Figure 5.** Friction factor coefficients versus Reynolds number values (a) for the surface with the multi-row arrangement of oval-trench (b) and oval-arched (c) dimpled surfaces.

The comparison of the friction factor values for the flow over the surfaces with the oval-trench and oval-arched dimples \((h/b=0.25, l/b=7)\) with the multi-row dimple arrangement is presented in Figure 5. Similar results were obtained for the single-row dimple arrangement and presented in Figure 6. The comparison of results has revealed that the friction factor values for the flow over the surfaces with the oval-arched dimples are lower by 10-13% compared with the oval-trench dimpled surface, with the same geometrical parameters. Wherein, the friction factor values are at the same values as for the flow over the surface with the hemispherical dimples.
3.2. Heat transfer augmentation
Preliminary results of the infrared imaging comparison of the dimpled surfaces have revealed the better thermal performance of the oval-arched dimpled surfaces. The flow to surface temperature difference is found to be 5-20% lower than for the surface. Those findings are supported by the results of the experimental determination of the heat transfer coefficient values for the surfaces with the single- and multi-row arrangement of oval-arched and oval-trench dimples (Figure 7).

Analysis of the results, presented in Figure 13 reveals that the average heat transfer coefficients values for the surfaces with oval-trench dimples are 5-10% lower than the ones for the surfaces with the oval-arched dimples (h/b=0.33, l/d=7). Moreover, the thermal performance of the oval-arched dimple surfaces exceeds the heat transfer coefficient values specified for the surface with the hemispherical dimples for the same spot area of a single dimple.

![Figure 7](image)

Figure 7. Heat transfer coefficient values for the surfaces with a single row (b) and multi-row dimple arrangement of the oval-trench and oval-arched dimples

3.3. Thermohydraulic efficiency
The thermal performance of the dimpled surface was evaluated by comparing the values of the friction factor augmentation $\xi/\xi_0$, heat transfer augmentation $\text{Nu}/\text{Nu}_0$, and the thermal-hydraulic performance $E=(\text{Nu}/\text{Nu}_0)/(\xi/\xi_0)$. The subscript 0 denotes the smooth surface. The results of the comprehensive comparison are listed in Table 1. It can be noticed that the thermal-hydraulic efficiency values are almost identical for the single- and multi-row dimple arrangements.

| Surface type | Dimple type       | Relative dimple size | Flow parameters | Nu/Nu$_0$ | $\xi/\xi_0$ | $E_{\text{max}}$ |
|--------------|-------------------|----------------------|----------------|-----------|-------------|-----------------|
|              |                   | h/b                  | S/b            | l/d       | Re          | Pr             |                 |
| Multirows    | Oval-trench       | 0.25                 | 6              | 7         | 4·10$^3$-2·10$^4$ | 0.72 | 2.3-2.43 | 2.5-3 | 0.97 |
|              | Oval-arched       |                      |                |           |             |                 | 2.5-2.6 | 1.11 |
|              |                   | ~2.5                 | ~1.4           | ~1.25     | 1.37        |
| Single row   | Oval-trench       | 0.33                 |                | 3.2·10$^3$-9·10$^4$ | 1.2-1.55 | ~1.4 | 1.11 |
|              | Oval-arched       |                      |                |           |             |                 | 1.2-1.71 | ~1.25 | 1.37 |

The further experimental results on the friction factor values for the flow in the square ducts carried out by Micheev and Dushin were analogous to the findings presented above. The friction factor values were obtained for flow over the oval-trench dimpled surfaces (with h/b=0.25, l/b=4.5, $\varphi=45^\circ$, S/b=2) in the rectangle duct (h=10 mm). It was shown that the magnitude of the friction factor values...
for the flow over the with the two-row wedge-shaped systems of oval-trench dimples along and towards the flow differs by the factor of 1.6 to 2.25 (Fig. 8).

The experimental investigations on the friction factor determination for the flow for the single-row oval-trench dimple arrangements with $h/b=0.25$, $l_d/b=4.5$, $\varphi=45^\circ$, $S/b=2$ (Fig. 9) were carried out in order to extend the range of relative-notch steps shown in the Figure 6.

**Figure 8.** Friction factor coefficients versus Reynolds number values for the surface with the two-row wedge-shaped systems of oval-trench dimples along and towards the flow.

**Figure 9.** Friction factor coefficients versus Reynolds number values for the surface with the two-row wedge-shaped systems of oval-trench dimples along and towards the flow.

### Numerical simulation of dimpled channel flows

The dimple arrangements on the heat transfer surface have an impact on the flow in the duct. These effects were investigated by carrying out numerical research for the flow over the single-row dimple arrangements $h/b=0.25$, $l_d/b=7$, $\varphi=45^\circ$ in the duct with $h_d/b=1$ as it was recommended in [2-7, 9-11]. The comparison of the longitudinal velocity fields $u$ in the middle cross-section of the periodic numerical domain for the various value of $S/b$ in the duct is presented in Figure 10. The obtained results in Figure 10 were used to determine the isosurfaces of relative longitudinal velocity $u/u_m=1.3$ in the duct (Figure 11). The $u_m$ is the mass average flow velocity. The flow realignment could be noticed. The flow acceleration exists on the boundary layer outer frontier in the vicinity of the duct walls. Thus, the higher local heat transfer coefficients are presented in those areas which raise the overall heat transfer coefficient of the dimpled-surface (Figure 12). However, the flow realignment is related to the dimple arrangements. For the multi-row dimple arrangements the flow acceleration as well the moderate flow velocity decline were noticed. This affects the variance in the friction factor value magnitudes for flow over the surfaces with various dimple arrangements in Figures 5, 6, and 8.
3.4. Verification and recommendations
The comparison of the outlined values with the results of the numerical simulation [11] for similar flow and geometrical parameters has been carried out. The data comparison is presented in Figure 13. A good data value agreement has been observed. The friction factor augmentation value in the numerical simulation is found to be $\xi/\xi_0=1.39$ while the experimental measurements for $S/b=6$ reveal the value of $\xi/\xi_0=1.4$. The friction factor augmentation value in the numerical simulation is found to be $\xi/\xi_0=2.24$ while the experimental measurements for $S/b=2$ reveal the value of $\xi/\xi_0=2.14$. Thus, the data values difference is within 1-5%. The corresponding values of the heat transfer augmentation for the numerical and experimental measurements for $S/b=6$ are found to be $\text{Nu}/\text{Nu}_0=1.54$ and $\text{Nu}/\text{Nu}_0=1.55$, respectively.

**Figure 10** The longitudinal velocity $u$ fields in the middle cross-section of the periodic domain of the smooth (a) and dimpled surfaces for $S/b=8$ (b); $4$ (c); $2.667$ (d); $2$ (e) with imposed secondary flow structure.

**Figure 11** The isosurfaces of relative longitudinal velocity $u/u_m=1.3$ in the periodic domain with dimpled surfaces $S/b=8$ (a); $4$ (b); $2.667$ (c); $2$ (d).

**Figure 12** The relative local Nusselt number fields on the dimpled surfaces with various dimple arrangements. $a-\text{H}=8$; $b-4$; $c-2.667$; $d-2$. (with superimposed streamlines)

**Figure 13** The values of relative Nusselt number $\text{Nu}/\text{Nu}_0$ (1), friction factor $\xi/\xi_0$ (2), and the thermal-hydraulic performance coefficient $\text{Nu}/\text{Nu}_0$ (3) versus the relative notching step $S/b$ value. Points 4, 5, 6 are the experimentally determined values of $\text{Nu}/\text{Nu}_0$, $\xi/\xi_0$, and $E$ respectively. Points 7 and 8 are the neural network predictions of the $\text{Nu}/\text{Nu}_0$, $\xi/\xi_0$ values. Point 9- the $\xi/\xi_0$ values obtained by Micheev and Dushin.
3.5. Artificial neural network application
The literature analysis reveals the potential for the data prediction on the basis of the experimental results database [13,14]. The preliminary analysis was made to apply the artificial neural networks to calculate the friction factor and heat transfer coefficients for the flow over the dimpled surfaces.

The database contained 200 data points, presented in [2-7, 9-11]. A model of the artificial neural works contained two hidden layers of 6 and 4 neurons. The program code was written in Python 3.0 with the use of Keras and Tensor flow Libraries. The data were separated into test and training sets as 20 to 80% ratio to overcome the over-prediction problem. By means of the back propagation, the weight factors of the model were determined.

The geometrical parameters of the sample were used as the input parameters for the model. The mean square error over the whole database was within 5-8% for the friction factor heat transfer augmentation values both for the test and training datasets. However, a more comprehensive analysis is required.

4. A comparison with other surface heat transfer augmentation techniques
The comparison of the experimentally determined friction factor and heat transfer coefficient values presented above (Figures 5, 6, and 7) with the widely used surface heat transfer augmentation techniques was based on the method proposed in Khalatov [15]. The result of the comparison is presented in Figure 14. The result of the comparison is presented in Figure 14. It could be noticed that the thermal-hydraulic efficiency of the dimpled surfaces is relevant and correlates well with the values obtained for widely used turbulent promoters: rib turbulizators, pin fins, swirl chambers, dimpled channels, ducts with protrusions, ducts with surface roughness.

The thermal performance of the heat transfer surfaces could be evaluated also by the method proposed by Ligrany [16]. The result of the comparison is shown in Figure 15. It could be noticed that the oval-trench dimples are among the best heat transfer enhancement techniques for the value of the relative Nusselt number augmentation Nu/Nu0. The corresponded values are in accordance with the values obtained for the dimpled surfaces, surfaces with protrusions, ducts with the protrusions inclined to the main flow.

![Figure 14](image1.png)  
**Figure 14.** The thermal performance of the various surface heat transfer augmentation techniques versus the relative friction factor increase, based on Khalatov [15]

![Figure 15](image2.png)  
**Figure 15.** A relative heat transfer augmentation versus the relative friction factor augmentation for the ducts with the dimpled surfaces and ducts with protrusions based on Ligrani [16]
Conclusions
The experimental investigation on the friction factor and heat transfer coefficients for the flow in the ducts with the dimpled surface was made to validate the numerical results presented in [2-7,9-11]. The great potential for the oval-arched and oval-trench dimples over the hemispherical and oval dimples on the heat transfer surfaces is pointed out. The highest value of the thermal-hydraulic efficiency coefficient value is found to be E=1.11 for the flow over the multi-row oval-arched dimples arrangement on the duct surface. The effect is achieved due to the flow acceleration by the dimple geometry change from the oval-trench to the oval-arched dimple shape. Thus, the application of such type of dimple on the heat transfer surface yields the notable decrease in a heat exchanger size.

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References
[1] Rashidi S, Hormozi F, Sunden B, Mahia O 2019 Applied Energy 250 149–54
[2] Isaev S A, Leontiev A I, Mityakov A V, Pyshny I A 2003 Journal of Engineering Physics and Thermophysics 76(2) 31–4
[3] Isaev S A, Popov I A, Leontiev A I, Gul’tsova M E 2015 Technical Physics Letters 41(6) 606
[4] Isaev S A, Schelchkov A V, Leontiev A I, Gortyshov Yu F, Baranov P A, Popov I A 2017 Int. J. Heat and Mass Transfer 109 40–62
[5] Isaev S, Leontiev A, Chudnovsky Y, Popov I 2018 J. Enhanced Heat Transfer 25(6) 579–604
[6] Isaev S A, Baranov P A, Leontiev A I, Popov I A 2018 Technical Physics Letters 44(5) 398–400
[7] Isaev S A, Leontiev A I, Milman O O, Popov I A, Sudakov A G 2019 Int. J. Heat and Mass Transfer 134 338–58
[8] Patent RU 2 684 303
[9] Isaev S A, Gritskevich M S, Leontiev A I, Popov I A, Sudakov A G 2019 High Temperature 57(5) 797–800
[10] Isaev S A, Gritskevich M S, Leontiev A I, Milman O O, Nikushchenko D V 2019 Thermophysics and Aeromechanics 26(5) 697–702
[11] Isaev S A, Gritskevich M S, Leontiev A I, Milman O O, Nikushchenko D V 2019 International Journal of Heat and Mass Transfer 118737
[12] Patent RU 2 716 958
[13] Jambunathan K, Hartle S L, Ashforth-Frost S, Fontama V N 1996 International Journal of Heat and Mass Transfer 39(11) 2329–32
[14] Zdaniuk G J, Chamra L M, Walters D K 2007 International Journal of Heat and Mass Transfer 50(23–24) 4713–23
[15] Khalatov A A, Okishev A V, Onistchenko V N 2010 Industrial Heat Engineering 32(5) 5–12
[16] Ligrani P M, Blaskovich T, Oliveira V V 2003 AIAA Journal 41(3) 337–362