Comparative analysis of the results of experimental and numerical studies of thermal and hydraulic characteristics of micro-hilly and corrugated plate surfaces

I S Verbanov¹, I A Gulimovskiy¹, A L Svetlakov¹ and A R Lepeshkin¹,²,³

¹Central Institute of Aviation Motors, 2 Aviamotornaya Street, Moscow, 111116 Russia
²Moscow Aviation Institute - National Research University, 4 Volokolamskoe Shosse, Moscow, 125993, Russia
³National Research University «MPEI», 14 Krasnokazarmennaya, Russia, Moscow, 111250, Russia

E-mail: isverbanov@ciam.ru

Abstract. The experimental analysis of heat transfer in plate heat exchange channels, presented in this paper, was carried out on the installation of visualizing the temperature distribution on the surface of the electro-heated air flow cooled heat transfer envelope using a thermal camera. Experimental studies were carried out in a wide range of air flow. In the comparative analysis, the average deviation of the experimental values from the numerical simulation for the Nusselt criterion was about 15%. The worst systematic discrepancy between the Euler numbers (excess of the calculated losses over the experiment) is observed for the hydraulic characteristics of the hilly surface, but it does not exceed +22% in the transition region, decreasing to +14% in the region of developed turbulence. Within these errors, it is possible to use numerical 3d calculations for the development of databases of criteria dependences of corrugated and hilly heat exchange surfaces. To calculate the full-size heat exchangers obtained by calculation criterion dependences for different geometric characteristics of the models of corrugated and hilly surfaces are presented.

1. Introduction
The design of plate type heat exchangers (HE) allows you to flexibly change the combination of their thermal and hydraulic characteristics within the selected HE concept (fluids, flow diagram, layout restrictions-up to the size of the matrix). This variability is provided by a wide range of applied types of plate heat exchange surfaces, characterized by a large number of internal geometric parameters. According to the results of earlier studies, for a detailed variable design of the matrix of the plate HE, corrugated plates (Frenkel surfaces) [1-4], micro-hilly surfaces (widely used by such firms as Danfoss and similar "pillowcase" type surfaces [5-7] can be selected. The design of the heat exchange surface is reduced to the selection of specific geometric parameters of each of the considered surfaces with a comparative assessment of their efficiency as part of the HE. The method assumes a choice between geometries of heat exchange surfaces of various types, the thermal-hydraulic characteristics of which can be presented in a criterion form based on the results of model experiments or detailed 3D calculations. The available or newly obtained criteria dependences are validated and used in standard engineering methods for design and verification calculation of heat
exchangers by average parameters, which allows you to choose the best possible geometries of heat exchange surfaces under specified operating conditions and under specified design constraints.

2. Installation for testing electro heated envelopes of HE plate

A conceptual scheme for the study of temperature fields on envelopes of corrugated plates is shown in figure 1. The installation includes:
- shaded compartment with internal lighting to accommodate the work area;
- working area with an envelope of corrugated plates and air supply and exhaust units;
- air supply system with control elements and measuring section with washer or turbine flow sensor (TDR);
- a control and data acquisition system with a power source, a light controller and a computer with a primary processing program.

![Diagram](image)

**Figure 1.** The scheme of installation for research of temperature fields on the envelope of corrugated plates.

Experimental heat exchange characteristics of the base surfaces welded into a single envelope with a dimension of 90 × 60 mm were obtained on an installation with electric heating of the studied envelope and visualization of the temperature field using a thermal imager Testo 890-2.

The investigated range of Reynolds numbers $Re_h = \frac{u h}{\nu}$, in this case recorded for the height of the interlaminar channel $h$ as the characteristic size, velocity $u$ and kinematic viscosity $\nu$ when their characteristic values at the entrance to the envelope, made up $Re_h \sim$ (from 450 to 2150) for micro-hill and $Re_h \sim$ (from 450 to 2150) for corrugated surfaces, which apparently covers partially turbulent and transitional flow region. The test results for the validation procedure were presented as dependencies of the Nusselt $Nu_h = \frac{u h}{\lambda}$ number and Euler numbers $Eu = \frac{\Delta P}{\rho u^2}$ from the Reynolds number, where $\alpha$ is the average surface heat transfer coefficients, $\lambda$ is the coefficient of thermal conductivity of the medium at the average wall temperature, $\rho u^2$ is the velocity head at the entrance to the envelope, and $\Delta P$ - the pressure difference between the input and output collectors of the envelope.

The appearance of micro-hilly and corrugated plates and the experimental temperature fields of the tested envelopes are shown in figure 2. The height of the interplate channel in both manufactured envelopes was 1.6 mm, the angle of intersection of the corrugations (the angle of the unit cell for corrugations) is 120 °, the angle of the unit cell, in the corners of which there are micro-hills-90°.
The geometry of the plates was checked against the original drawing using 3D scanning of the manufactured plates. The micro-hilly envelope, in addition to welding along the periphery, was held together by spot welding at all the joints of the hills. Due to the inability to carry out such a fixation for the corrugated envelope, there was a residual gap between the tops of the corrugations, which according to the results of measurements was 0.3 mm (tests were carried out under conditions of negative pressure inside the envelopes, i.e. with air suction, and small values of compressive pressure in this case did not have a noticeable effect on the value of the residual gap).

![Figure 2](image1.png)

**Figure 2.** General view of the corrugated and micro-hilly plates (top) and the characteristic thermal distribution of temperatures on them during tests for $Re_h = 2000$ with a characteristic line $P1$, which was used for averaging the temperature (bottom).

### 3. Numerical study of thermos-hydraulic characteristics of micro-hilly and corrugated plate surfaces

Validation of numerical 3D calculations using experimental data was carried out on the calculated areas shown in figure 3, which geometrically fully corresponded to the Central area of the envelope with the width of an elementary cell and included the inlet and outlet collectors 1 and 3 with the heat carrier cavity 2, which allowed taking into account the edge effects of the flow input-output from the envelope. Limiting the calculation area to the sides allowed for sufficient grid resolution and reasonable counting time in the conditions of available computing power. The boundary conditions on the side surfaces of the calculated area were set according to the flow principle, which was a deviation from reality, since it described exactly only the Central area of the envelope (or an infinite plate in the direction perpendicular to the flow), and thus did not take into account the edge effects on the sides of the envelope. At the same time, the significance of such edge effects on the full-size plates of the designed heat exchanger will be significantly less, since the number of unit cells along the characteristic axes in it is an order of magnitude greater than in the model envelope. The heat supply in the calculations was set from the condition of constant heat flow $q_w=\text{const}$, which corresponds to the conditions of ohmic heating of the plates.
The peculiarity of the micro-hill model was the exclusion of the spot welding area from the calculation at the points of contact of hills (however, the results were averaged over the entire area of the plate). Calculations of corrugated envelopes for validation purposes were carried out taking into account the gap between the plates, but for further use in the design calculations of heat exchangers, the results obtained in a numerical experiment for fully closed plates were developed, assuming the development of special methods for screed / closing plates in a real design.

![Figure 3](image)

**Figure 3.** The calculated area for micro-hilly and corrugated surfaces in the numerical experiment (above) and the implemented calculated grid on the example of a micro-hilly surface (below).

Numerical experiments were performed using the commercial CFD-package ANSYS 18. The calculated models of the interplate space of micro-hilly and corrugated packages were divided according to the uniform method of hex-grid with the number of elements of the order of 15 million (see figure 3). The prismatic boundary layer consisted of 40-50 rows with a thickness of the first layer of the order of 0.5 microns and a growth coefficient of 1.15. In all areas of the calculation model under different modes, the parameter $y_{mesh} \leq 0.5$. The obtained grid resolution allowed not only to reliably calculate the wall parameters, but also to model secondary flows inside the elementary cells, preserving the flow structure at this scale.

Taking into account the detailed description of the boundary conditions, the choice of turbulence models, as well as the schemes of advection and transfer of turbulent characteristics described in [8], the calculation was performed in a stationary setting by the finite element method using the SST GTM turbulence model (Transition SST).
Validation comparison of experimental and calculated thermal-hydraulic characteristics microhabitat and corrugated surfaces in the form of $Nu_h(Re_h)$ and $Eu(Re_h)$ shown in figure 4.

![Figure 4](image)

**Figure 4.** Validation of 3D calculation of thermal-hydraulic characteristics based on experimental data.

The coefficients of the criterion of empirical dependencies of the type $Nu_h = A \cdot Re_h^B \cdot Eu = C \cdot Re_h^D \cdot L$, where $L$ - is the total length of the plate heat exchange matrix in the flow direction, normalized to 1 m, which were uniformly obtained on the basis of validated 3D models of calculation and were used in the design presented in table 1.

| Type and corner in the cell | Height of plate, mm | Step by thread, mm | Step across the flow, mm | A   | B   | C    | D    |
|----------------------------|---------------------|--------------------|--------------------------|-----|-----|------|------|
| Corrugation 60°            | 1.6                 | 4.8                | 2.66                     | 0.0949 | 0.6629 | 1346.0 | -0.446 |
| Corrugation 120°           | 1.6                 | 2.66               | 4.8                      | 0.4411 | 0.5092 | 692.1  | -0.093 |
| micro-hill 90°             | 1.6                 | 4.71               | 4.71                     | 0.0251 | 0.8093 | 288.9  | -0.078 |
| micro-hill 60°             | 1.6                 | 6.2                | 3.58                     | 0.3666 | 0.4900 | 471.2  | -0.302 |
| micro-hill 120°            | 1.6                 | 3.58               | 6.2                      | 0.1774 | 0.6029 | 553.2  | -0.134 |

4. Conclusion

The results of thermos-hydraulic experiments allowed us to validate the 3D calculation methods for calculating such heat exchange surfaces in terms of choosing the dimension of the calculated area. Validation allowed us to expand the number of considered options for calculating plates of different geometries in the vicinity of experimentally studied samples. It can be seen that there is a good convergence between the calculation and the experiment (no more than ±5% for the thermal characteristics of both plates and the hydraulic characteristics of the corrugated surfaces). The worst systematic discrepancy (excess of calculated losses over the experiment) is observed for the hydraulic characteristics of a micro-hilly surface but it does not exceed +22% in the transition region decreasing to +14% in the area of developed turbulence.
References

[1] Abou Elmaaty T M, Kabeel A E and Mahgoub M 2016 Corrugated plate heat exchanger review Renewable and Sustainable Energy Reviews 70 pp 852–860

[2] Ayub Z H, Khan T S, Salam S, Nawaz K, Ayub A H and Khan M S 2018 Literature Survey and a Universal Evaporation Correlation for Plate type Heat Exchangers International Journal of Refrigeration 99

[3] Raja B D, Jhala R L and Patel V 2017 Thermal-hydraulic optimization of plate heat exchanger: A multi-objective approach International Journal of Thermal Sciences 124 pp 522–535

[4] Jin S and Hrnjak P 2016 Effect of end plates on heat transfer of plate heat exchanger International Journal of Heat and Mass Transfer 108 pp 740–748

[5] Arsenyeva O, Tran J and Kenig E Y 2018 Thermal and hydraulic performance of pillow-plate heat exchangers 28th European Symposium on Computer Aided Process Engineering pp 181–186

[6] Arsenyeva O, Tran J, Piper M and Kenig E 2018 An approach for pillow plate heat exchangers design for single-phase applications Applied Thermal Engineering 147 pp 579-591

[7] Zhang Y, Jiang C, Yang Z, Zhang Y and Bai B 2016 Numerical study on heat transfer enhancement in capsule-type plate heat exchangers Applied Thermal Engineering 108 pp 1237–1242

[8] Gulimovsky I A, Verbanov I S and Svetlakov A L 2018 Numerical study of thermohydraulic characteristics of micro-hilly and zigzag surfaces of increased turbulence Industrial power engineering 2018 3 pp 26–31

[9] Andreev M M and Berman S S 1963 Heat-exchange equipment of power plants (Moscow: Mashgiz)