Calculation and experimental researches into the heat exchange surface formed by oppositely directed truncated cones with saddle-shaped connection straps

A Ye Baranov, D N Ilmov, V A Mavrov, Yu N Mamontov, A S Skorokhodov

SSC Keldysh Centre, Moscow, Russia
E-mail: ilmovdn@mail.ru

Abstract: The article has presented experimental data on the investigation of thermal and hydraulic characteristics of heat exchangers made of thin-walled panels with the surface formed by oppositely directed truncated cones with saddle-shaped connection straps. An approach to the mathematical description of surfaces of given class have also been proposed, and numerical modeling of stream and heat transfer have been performed. Results of numerical modeling have been compared with the experimental ones.

1. Introduction
At the present time a concept to the creation of the radically new spacecraft with nuclear power systems of 1 MW power and more is under consideration [1]. Special operating features of such a system imply high temperatures (over 1000 K) and big differential pressures (tens of atmosphere) between the heat-transfer agent circuits. In so doing, there are weight and size restrictions, which eliminate the use of conventional constructions for practical purposes. At SSC Keldysh Centre a new design based on stamped plates has been developed [2]. As the surface structure of heat exchanger plates is unique, the research on its thermal and hydraulic characteristics is urgent. For this purpose the "air-air" research with using heat exchangers with a different quantity of the plates was conducted. Progress of these experimental researches was carefully considered in [3].

2. Experimental
Shown in Fig. 1 is the schematic sketch of the plates and the panel for the creation of experimental cross-flow models of the heat exchangers.

![Fig. 1. The schematic relief of the plate (on the left) and the plate for the creation of heat exchanger models (on the right).](image)

The following designations were introduced: $2\ell$ – the spacing between the adjacent bulges on the plate (8 mm); $H$ – the height of the plate after stamping (1.5 mm); $\delta$ – the thickness of the original
blank (0.2 mm); $h=H-\delta$ – the maximum height of the flow section. The coefficient of surface development (ribbing) for this plate is calculated by the formula:

$$k_F = \frac{F_1}{L_A L_B} = 1 + \frac{\pi}{4z} \left( \sqrt{h^2 + z^2} - z \right),$$  \hspace{1cm} (1)

where $L_A$, $L_B$ и $F_1$ are the width, the length and the area of the plate surface after the stamping.

Refined manufacturing process yields the surface development 3 – 6 %, besides; the specific shape of channels makes the flow turbulent, which ensures heat exchange intensification.

The equivalent diameter of channels shaped in matrix is defined by their summary volume $V$ and the total area of the “wetted” surface $F_F$:

$$d_e = 4V / F_F = 2h / k_F.$$  \hspace{1cm} (2)

In the heat exchangers, the heat-transfer agent, being under lesser pressure and greater temperature (hot), is supplied to the cavity A. The cavity A is formed by a space between the plates and the two collectors, formed by the two 12 mm - diam. openings and one 16 mm - diam. opening. Another heat-transfer agent circulates in the external cavity B. It comes to the plates from a segment formed by the outer case and the heat-exchange matrix. The path length of the heat-transfer agents - $L_A=88$ mm $L_B=94$ mm (Fig. 1). The coefficient of surface development $k_F = 1.04$; the equivalent diameter $d_e = 2.5$ mm. The effective heat exchange area was derived from the formula $F = k_F L_A L_B N$ (where $N$ is the number of panels). The areas of the flow section of the heat-transfer agents are $S_A = hL_B N_A$, $S_B = hL_A N_B$.

An experimental installation providing the air pumping at the mass flow rate up to 500 g/s and a feasibility of its heating to 600 °C, was made for testing of the heat exchangers. Experiments were carried out in the steady-state mode in which the air mass flow rate $G$, temperatures and static pressures at the heat-exchanger inlets/outlets were recorded.

On mathematical processing of results it has been suggested that the air is an ideal gas with the gas constant $R = 287$ J/(kg K). Thermal and physical properties were believed to be dependent on the mean temperature in the hot and the cold cavities from the data [4], the metal thermal conductivity $\lambda_{ME} = 16$ W/(mK). The amount of transferred heat per time unit $Q$ was determined from the thermal balance. The thermal efficiency was further determined:

$$\eta = Q / Q_{ID},$$  \hspace{1cm} (3)

where $Q_{ID} = c_{pc} G (T_{H1} - T_{C1})$ is the ideal heat power ($c_{pc}$ is the heat capacity of the cold heat-transfer agent, $T_{H1}$ and $T_{C1}$ are the inlet temperatures of the hot and cold heat-transfer agents). At the known thermal efficiency, the quantity of units of carrying over of warmth $Z$ (NTU) was determined for a single crossed current from solving the following equation [5]:

$$\eta = 1 - \exp \left\{ \left[ \exp \left(-Z^{0.78} \right) - 1 \right] Z^{0.22} \right\}. $$  \hspace{1cm} (4)

After determination of the quantity of units of carrying over of warmth, the overall heat transfer coefficient was calculated:

$$K = Zc_{pc} G / F.$$  \hspace{1cm} (5)

On the other hand, the common heat transfer coefficient is defined by the convective heat exchange factors $\alpha_A$ and $\alpha_B$ of the heat-transfer agents and the thermal resistance of the plate, $\delta / \lambda_{ME}$, by the formula:

$$K = \left( \frac{1}{\alpha_A} + \frac{1}{\alpha_B} + \delta / \lambda_{ME} \right)^{-1}. $$  \hspace{1cm} (6)
For the average values of the convective heat exchange factors, it was assumed that $\alpha_A = \alpha_B = \alpha$. This approximation is completely justified, as $s_A \approx s_B$, and the mass flow rates of the heat-transfer agents were equal.

Two closely related dependences in the dimensionless form were obtained: $\mathrm{Nu}_A = \frac{ad_E}{\lambda_A}$ as a function $\mathrm{Re}_A = \frac{Gd_E}{(s_A\mu_A)}$ and $\mathrm{Nu}_B = \frac{ad_E}{\lambda_B}$ as a function $\mathrm{Re}_B = \frac{Gd_E}{(s_B\mu_B)}$. Through interpolation the final result was represented in the form of the general averaged dependence $\mathrm{Nu}/\mathrm{Pr}^{0.4} = f(\mathrm{Re})$, where $\mathrm{Pr} = \mu c_p / \lambda$ is the Prandtl number.

The drag coefficient was calculated by experimental results separately for the hot and the cold cavities on a basis of the generally accepted dependence:

$$\Delta p = \xi \frac{pU^2}{2} \frac{L}{d_E},$$

where $U = G / (\rho S)$ is the mean – average velocity, $\xi$ is the required drag coefficient, $\Delta p$ is the pressure losses.

Apart from experimental determination of dimensionless characteristics of surfaces for the heat exchangers, the task of development of mathematical modeling methods was set. Numerical modeling was grounded on solving Navier-Stokes averaged equations, that are also referred to as Reynolds equations [6]. Parameters in the equations (velocity, density and others) are time-averaged. The steady-state continuity equation, the equations of motion and energy for the compressible medium (ideal gas) are solved. Averaging the Navier-Stokes equations and the energy equations leads to the appearance of unknown terms in them, related to turbulent pulsations. To close a set of equations, the Boussinesq hypothesis of supplemental turbulent viscosity and heat conduction is applied. The turbulent viscosity: $\nu_T = \rho v_T l_T$, where $v_T$, $l_T$ are the velocity scale and the spatial scale of turbulent pulsations. For determination of the latter, the semi empirical model $k - \epsilon$ of the second closure order is used. It suggests a solution of the two transfer equations for kinetic turbulent energy $\langle k \rangle$ and for turbulent dissipation rate $\langle \epsilon \rangle$. The form of the equations of the $k - \epsilon$ model and values of semi empirical constants can be found e.g. in [7]. An analysis of the efficiency of the application of various models of turbulence to the assigned task was not within the scope of this article. However, we note that the application of various models, including "large eddy simulation", as well as the direct solution of Reynolds equations without turbulent components, led to very close results.

Solving the set of the Reynolds equations and the equations of the $k - \epsilon$ model is labour-consuming enough, but a well developed problem. Let us consider the application of this mathematical instrument to the task set.

For numerical modeling the two approaches can be proposed: thorough reproduction of geometry (3D scanning), which demands great computational expenditures or search for simplified analogues. As an equivalent, a columnar surface with the “contracted” section, similar in the area and hydraulic diameter, is here proposed (Fig.2).
Fig. 2. Construction of the equivalent design domain by the “contracted” sections (1 – the “contracted” section, 2 – the section of the design model).

Fig. 3 shows the velocity and temperature fields, obtained with the aid of the given simplified model. The solution was fulfilled with using the software package FLUENT ANSYS. It is seen that the jet flow pattern, uniform in width, is being formed rather quickly.

Fig. 3. The velocity and temperature fields as a result of the modeling.

Given in Fig.4 are the values of the drag coefficients for the cavities A and B, experimentally obtained through the numerical modeling on the 20-panel heat exchanger. The drag coefficient of the inner cavity A is higher 1.5 to 1.6 times, than that of the cavity B. The distribution of experimental points needs some explanation. In the process of the research, it became clear, that the drag depends on the pressure differential between the cavities. This differential increases with the flow rate. One series of experimental data was acquired at the start of the installation, when the inner cavity A is contracted and its drag coefficient has higher values, the outer cavity B is extended and its drag coefficient has relatively low values. Another series of experiments was conducted at the start, when the pressure differential between the circuits was opposite. It is seen, that a deformation of the heat
exchange matrix changes the drag of the circuits A and B by 15 to 20%. The approximation lines of experimental points were built for both series of the data. Just these curves should be compared with those obtained by the modeling. Thus, the modeling results accord well with the experimental results. It is precisely the channels “contracted” section, that supposedly governs pressure losses.

![Image](image_url)

**Fig. 4.** The dependence of the drag coefficient on the Reynolds number (Re) for the 20-panel heat exchanger: 1 – the circuit A; 2 – the circuit B; 3 – the modeling for the circuits A and B.

Presented in Fig.5 are the values of the dimensionless heat-transfer coefficient $\frac{Nu}{Pr^{0.4}}$ at different Reynolds numbers, obtained in the experiments on the 2-, 4-, 20- and 40 panel models of the heat exchangers. The empiric results agree closely with the approximating function and between each other:

$$Nu=0.18 Pr^{0.4} Re^{0.69}$$  \hspace{1cm} (8)

The dependence obtained by the modeling accords satisfactorily with the experimental results. At the Re numbers less, than 2500, the calculation yields the overstated Nu number; at the great Re numbers – the understated. Nevertheless, in the range $2000 < Re < 4000$ the modeling error does not exceed 10%.

It may be inferred that the proposed approach to the simplified construction of the design domain of heat exchange surfaces, based on the equality of the flow areas and the equivalent diameters of the contracted section, actual and model channels, has been completely justified in case of calculation of the drag coefficient, and may be as a whole considered applicable to the modeling of heat exchange.
3. Conclusion
Thus, the computational and experimental research into thermal and gas dynamic characteristics of stamped plates with the specific relief of the surface was performed. The dependences of the dimensionless heat-transfer coefficient and the drag coefficient on the Reynolds number were obtained. The mathematical model for research of the working process regularities in energy-intensive heat-exchange devices of new-generation space systems was developed and verified.

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