

## Experimental study of parameters of surfaces coated with regular relief

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## Introduction

Recently, in connection with constantly growing prices of energy resources vital importance is imparted to the efficiency of production and consumption of the energy, as well as to energy saving techniques. A most frequent solution in this sphere is the installation of heat exchange skids (utilizer boilers, recuperators, waste gas heaters). Naturally, the requirements to the heat exchange equipment are similar to those of the main equipment: they are high efficiency and reliability, failure freedom etc.

For fluid heat exchangers the power consumption for overcoming of friction and resistance (circulation flow energy) occurring during the fluid flow through the heat exchanger are generally negligible as compared with the respective calorific transfer. Accordingly, the influence of the consumed power for compensation of friction and resistance are rarely a crucial factor. But as for gaseous media heat exchangers the consumption of the mechanical energy for friction overcoming can rather easily reach values comparable with the value of the calorific transfer. Thereby it is necessary to remember that in the majority of industrial plants the mechanical energy “costs” are by several times (usually 3-5) more “expensive” than the equivalent thermal energy.

It is easy to demonstrate that for the majority of channel shapes used to construct heat exchange surfaces the specific thermal load of the surface can be increased by the velocity increase of the coolant and the thermal load change is proportional to the velocity change raised to power close to 1.

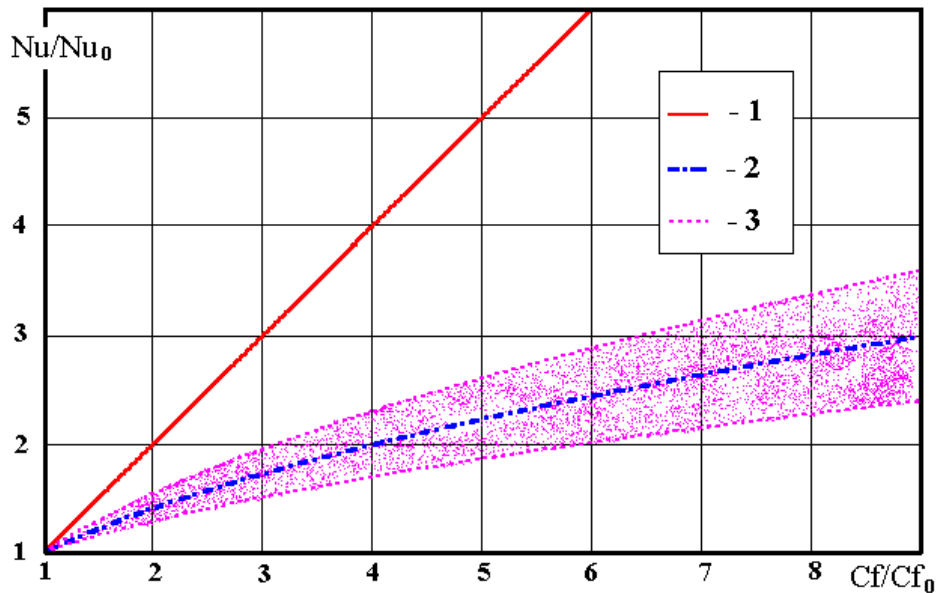
The energy input for the overcoming friction is also increased along with the growth of the flow velocity but it is generally changed proportionally to second power of the velocity. This is the characteristic feature of the correlation of the parameters of the calorific load and the friction (power for coolant pumping) which determines many features of heat exchangers of different classes. To compare surfaces with various ways of the heat exchange intensification a correlation is introduced between the efficiency growth of the heat transfer (correlation of Nusselt number on a surface with heat transfer intensifiers with Nusselt number on a smooth surface) and the resistance growth with the introduction of transfer intensifiers. This correlation is called Reynolds analogy with the following generalized appearance

$$\kappa = \frac{Nu/Nu_0}{C_f/C_{f0}},$$

where  $Nu, C_f$  is Nusselt number and friction value of a surface with heat transfer intensifiers,  $Nu_0, C_{f0}$  is Nusselt number and friction value on a smooth surface,  $k$  is the criterion of Reynolds analogy. For air flow  $k=1$  is often taken, accordingly, Reynolds analogy may have the following appearance:

$$Nu/Nu_0 = C_f/C_{f0}.$$

It is known that almost all intensification ways are excellently described by Nunner curve ( $Nu/Nu_0 = \sqrt{C_f/C_{f0}}$ ) the diagram, see Fig. 1. Ibidem, see the diagram of the simplest variant of Reynolds analogy and the area of known experimental data (except for vortex intensification methods).



1 – Reynolds analogy  $Nu/Nu_0 = Cf/Cf_0$  ; 2 – Nunner curve  $Nu/Nu_0 = \sqrt{Cf/Cf_0}$  ; 3 – area of known experimental results (except for vortex intensification methods)

Fig. 1 – Influence of heat transfer intensification ways on the correlation  $Nu/Nu_0$  dependent on  $Cf/Cf_0$

To reduce the power consumption for the overcoming of the friction one may lower the flow velocity and increase the heat transfer surface which in its turn will also lead to a power consumption increase for coolant pumping but proportionally to the heat transfer surface growth. But usually this way is not advisable for economic reasons since the price of heat transfer equipment (due to considerable metal consumption and expensive materials) for the time being may be comparable with the price of the whole power plant.

For the solution of this problem there is apparently only one way – heat transfer intensifying at equivalent or lagging resistance growth.

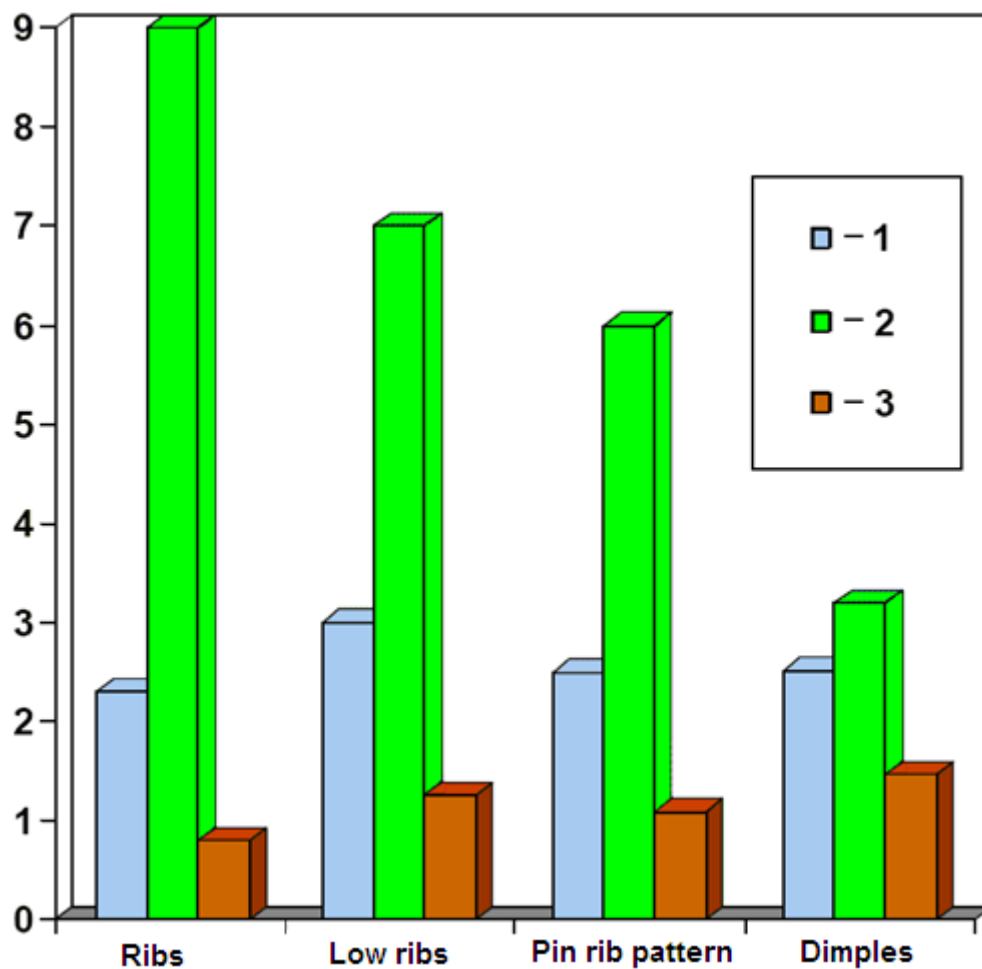
## 1. Analysis of specialized literature data on investigation of Reynolds analogy

One of the most effective methods to increase the efficiency of power plants is the intensifying of the process of mass heat exchange allowing for the reduction of heat exchange equipment, accordingly, its price. The most acceptable heat exchange intensifying method in heat exchangers is the use of various flow turbulizers. There are various types of heat transfer intensifiers realizing this approach. The diagram in Fig. 2 shows the relative growth of the hydraulic resistance, the heat exchange intensification and the thermal-hydraulic efficiency of tested surfaces as compared with

smooth surface. As a parameter of the thermal-hydraulic efficiency the equation derived from the equation for Nunner curve was taken

$$k1 = (Nu/Nu_0) / (\sqrt{C_f/C_{f0}}).$$

Fig. 2 shows that the use of heat transfer intensifiers usually leads to a significant growth of the hydraulic resistance. The more interesting are the results obtained on heat exchange surfaces with vortex type intensifiers (“dimples”) since in this case sufficient increase of the heat exchange rate is accompanied by a practically equivalent proportional increase of the hydraulic resistance. Due to these features surfaces with vortex type intensifiers have been an object of keen interest in research centers all-over the world. There are numerous papers dedicated to experimental and numerical studies of such surfaces (thereof the fundamental works are [1, 2, 3, 4, 5]). Thereby there is still no answer to the question of optimal (from the point of the thermo-hydraulic efficiency) shape although one of the pathfinder investigations of the use of dimples punched onto the surface (hemispherical cavities) for heat exchange intensification appeared already in 1961 [6]. All this indicates necessity for additional research work.



1 – heat exchange intensification as compared with smooth surface; 2 – hydraulic resistance growth as compared with smooth surface; 3 – thermal-hydraulic efficiency parameter  $kI$ .

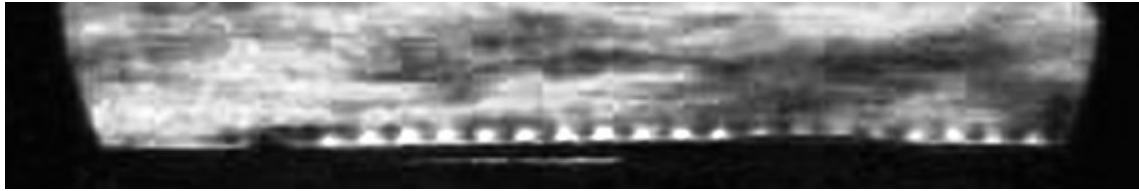
Fig. 2 – Thermal-hydraulic efficiency of surfaces with various heat transfer intensifiers

When gas or fluid passes dimple type cavities large-scale dynamic vortex structures arise which are observable both in laminary and turbulent flow mode. Such structures are observed also at low subsonic velocities (at Mach number  $M < 0,3$  when the compressibility of the media is negligible), and at high subsonic velocity, and supersonic velocity (when the compressibility of the medium cannot be neglected).

The presence of vortex structures on the heat exchange surface leads to the disturbance of the border layer, accordingly, the heat exchange processes are intensified. It is natural that the occurrence of disturbing factors in the flow influences the flow hydrodynamics. Shadow patterns of washing of the surfaces with a regular relief pattern by air flow, see Fig. 3.

Fig. 3.a shows subsonic washing of the surface. As apparent in the picture, in this case there is a visible optical non-uniformity over the dimples in a shape of "columns". This phenomenon can be

explained by vortices in the dimples hovering over them and providing for optical non-uniformity. The photo demonstrates that the height of a vortex column over a dimple has comparable dimensions with it.



a.



b.

a – washing of a plate with regular relief pattern by subsonic air flow; б - washing of a plate with regular relief pattern by supersonic air flow.

Fig. 3 – Shadow pattern of washing of a plate with regular relief pattern (dimples) by air flows

Fig. 3b shows a photo of washing of a plate with a regular relief pattern by supersonic air flow. It is visible that the slanting density leaps (the angle of the density leaps is almost equivalent to the characteristics angle at given Mach number) are located between the dimples, and due to the small distance between them one may assume that they are located before the front edge of the subsequent dimple. This fact substantially complicates the gas flow pattern along a dimpled plate.

It is known that the generated vortices cause intensive heat exchange between the border layer and the would-be flow core. There are three patterns of vortex generation on the obstacle surface.

The first pattern is when vortices occur due to the unstable border layer on a concave wall – they are so-called Taylor – Gertler vortices.

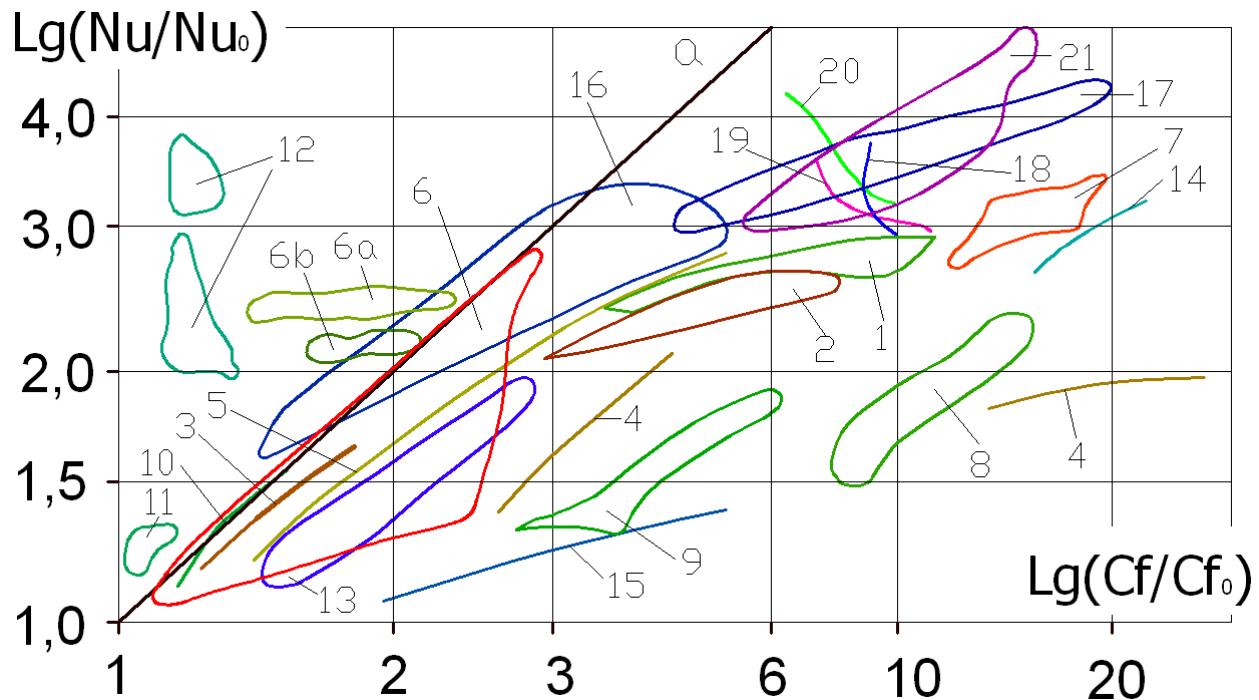
The second pattern is when vortices are generated in a non-viscous gas. The system of vortex cords belongs here, too, which are formed in rotating fluid (gas) and vortices on the rear edge of a wing of limited span.

The third pattern comprises the mechanism of vortex formation due to the presence on the surface of vortex-causing elements: cavities, hubs, grooves etc. In particular it is the case when vortices

are generated at washing of a relief in the shape of hemispherical cavities (dimples) and projections (see, e.g. papers [7, 8, 9, 10]).

Surfaces with vortex-forming relief enable a considerable increase of the heat transfer rate along with moderate resistance growth. A number of works provide experimental proof that in case of hemispherical cavities the heat transfer rate increase is not accompanied by the typical second power increase of the hydraulic resistance (especially evident such resistance increase is in case of slot-shape channels). A comparative analysis diagram of different heat exchange intensification ways (data from the paper [3]), see Fig. 4. Ibidem there are data of other works.

From the multitude of other intensification ways the most remarkable is the use of various surface cavities (dimples) whose high heat exchange rate and relatively simple manufacturing technology is combined with low flow resistance and excellent thermal-hydraulic characteristics.



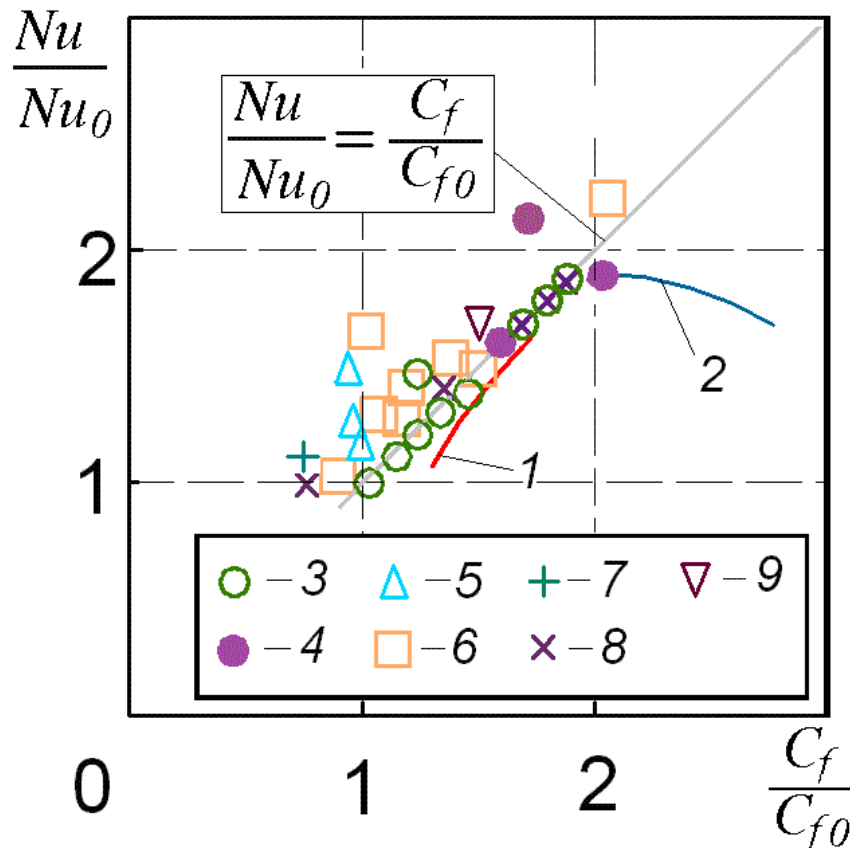
1 – internal ring-shape finning; 2 – spheric knobs; 3 – twisted bands; 4 – cylindrical pins; 5 – vortex matrix [11]; 6 – hemispheric dimples [12]; 6a – spherical cavities [13]; 6b – spherical cavities [Moon H.K., O'Konnel T., Glezer B.]; 7 – internal spiral grooves; 8 – alternate spherical knobs and cavities; 9 – internal grooves; 10 – screw-shape tubes; 11 – riblet; 12 – spherical cavities (boiling); 13 – channels of alternating cross-section; 14 – zigzag channels; 15 – curved channels; 16 – oval twisted tubes; 17 – whirled flow inside a tube; 18 – 90° cross-cut fins [13]; 19, 20 – 60° continuous and cross-cut fins [13]; 21 – internal cyclon-type cooling (three-channel loop system); a: Reynolds analogy line.

Fig. 4 – Comparative analysis of different heat exchange rate intensification types

Thermal and hydraulic characteristics of surfaces with spherical cavities depend on numerous factors, e.g. on the form of the dimples (with sharp or smooth edges), their density, longitudinal and cross-pitch, relative depth, relative channel aperture etc. Beside that the hydraulic resistance and the heat transfer in the channels are influenced by presence and relative location of dimples on adjacent planes.

Inspite of numerous experimental and numerical investigations there are still no trustworthy dependences and calculation recommendations as to dimpled surfaces ensuring acceptable accuracy in a wide range of parameter values.

The conducted analysis of publications demonstrated that the most studies of hydraulic resistance and heat exchange intensification had been based on shallow spherical cavities (dimple depth  $h$  vs. dimple dia.  $d$  up to  $h/d < 0,2$ ). As visible from Fig. 5, the main array of the experimental points obtained in such cavities is concentrated in the vicinity of the line  $Nu/Nu_0 = C_f/C_{f0}$ .

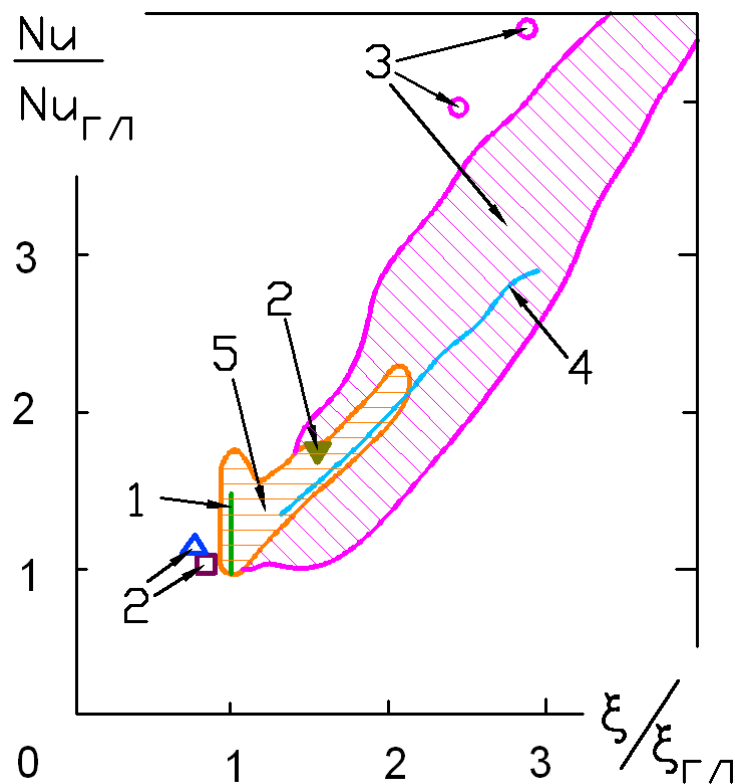


1 - [14]; 2 - [15]; 3, 4 - [16], [17] for unilateral location of smooth-edge cavities and bilateral location of sharp-edge cavities respectively; 5 - [18]; 6 - [19] coaxial ring channel; 7, 8 - [20] tube bundle in staggered and corridor order respectively; 9 - [21].

Fig. 5 – Heat exchange intensification efficiency of spherical cavities (at  $h/d = 0,1..0,2$ )



Fig. 6 demonstrates the results of studies of dimpled surfaces of different configurations [22]. In that study instead of the friction values correlation  $C_f/C_{f0}$  the correlation of corresponding hydraulic resistances  $\xi/\xi_{\Gamma\Delta}$  was used.



1 - V.N. Afanasyeva et al.; 2 - M.Ya. Belen'ky et al. (staggered and corridor order tubes bundle); 3 - Yu.F. Gortyshova et al.; 4 - G.P. Nagogi et al.; 5 - M.Ya. Belen'ky et al.

Fig. 6 – Results of studies (flat channel) for surfaces with spherical cavities.

However, as follows from the analysis of the studies performed theoretical and experimental thermal-hydraulic efficiency significantly differ (for details, see [23, 24]). In connection with the set forth above there is a necessity for new experimental studies. It is especially vital for non-spherical dimples. The analysis of the results of numerical studies (e.g., see [25, 26, 27, 28, 29, 30]) of the heat exchange intensification demonstrated that a replacement of spherical dimples by prolate (oval dimples) brings a considerable heat exchange growth along with considerably lagging resistance growth.

## 2 Description of the test fixture and research methods

In order to obtain data on heat exchange parameters and resistance of surfaces with vortex-causing relief the experimental bench had to be upgraded.

**Experimental test fixture and measurement methods.** Under recognition of existing methods of experimental definition of thermal-hydraulic features of relief surfaces the following test fixture was built-in into the bench to measure the parameters of heat exchange and resistance, see Fig. 7.

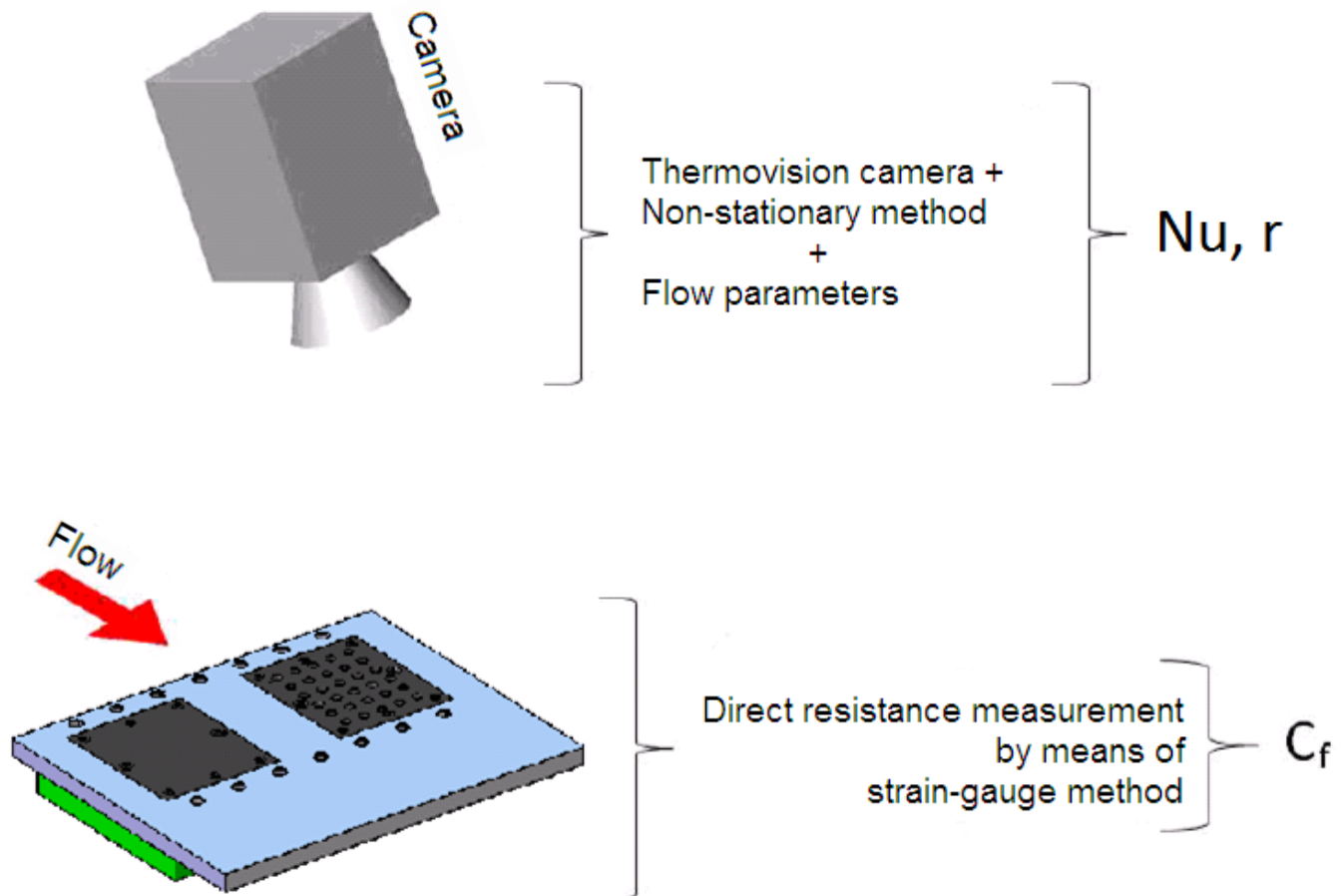


Fig. 7 – Arrangement of parameter measurement of heat exchange and flow resistance during the experimental research

The flow resistance of surfaces is determined by the method of direct force measurement for what a special weight cell was made.

The heat exchange parameters are measured by thermal imaging equipment thereby the heat transfer value is calculated after the cooling tempo of the surface (non-stationary method), and the

temperature recovery value was calculated after the wall temperature and the flow parameters (for compressible gas flow).

Special about such experimental fixture is that the measurement equipment does not disturb the flow and does not pierce the material of the plates tested. Thereby the indicated parameters are measured in a single course for two surfaces (one of which has a relief pattern, the other being smooth).

On this stage the weighing method of the plate was adjusted allowing high-precision measurement of hydraulic surface characteristics. The adjustment of the methods of the definition of the values of the temperature recovery factor  $r$  and Nusselt criterion  $Nu$  (heat transfer factor) had already been performed and described in the papers [31,32, 33, 34, 35].

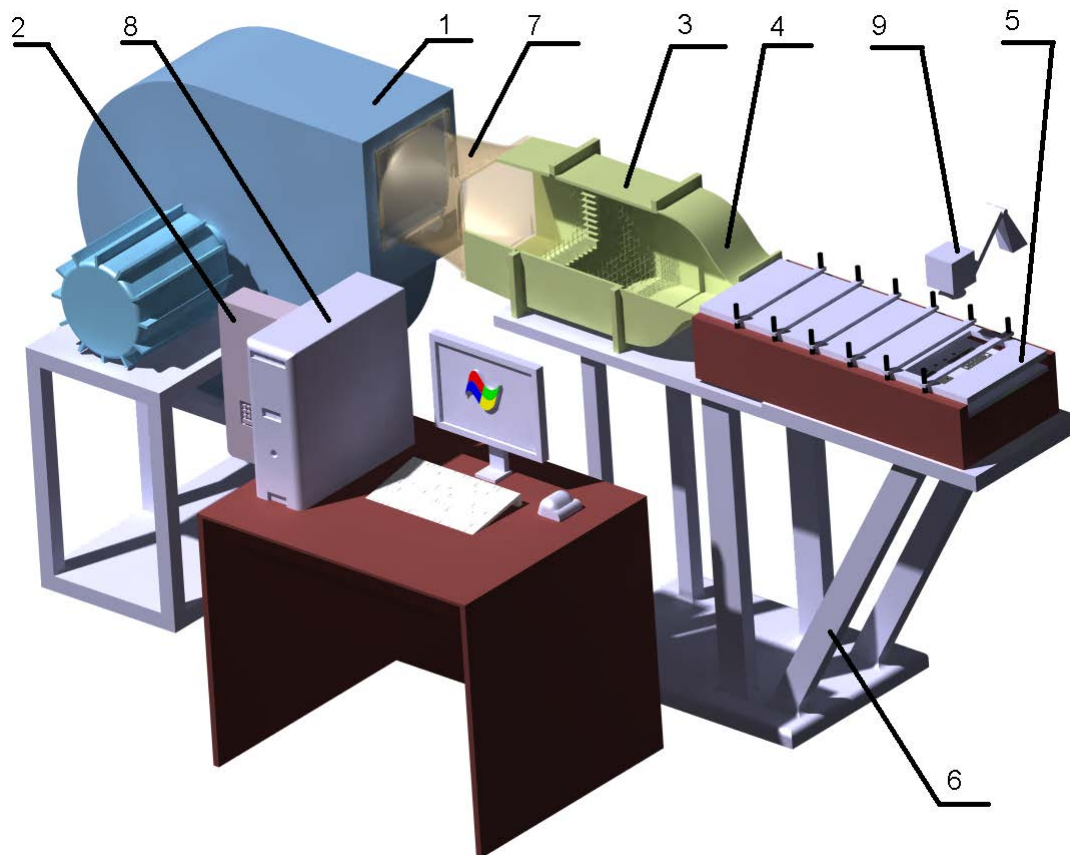
Thereby the value of Nusselt criterion (thermal transfer factor) is essential for the efficiency analysis of the thermal transfer whereas the definition of the temperature recovery factor  $r$  is necessary to monitor the heat transfer specifics of compressible media flows (at Mach numbers  $M < 0,3$   $r$  is practically unmeasurable). This monitoring is necessary, e.g. during the calculation of a gasdynamic temperature stratification (for details, see [36, 37, 38, 39]).

The experimental studies were conducted in the Scientific Research Institute of Mechanics of Moscow State University on the basis of the small aerodynamic test bench working after pressure supply principle. The arrangement of the test fixture, see Fig. 8, and the exterior, see 9.

The fixture comprises the following elements (see Fig. 8):

- High-pressure centrifugal fan (BIQ6-20-8-01, “MOVEN” Company, Russia) (1);
- Frequency converter (VFD450F43A, “Delta Electronics, Inc”, China) (2) for smooth rpm control of the fan and air flow velocity variation;
- Flow conditioning chamber (3) with two de-turbulization grids and a honeycomb for destruction of vortex structures and obtaining of a uniform air flow in the channel);
- Nozzle (4);
- Measurement channel (5);
- Measurement channel frame (6) detached from the fan to reduce the impact of the fan’s vibrations on the measurement channel);
- Flexible joint (7) between the fan (1) and the flow conditioning chamber (3) for damping of vibrations and breach of a rigid mechanical connection between the fan and the measurement channel;
- PC (8) with a data collection and processing system based on equipment and software by National Instruments (USA);

- Thermal vision camera (9) TernaCAM SC3000 (Sweden) for studying of thermal surface characteristics.



1 – fan; 2 – frequency converter; 3 – flow conditioning chamber; 4 – nozzle; 5 – measurement channel; 6 – support frame of the measurement chamber; 7 – connection hose; 8 – PC with data collection and processing system; 9 – thermovision camera.

Fig. 8 – Test fixture arrangement based on the small aerodynamic test bench of the SRI of Mechanics of MSU)

The measurement section of the fixture is a flat channel 1 m long, 0,3 m wide and 0,03 m high. For placement of the test plates both in the turbulent section and in the start flow section the bottom wall of the channel is made sectional consisting of three equal parts. Instead of one of the sections the measurement section is built-in.





Fig. 9 – photo of the small aerodynamic test bench of the SRI of Mechanics of MGU

The use of up-to-date measurement equipment, advanced software and PC allows monitoring the above-listed parameters in the real-time mode.

**Data collection and processing system.** The collection and processing of the experimental data is performed by means of specialized up-to-date equipment consisting of a connector unit SCB-100, analogue-digital transducer NI-1071E, (by National Instruments, USA), interfaced with a high-performance PC. For data processing LabVIEW software is used allowing for real-time data registration and processing, and for real-time visualization of the measurement. For accuracy enhancement the data reading from any measurement transducer is performed at a rate of  $1000\text{ s}^{-1}$  with the output of the average signal level value. LabVIEW (Laboratory Virtual Instrumentation Engineering Workbench by National Instruments, USA) is both programming environment and program execution platform created in the graphical language “G”.

For pressure data measurement transducers by Honeywell International (USA) were used, for temperature measurements cold-soldered L-type thermocouples are used, temperature-controlled at 0°C.

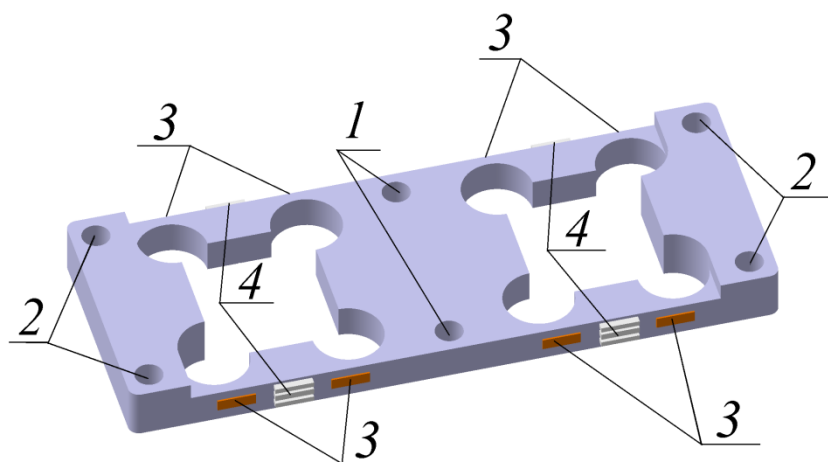
In the course of the experiments the following flow parameters were measured: temperature and pressure (after static and slowing parameters) at the channel intake and at the intake of the measurement section. Beside that forces were measured on the elastic element of the strain-gauge scale.

For the function of the strain-gauge scale spaces are mandatory between the tested surface and the flow part. Whereas the pressures in front and at the rear of the experimental section are different the spaces were provided with pressure measurement nipples (3 per each face edge of each plate) for correct monitoring of the force arising from the pressure difference on the front and rear edges of the test plates. After the values had been obtained corrections were introduced.

The elastic element of the strain-gauge scale was developed for resistance measurement of both smooth and relief surfaces. The form of the elastic element is a double parallelogram (structure, see Fig. 10; exterior, see Fig. 11) on each half of which either a smooth or the tested plate (with surface relief pattern) can be mounted whereas the central part is fastened to the bench's ground frame (see Fig. 12). Based thereon each tested surface can be fastened to a horizontal two-point support carrier which can only be deformed in one direction (one-component strain-gauge scale).

The elastic element had been machined with high precision in a CNC milling centre. As material of the elastic element alloyed steel 38XC with high chromium content was selected to provide for good elasticity and long life at recurrent force impacts.

The strain-gauge sensors are glued to four flexible elements (in the places of the largest deformation) of each parallelogram whereby two sensors are extended and two of them are compressed thus constituting a complete measurement bridge (see Fig. 10).



1 – fastening holes; 2 – connection holes for test plates; 3 – strain-gauge sensors; 4 – measurement bridge.

Fig. 10 Elastic element structure



Fig. 11 Photo of the elastic element of the strain-gauge scale

Whereas the elastic element deformation values are equal (max. 0,1 mm at max. attack flow velocity) the signals of the floating element were amplified by means of a specially developed and manufactured amplifier (10x gain).

The dimensions of the test surfaces were 100 x 125 mm. The plates were made of acrylic glass (for accuracy enhancement of the heat transfer measurement a material with low thermal conductivity was necessary). Prior to each measurement the working surface of the plates and the bottom channel surface were ground-in together and coated with special paint to obtain the required black surface grade.



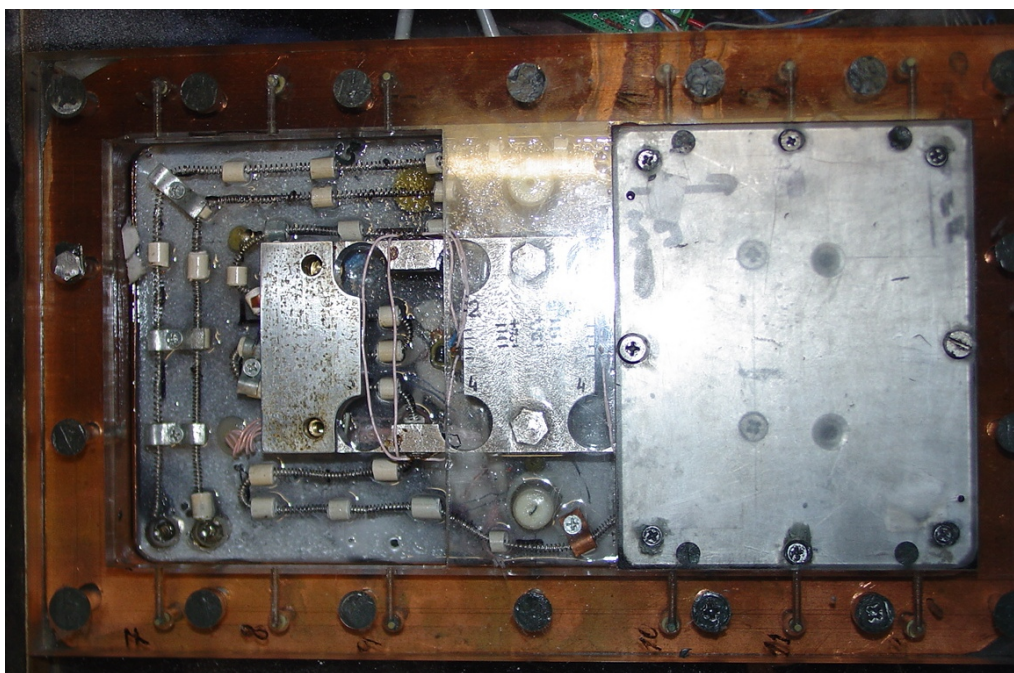


Fig. 12 Measurement section photo with the elastic element of the strain-gauge scale and one (smooth) plate (prior to grinding-in and painting)

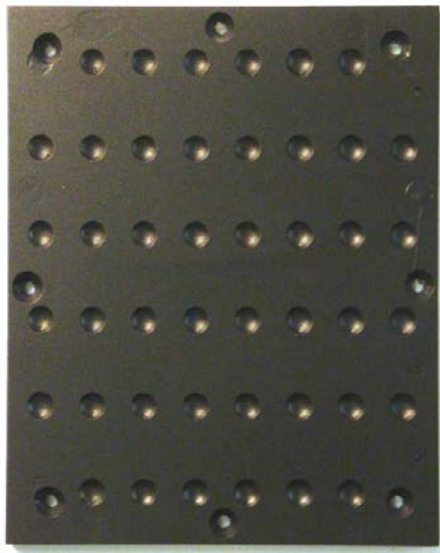
### 3 Experimental results

Immediately before the experiment was conducted with relief surfaces experiments with two smooth surfaces had been carried out. Subjects of measurement were the velocity profile, the influence of the bench vibrations, the measurement method was adjusted. Test measurements showed that resistance and heat transfer values of the smooth surfaces were in line with the calculations after commonly known regularities.

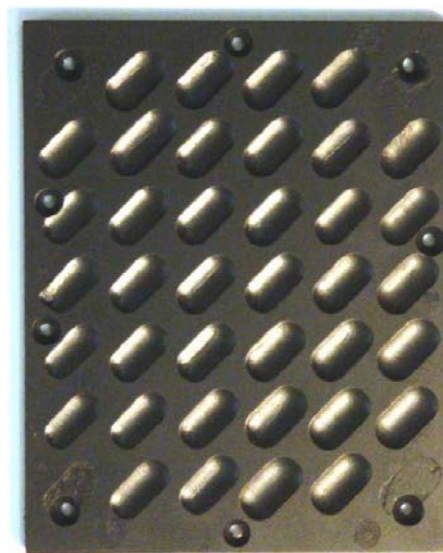
The adjustment of the "weighing" of the plate was made for it to be capable of high-precision (about 5% and better) measurement of the hydraulic surface characteristics. For the verification experimental data were used obtained before the upgrade of the test bench and presented in the paper [40].

During the experiment 2 relief patterns were studied: hemispherical dimples with corridor arrangement and oval dimples. The surface geometry was in line with the data presented in the paper [40]. The exterior of the test surface, see 13.





Surface No.1



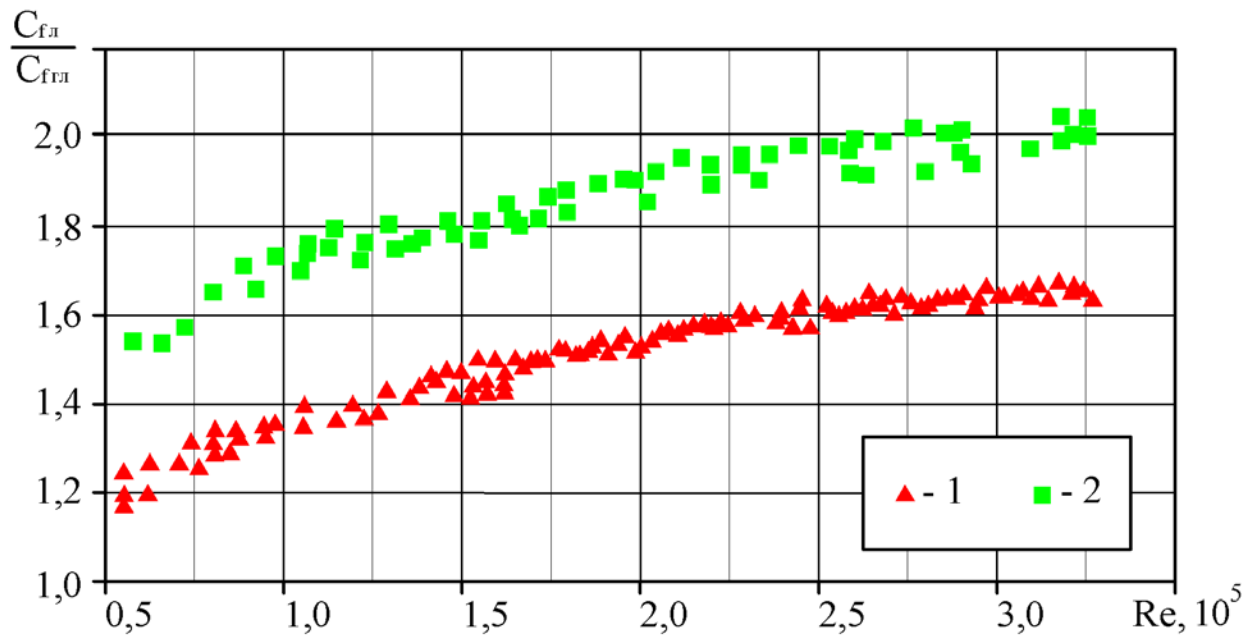
Surface No. 2

Fig. 13. Photos of the test surfaces

The results of the experimental study of hydraulic resistance growth are visible in the Fig. 14 in the appearance of the dependence of the resistance factor of the dimpled surface and the resistance factor of the smooth surface measured in the same experiment on the Reynolds number. As characteristic dimension the hydraulic aperture of the channel was taken.

From the presented results it is evident that the resistance of the surfaces with oval dimples (trenches) is higher than the resistance of the surfaces with hemispherical dimples by 20-35% whereby the machining of the oval dimples requires more sophisticated equipment. But from the papers [27, 28, 29, 30] it is evident that the thermal transfer factor of such surfaces is higher by 90% than that of the surfaces with hemispherical dimples.

From the analysis of the temperature distribution on the surface with cavities it was found out (similar to the studies [10, 32]) that the largest temperature drop on the surface occurs immediately after the cavity at a distance approximately equal to half the diameter of the cavity. In the centre of the cavity the temperature is practically equal to that of a smooth surface.



1 – data for surface No.1 (see Fig. 13) 2 – data for surface No. 2 (see Fig. 13).

Fig. 14 Measurement results of the surfaces with dimples of spheric and oval shape as to smooth surface

## 4 Summary

In the course of the experiment the following data were obtained:

An overview of available experimental and numerical studies of thermal-hydraulic characteristics of heat exchange surfaces with vortex-causing reliefs was made including the studies dedicated to direct resistance measurement (strain-gauge scale). Geometrical parameters of test surface types were defined.

To study the influence of the surface cavities on the heat exchange and the resistance a test bench was upgraded for simultaneous measurement of the heat exchange parameters and the resistance of the test surfaces with parallel arrangement of the latter in the flow. The main advantages of the fixture are the direct force measurement of the resistance and the measurement of the heat transfer (non-stationary method) by means of up-to-date thermal vision equipment. Adjusted were the processing methods of data obtained in the course of the experiment.

Experimental studies of the resistance parameters of surfaces with hemispherical and oval (trench-shape) cavities were conducted. The analysis of the obtained experimental data demonstrated

that surfaces with oval shape of the cavities have larger (by 20-35%) resistance than those with hemispherical cavities.

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