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Experimental investigation of heat transfer and drag on surfaces coated with dimples of different shape

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ABSTRACT

The results of an experimental investigation of the heat transfer and the drag on surfaces coated with dimples are presented. The dimples of six different shapes were considered, namely, spherical, oval, and teardrop dimples, spherical dimples with rounded edges, turned teardrop dimples, and dimples obtained by milling a sphere along a circular arc. The distinctive feature of the study is that the relative drag and heat transfer coefficients were simultaneously recorded during the same run of the experimental setup. The drag coefficients were determined by directly weighing the models under study using a one-component strain gauge balance. The heat transfer coefficients were determined by means of the unsteady heat transfer method using the IR camera. The Reynolds number based on the boundary layer length ranged from $0.2 \cdot 10^6$ to $7 \cdot 10^6$. The two-dimensional fields of the heat transfer coefficient on the dimpled surfaces are presented, together with the data of the flow visualization. The Reynolds number effect on the heat transfer enhancement, the drag increase, and the heat-hydraulic efficiency is determined. The average values of the above-mentioned parameters are presented for all the surfaces considered.

1. Introduction

At present, great interest has been expressed in the problem of increasing the efficiency of the heat transfer equipment, internal cooling of turbine airfoils, and gasdynamic energy separation devices [1]. In many cases the solution of this issue reduces to the solution of the problem of heat transfer enhancement in heat exchanger channels or the passages of cooled turbine airfoils. In the last few decades these are ribs, pins, dimples, and their combinations that are considered as heat transfer intensifiers [2]. For most of these surfaces the heat transfer intensification (by a factor of 2–5 and 2 to 3, for the ribs and the pins, respectively) is accompanied by a considerable increase in the pressure losses (by a factor of up to 70 for the ribs and up to 80 for the pins). Against the background of most of the known heat transfer intensifiers it is the dimples, that is, variously shaped depressions arranged on the surface in staggered or in-line patterns, that stand out. These ensure an increase in the heat transfer by a factor of 1.3–3 at a slight increase in the drag (by a factor of 1–5).

Afanasyev et al. [3] studied flow over ten plates coated with different configurations of spherical dimples. The drag and heat transfer coefficients were determined by means of measuring the velocity and temperature profiles, respectively. An increase in the heat transfer ($St/St_0 = 1.3$–1.4) was accompanied by the absence of a noticeable increase in the drag ($c_x/c_{x0} = 0.94$–1.0). The authors of that study supposed that such heat-hydraulic characteristics of the dimpled surfaces are due to the formation of vortical (tornado-like) flows, similar in their structure with free vortices [4,5]. However, certain researchers believe that the concept of tornado-like vortices is erroneous (the results of LES confirming absence of any pronounced vortex structure are discussed by Kornev et al. [6]).

The most intriguing is the issue of a possible drag reduction on dimpled surfaces to the values smaller than those on a smooth surface. We note that such results were experimentally recorded on surfaces with shallow dimples. However, a closed analysis of such

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Nomenclature

a	- Length of forward region of dimple, mm;	St	- Stanton number, St=$\alpha/(\rho \cdot c_p \cdot V)$;
a_1, a_2	- Major and minor axis of oval dimple, mm;	t	- Student's multiplier;
c	- Specific heat of a solid, J/(kg·K);	t_x, t_y	- Dimple array pitch, mm;
c_f	- Friction coefficient;	T	- Solid temperature, K;
c_p	- Specific heat at a constant pressure, J/(kg·K);	T_S	- Surface temperature, K;
c_x	- Drag coefficient;	T_0	- Flow core temperature, K;
D	- Dimple sphere diameter, mm;	u	- Velocity in dynamic boundary layer, m/s;
D_h	- Hydraulic channel diameter, mm;	V	- Core velocity, m/s;
D_p	- Dimple print diameter, mm;	V_*	- Dynamic velocity, m/s;
e	- Excentricity of tear-drop dimple, mm;	x, y, z	- Streamwise, spanwise and transverse coordinates, m;
F_Σ, $F_{\Delta pl}$, F_x	- Total force; force due to the pressure difference in the floating element gap, force determining the drag; N;	$(U_\alpha)_{0.95}$	- Uncertainty in the measurement of α;
		$(U_{St/St_0})_{0.95}$	- Uncertainty in the measurement of St/St$_0$;
		$(U_{c_x/c_{x0}})_{0.95}$	- Uncertainty in the measurement of c_x/c_{x0};
H	- Channel height, mm;	$(U_{Re_x})_{0.95}$	- Uncertainty in the measurement of Re$_x$;
h	- Dimple depth, mm;	X_u	- The undisturbed boundary layer length, m;
h_1	- Protrision height, mm;	α	- Heat transfer coefficient, W/(m^2·K);
p^*	- Total pressure, Pa;	Δp_{pl}	- Pressure difference in the gaps of the floating element, Pa;
p_s	- Static pressure, Pa;		
Pr	- Prandtl number, Pr=$\mu \cdot c_p/\lambda$;		
r_1, r_2	- Roundness radius of model 5 and 6, mm;	λ	- Thermal conductivity, W/(m·K);
R'	- Radius of circular arc for model 2, mm;	ΔT_0	- Bias limit of T_0 measurement, K;
RAF	- Reynolds analogy factor, RAF=(St/St$_0$)/(c_x/c_{x0});	ΔT_s	- Bias limit of T_s measurement, K;
Re$_L$	- Reynolds number, Re=$V \cdot L \cdot \rho/\mu$, L—length, m;	ΔF_Σ	- Bias limit of F_Σ measurement, N;
S$_1$, S$_2$	- Spanwise and streamwise spacing of the dimple, mm;	$\Delta(\Delta p_{pl})$	- Bias limit of Δp_{pl} measurement, Pa;
		$\Delta(p^*$-$p_s)$	- Bias limit of dynamic head measurement, Pa;
S_{T_0}	- The precision index of T_0 measurement, K;	μ	- Dynamic viscosity, Pa·s;
S_{T_s}	The precision index of Ts measurement, K;	ν	- Kinematic viscosity, m^2/s;
S_{F_Σ}	- The precision index of F_Σ measurement, N;	ρ	- Density, kg/m^3;
$S_{\Delta p}$	- The precision index of Δp_{pl} measurement, Pa;	θ	- Time, s;
$S_{(p^*-p_s)}$	- The precision index of (p^*-p_s) measurement, Pa;	τ_w	- Shear stresses on the wall, Pa;
S_{dimple}, S_{side}, S'	- Dimple print area; area of the surface of the floating element; area of the floating element endface, m^2;	φ, η	- Nondimensional universal boundary layer coordinates;
		St$_0$, c_{x0}	- Heat transfer and drag coefficient for the smooth surface;

surfaces made by Lienhart et al. [7] showed the same increase in the drag both in the experiment ($c_x/c_{x0} = 1.02$) and in direct numerical simulation ($c_x/c_{x0} = 1.04$). At the same time, the up-to-date accurate experimental measurements again report on the possibility of reducing the drag. Thus, Tay et al. [8] note a 3% drag reduction ($c_x/c_{x0} = 0.97$), while Nesselrooij et al. [9] report on a 4% reduction ($c_x/c_{x0} = 0.96$). The authors of the latter paper attribute the results obtained to the interaction between the dimples in a staggered array leading reduction of turbulence-induced friction. One thing is clear: actually a decisive answer to this question cannot be given.

Isaev et al. [10] numerically investigated the effect of the Reynolds number and the spherical dimple depth on the heat transfer and the drag in a slot channel. According to the results presented in that study, the heat-hydraulic efficiency of a dimple diminishes with increase in the dimple depth and the Reynolds number.

Hu et al. [11] present the results of an experimental investigation of the turbulent boundary layer over a dimpled wall. The measured data are compared with the case of a conventional flat wall. The drag coefficient was determined by means of measuring the pressure distribution over the dimple surface. A PIV system was employed in investigating the parameters of the turbulent boundary layer and unsteady vortex structures within a dimple. It was found that the flow separates near the leading edge of the dimple with the formation of unsteady Kelvin—Helmholtz vortices and reattaches to the trailing edge of the dimple with the formation of an intense upward stream.

The published studies [3,6–27] indicate that both the thermal and the hydraulic parameters of dimpled surfaces depend on many different factors, such as the dimple shape, the density of the arrangement over a surface (including the longitudinal and spanwise pitches of the dimples), their relative depth, and others. Moreover, the drag and the heat transfer in channels is considerably influenced by the presence and mutual arrangement of dimples on neighboring surfaces. The investigations of nonspherical dimples [12–26] indicate the possibility of a greater increase in the heat-hydraulic efficiency compared with the case of spherical dimples (see Table 1 and Fig. 1). It should be noted that the results of the recent experimental studies indicate a slight (maximally twofold) increase in the heat transfer enhancement on dimpled surfaces (see Table 1 and Fig. 1).

Ha et al. [12] numerically investigated the heat transfer and the drag on the surfaces coated with dimples with an internal protrusion. It was shown that mounting a protrusion within a dimple reduces the recirculation zone size, which leads to an increase in the heat transfer enhancement compared with a conventional spherical dimple. However, the relative drag and heat transfer coefficients diminish with increase in the protrusion height.

The use of an internal protrusion to provide a greater (compared with the case of a conventional spherical dimple) increase in the heat-hydraulic efficiency was also numerically investigated in the study by Xie et al. [13]. The heat-hydraulic efficiency of the dimples with internal protrusions is greater than that of a conventional spherical dimple. The variously shaped dimples (cylindrical, triangular in different combinations, inverse triangular and inclined cylindrical dimples) were also studied in the study by Ligrani et al. [14].

The study of Chyu et al. [15] is one of the first studies devoted to dimples of complicated shape. There the dimples of spherical and teardrop shapes were compared. Flows in a slot channel with one or two dimpled walls were considered. The relative depth of the