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Heat transfer behavior of flat plate having 45° ellipsoidal dimpled surfaces

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ABSTRACT

Flat surface with ellipsoidal dimple of external flow was investigated in this study.10 types of dimple arrangements and dimple intervals are studied. The stream of air flows over the heated surface with dimples. The velocity of the air stream varies from 1 to 5 m/s. The temperature and velocity of air stream and temperature of dimpled surfaces were measured. The heat transfer of dimpled surfaces was determined and compared with the result of smooth surface. For the staggered arrangement, the results show that the highest heat transfer coefficients for dimpled surfaces are about 15.8% better than smooth surface as dimple pitch of $S_T/D_{minor}=3.125$ and $S_L/D_{minor}=1.875$ yield the highest heat transfer coefficient values. And for the inline arrangement, the results show that the heat transfer coefficients for dimples surfaces are about 21.7% better than smooth surface as dimple pitch of $S_T/D_{minor}=1.875$ and $S_L/D_{minor}=1.875$ yield the highest heat transfer coefficient values.

1. Introduction

There are many thermal applications where the fluid-to-gas heat exchanger is a crucial element. The conventional enhanced heat transfer approaches such as fins, baffles, turbulizers etc., always increase either the heat transfer surface area or turbulence or both. These approaches are effective in increasing heat transfer rates; however, this also results in the significant pressure drops. The dimpled surface is a special method for improving the heat transfer rates without the significant pressure drop. Normally, the dimple generates the vortex flow within its hole and the augmentation of heat transfer is obtained [1].

Recently, dimples or concave surfaces have been in focus extensively. In the early investigations, Afansayev et al. [2] investigated the overall heat transfer and pressure drop for turbulent flow flat surface with staggered array of spherical dimples. Significant (30–40%) increases in heat transfer without appreciable pressure losses are reported.

Chyu et al. 1997 [3] studied the heat transfer coefficient distributions of air flow in the channel over flat surface indent with staggered arrays of two different shaped dimple. Their result shows that the local heat transfer coefficients are significantly higher than values in the channels with smooth surfaces. Enhancements are of about 2.5 times of smooth surface values, and pressure losses are about half the values produced by conventional rib turbulators.

Mahmood et al. [4] describe the flow structure above the dimpled flat surface. Flow visualizations showed vortical fluid as vortex pairs shed from the dimples. Their results showed the existence of a low heat transfer region in the upstream half

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Nomenclature

A	Area (m^2)
D	Dimple diameter (mm)
D_{minor}	Dimple diameter on monor axis (mm)
Nu	Average Nusselt number
Nu_x	Local Nusselt number
Nu_0	Baseline average Nusselt number of flat plate without dimple
Pr	Prandtl number
Re_x	Reynolds number base surface length (include dimples surface)
h	Average heat transfer coefficient (W/m^2 K)
h_0	Average heat transfer coefficient of flat plate without dimple (W/m^2 K)
h_x	Local heat transfer coefficient (W/m^2 K)
S_L	Streamwise pitch (mm)
S_T	Spanwise pitch (mm)
T	Temperature (°C)
V	Velocity (m/s)
x	Spanwise coordinate
y	Streamwise coordinate

of the dimple cavity followed by a high heat transfer region in the downstream half. Additional regions of high heat transfer were identified at the downstream rim of the dimple and on the flat surface downstream of each dimple.

Ligrani et al. [5] reported that as the H/D (channel height to dimple diameter) increases the secondary flow structures and flow mixing intensified decreased. Nevertheless, Moon et al. [6] obtained almost a constant heat augmentation ratio of 2.1 for a dimpled passage with H/D from 0.37 to 1.49.

Burgess et al. [7] reported that both the Nusselt number and the friction augmentation increased as the dimple depth increased. These are attributed to (i) increases in the strengths and intensity of vortices and associated secondary flows ejected from the dimples, as well as (ii) increases in the magnitudes of three-dimensional turbulence production and turbulence transport. The effects of these phenomena are especially apparent in the local Nusselt number just inside, and on the downstream edges of the dimples.

Wang et al. [8] investigated a novel enhanced heat transfer tube with ellipsoidal dimples. Their computed results indicated that the Nusselt number for ellipsoidal dimpled tube and spherical dimpled tube are 38.6–175.1% and 34.1–158% higher than that for the smooth tube, respectively. The friction factors of dimpled tube increase by 26.9–75% and 32.9–92% for ellipsoidal and spherical dimples compared with the smooth tube respectively.

Katkhaw et al. [9] studied the heat transfer of air flow over dimpled flat surface with the 14 types of dimpled arrangement. Their results show that heat transfer coefficients for dimpled surfaces are about 26% better than smooth surface for staggered arrangement. And for the inline arrangement, the results show that heat transfer coefficients for dimples surfaces are about 25% better than smooth surface.

This study proposes to employ the ellipsoidal dimple on the flat surface to observe the thermal characteristic of air flow. In addition, the effect of dimple arrangements, and dimple pitch will be included in this work. The results of this research aim to serve the heat exchanger application.

2. Experimental setup and procedure

The schematic diagram of the experimental setup for heat transfer measurements is shown in Fig. 1. The fluid used in the apparatus is air at room temperature, which is generated by an air blower. The stream of air flows through the test section and flows over test surface. In this experiment, the velocity of air stream was controlled by the frequency inverter with the controllable range of 1–5 m/s. The velocity of air stream was measured by a hot wire anemometer with ± 0.2 m/s accuracy. The surface temperature of flat plate was measured by infrared imaging camera with ± 2 °C accuracy, where the measuring instrument was also calibrated to surface temperature with T-type thermocouple. The inlet temperature of air stream was measured by a T-type thermocouple with ± 0.1 °C accuracy.

The test kit comprises test plate and plate heater. Test plates are made from 1.5 mm thickness acrylic which is depressed in any configurations of dimple. The iron powder was filled beneath the tested plate to avoid the space between the plate and electric heater. The exterior of the test kit was insulated with three layers of 2.5 cm-thickness glass wool insulation to minimize heat loss. The electric heater supplies a constant heat flux, where the power to the heater was controlled and regulated by a variac power supply. T-type thermocouples are located between each layer to determine conduction losses.

Figs. 2 and 3 illustrate the geometric details of the tested surface indent with arrays of ellipsoidal dimple. Dimples were employed in an inline and staggered array. Table 1 lists the details of the test case.

3. Data reduction

During the experiment, inlet air stream velocity, temperature, the tested surface temperature and power supply were measured to determine the local heat transfer coefficient (h_x), which is calculated from

$$h_x = q_s''/(T_s - T_\infty) \tag{1}$$