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Heat transfer and flow structure in turbulent channel flow over protrusions

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ABSTRACT

In this study, heat transfer characteristics and flow structures over protrusions in a turbulent channel flow were systematically investigated by DES numerically. Densely arranged protrusions with different depth ratios (h/D) are considered for Reynolds number Re_{2H} (based on bulk velocity and full channel height) between 3000 and 6000 while Prandtl number Pr is fixed at 0.7. It is found that larger height ratio induces higher friction factor and heat transfer. However the performance factor, on the other hand, first increases then reaches its asymptotic limit and even decreases with increasing height ratio. It is also observed that the highest skin friction, form drag and localized Nusselt number are found at the upstream portion of protrusion. Additionally, the distributions of friction factors and Nusselt number exhibit symmetrical features for the low protrusion configuration and asymmetric characteristics for the high protrusion arrangement. The asymmetric distribution of localized Nusselt number at large height ratio ($h/D \geqslant 15\%$) is found to be closely linked to the asymmetric vortex shedding from the back ridge of protrusions.

1. Introduction

The enhancement of heat transfer efficiency is important for a variety of practical applications such as in the internal cooling passages of turbine blades, cooling of electronic components, heat exchangers, cooling along the micro-fluidic passages and even biomedical devices in continuous operation. In the past years, dimpled surface has attracted wide attentions due to its higher transfer enhancement with less pressure loss penalty than the other conventional methods [1,2]. It has been reported that the channel height (H/D) and dimple depth ratio (h/D) have the largest impact on the thermal-hydrodynamic characteristics of dimpled surfaces among the different geometry parameters of dimples [3–5]. Generally, higher depth ratio (h/D) induces higher friction factor and heat transfer rate. However, heat transfer rate may decrease when depth ratio (h/D) increases to extremely high level [6].

As an extension of dimpled surface, channel with dimples and protrusions on opposite sides was experimentally found to have higher friction and Nusselt number than channel with dimples on both walls [7,8]. [9] also numerically investigated the heat transfer in a channel with dimples and protrusions on opposite sides, and found that larger dimple depth or protrusion height (h) induced higher heat transfer rate. However, the most important parameter—depth/height ratio (h/D) was kept as a constant in their study, and only two different depths/heights (h) were examined. On the other hand, Hwang et al. [10] experimentally studied heat transfer in a channel with dimple or protrusions on single or both walls, and heat transfer and friction augmentation in channel with protrusion higher than those in dimple case was reported. It was also found that the highest local heat transfer is located at the upstream region of protrusion, while the lowest is in the regions directly beneath and immediately downstream of protrusion.

In summary, protrusion can lead to higher heat transfer and friction than dimples. However, in the studies of channel with dimples and protrusions on opposite sides [7–9] both dimples and protrusion affect the flow and heat transfer in the channel, so the respective influences are difficult to be distinguished. In the research which focused on channel with only protrusions [10], flow structures have not been studied. In addition, the effects of the most important parameter—depth/height ratio (h/D)—is still unclear for protrusion. All these provide us with the motivation to undertake an in-depth study of the flow and heat transfer features in a channel with single protrusion wall, and to examine closely the thermal-hydrodynamic performance including Nussselt number, friction and performance factor, and the effect of changing the height ratio. Additionally, the distribution of friction factors and Nusselt number are studied with the objective of providing the connectivity, if any, between them and the flow/vortex patterns over the protrusion.

2. Methodology

In this study, Spalart–Allmaras (S–A) model [11] based Detached Eddy Simulation (DES) method were implemented to simulate the turbulent channel flow with massive separations.

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The grid resolution of DES is not as demanding as a pure Large Eddy Simulation (LES) and Direct Numerical Simulation (DNS) approaches, thereby considerably cutting down the cost of the computation. By taking advantage of the DES approach over other turbulent models, an in house finite-volume-based parallel DES code modified from the DNS code of [12] is applied in the present work. More details of DES model implemented in this study can be found in the authors' previous work [13].

2.1. Governing equations

In this study, fluid flows inside a channel with length L, width W and height $2H$ in the x, z and y direction respectively (Fig. 1). For all the cases discussed here, only the lower wall consists of protrusions, while the upper wall is always flat. The protrusion's print diameter is a constant $D = 5H$, and its height h varies from $5\%D$ to $25\%D$.

The non-dimensional governing equations of incompressible flow are as follows:

$$\frac{\partial u_i}{\partial x_i} = 0, \tag{1}$$

$$\frac{\partial u_i}{\partial t} + \frac{\partial (u_i u_j)}{\partial x_j} = -\frac{\partial p'}{\partial x_i} + \frac{1}{Re_\tau}\frac{\partial^2 u_i}{\partial x_j \partial x_j} + \beta\delta_{i1}, \tag{2}$$

$$\frac{\partial T'}{\partial t} + \frac{\partial (T' u_j)}{\partial x_j} - \gamma\delta_{i1}u_i = \frac{1}{Re_\tau Pr}\frac{\partial^2 T'}{\partial x_j \partial x_j}, \tag{3}$$

where δ_{i1} is the Kronecker delta, while β and γ are the non-dimensional mean pressure and temperature gradients.

For purpose of nondimensionalization, the reference length scale employed is the half channel height H^* and the reference velocity is $u_\tau^* = \sqrt{\beta^* H^*/\rho^*}$, where ρ^* is the fluid density and β^* is the mean pressure gradient in the stream-wise direction. Therefore, the non-dimensional mean pressure gradient imposed in the stream-wise direction of channel was constant $\beta = 1$, and the friction Reynolds number based on half channel height defined as $Re_\tau = u_\tau^* H^*/\nu^*$ was fixed at 180 in this study. Herein, the parameters with superscript (*) represent dimensional quantities, while the corresponding parameters without superscript represent the non-dimensional quantities. Periodic boundary conditions for the velocity $\vec{u}$, pressure and temperature fluctuations (p' and T') are set at stream-wise and span-wise directions. On the other hand, the no-slip and constant heat flux ($q = \nabla T' = 1$) are taken as the boundary conditions on the top and bottom walls.

2.2. Calculation of the thermo-aerodynamic performance

To compare the thermal performances over different periodically dimple-protrusion patterned surfaces, it is important to evaluate the friction coefficient C_f and Nusselt number Nu. That is,

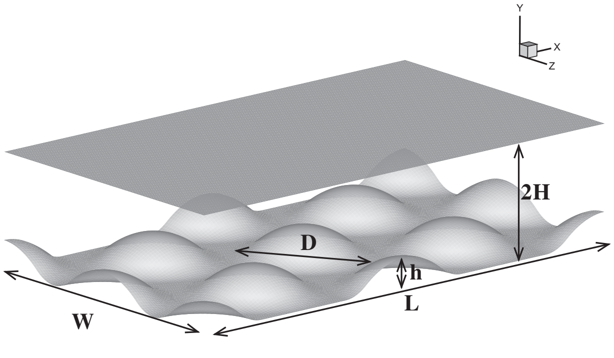

Fig. 1. Channel with protrusions.

$$C_f = \frac{\tau^*_{w\,eqv}}{\frac{1}{2}\rho^* U_b^{*2}} = \frac{\beta D_h}{2U_b^2}, \tag{4}$$

$$Nu = \frac{q^*(2H^*)}{k_f^*\left|T^* - T^*_{ref}\right|} = \frac{2}{\left|T' - T'_{ref}\right|}, \tag{5}$$

where $\tau_{w\,eqv}, D_h, U_b, q, k_f$ and T'_{ref} respectively represent the equivalent drag per unit projected area, hydraulic diameter, mean bulk velocity, heat flux at wall, thermal conductivity of the fluid and the mean-mixed temperature. The non-dimensional mean bulk velocity U_b is

$$U_b = \frac{\iint u_x dA_x}{\iint dA_x} \tag{6}$$

and T'_{ref} is mean-mixed temperature defined as:

$$T'_{ref} = \frac{\iiint |u_x| T' dA_x dx}{\iiint |u_x| dA_x dx}. \tag{7}$$

The surface-average Nusselt number is calculated by averaging over the surface S like

$$Nu_{avg} = \frac{\iint Nu dS}{\iint dS}.$$

Furthermore, the local skin friction drag and form drag per unit projected area on channel wall in the X-Z plane can be defined as:

$$Sm = \frac{\left[\left(\tau_{xx}\vec{i} + \tau_{xy}\vec{j} + \tau_{xz}\vec{k}\right)\cdot\vec{n}\right] dA_w}{\frac{1}{2}U_b^2 dA_{pro}}, \tag{8a}$$

$$Fm = -\frac{\left[(p' - \beta x)\vec{i}\cdot\vec{n}\right] dA_w}{\frac{1}{2}U_b^2 dA_{pro}}, \tag{8b}$$

where A_{pro} is the total projected area of channel wall in the X-Z plane.

Empirical friction coefficient C_f^0 and Nusselt number of a smooth flat channel Nu^0 are employed as reference to validate the numerical results, and are obtained using the Petukhov and Gielinski correlations [14] respectively:

$$C_f^0 = [1.58\ln(Re_{2H}) - 2.185]^{-2}, 1500 \leqslant Re_{2H} \leqslant 2.5\times 10^6, \tag{9}$$

$$Nu^0 = \frac{\left(C_f^0/2\right)(Re_{2H} - 500)Pr}{1 + 12.7(C_{f0}/2)^{1/2}\left(Pr^{2/3} - 1\right)},\quad 1500 \leqslant Re_{2H} \leqslant 2.5\times 10^6. \tag{10}$$

Note that the original Petukhov and Gnielinski correlations are rewritten here in terms of Re_{2H} rather than Re_{Dh}, where $Re_{Dh} = 2Re_{2H}$ for smooth parallel plates with infinite width ($2H$ is full channel height, and H is half channel height).

In order to evaluate the quantitative thermo-aerodynamic performance for the different heat transfer surface geometries, the performance factors, *area/volume goodness* ratios, are respectively defined as: [15]

$$Ga/Ga_0 = \frac{Nu/Nu_0}{C_f/C_{f0}}, \tag{11}$$

$$Gv/Gv_0 = \frac{Nu/Nu_0}{(C_f/C_{f0})^{1/3}}, \tag{12}$$

where Nu_0 and C_{f0} are the Nusselt number and friction factor of the smooth wall case. For a given pumping power and constant fluid thermo-mechanical properties, a higher *area/volume goodness factor* yields a comparatively smaller heat transfer surface area/volume, resulting in a smaller weight of heat exchanger matrix under a fixed plate thickness and material of heat transfer surface.

One may note that the assembled code has been extensively tested including against DNS for a flat channel flow and others.