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Double-sided cooling of heated slab: Conjugate heat transfer DNS

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A B S T R A C T

It is well known that turbulent temperature fluctuations penetrate into the heated wall that is being cooled with a turbulent flow. The present work represents a theoretical analysis of the heated slab that is being cooled with turbulent flow from both sides. Results of the direct numerical simulations predict penetration of the turbulent temperature fluctuations into the solid wall. For a sufficiently thick slab, temperature fluctuations from both sides of the slab do not interfere. As the slab gets thinner, fluctuations from both sides interfere and tend to a finite value as the slab thickness limits toward zero. Due to the non-coherent turbulent flows on each side of the slab, thermal fluctuations in the zero-thickness slab are actually lower than in the case of the zero-thickness wall, which is heated by the same turbulent flow from a single side and is isolated on the other side. Spectral numerical scheme was used for Direct Numerical Simulation of fully developed channel flow that is cooling the idealized slab heated with constant volumetric heat source. Implemented boundary conditions for liquid and solid energy equations correspond to the geometry that can be found in some experimental nuclear reactors with fuel in the form of parallel slabs. Periodicity of streamwise and spanwise directions was assumed for velocity and passive scalar temperature field. For temperature, periodicity of the computational domain was assumed also in the wall-normal direction. Most of the simulations were performed at constant friction Reynolds number 180 and Prandtl number 1 with various geometrical and material properties of the heated slab. Some additional analyses were performed also at friction Reynolds number 395 and Prandtl numbers 0.1 and 10.

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1. Introduction

Soon after the first direct numerical simulation (DNS) of the turbulent flow in the channel has been performed by Kim et al. in 1987 [1], the first DNSs of turbulent heat transfer in the channel emerged. Kim and Moin in 1989 [2] specified the heat source as a volumetric heating. Kasagi et al. in 1992 [3] considered the flow between the heated walls, which became a standard approach for DNS in turbulent heat transfer. In the years that followed, DNSs were performed at larger Reynolds numbers: DNSs at $Re_\tau = 180$, 395 and 590 are compared in [7], DNSs with up to $Re_\tau = 2000$ are reported in [4]. Review of turbulent channel flow DNSs has been made by Moin and Mahesh in 1999 [4] and by Jimenez and Moser in 2007 [5]. Various Prandtl and Reynolds number DNS studies are described in [8–11]. Useful DNS databases containing results obtained at various Reynolds and Prandtl numbers are being maintained by Kasagi et al. [3] and Kawamura et al. [8]. Review of the turbulent heat transfer DNS was given by Kasagi and Iida in 1999 [6].

In the past decade DNS of turbulent heat transfer was not limited only to fully developed channel flow. Today, heat transfer DNSs can be found in analyses of spatially developing boundary layers [12,17,16] and more complicated geometries, like annular pipes [14]. Attempts are being made to use DNS or at least the so-called quasi-DNS to simulate complex geometries like T-junctions [15] and even pebble bed reactors [13].

Details of near-wall heat transfer are not clearly specified in the papers mentioned above. Near-wall behavior of turbulent temperature field depends on the properties of the heated wall. Heated wall material properties determine whether the temperature fluctuations penetrate into the wall or they remain limited to the fluid. DNS of boundary layer with heat transfer [17] and DNS of turbulent channel flow [19] have clearly shown distinction between the heated wall with imposed constant dimensionless temperature boundary condition and constant dimensionless heat flux boundary condition: The former does not allow penetration of the turbulent temperature fluctuations into the wall and is thus called “non-fluctuating temperature boundary condition”, while the later exhibits non-zero temperature fluctuations at the fluid-wall contact and is thus referred to as “fluctuating temperature boundary condition” [18,20].

The most precise analyses of turbulent temperature fluctuations that penetrate into the wall require conjugate heat transfer

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Nomenclature

Bi	$\mathrm{Nu}(\lambda_f/\lambda_w)(d/2h)$ Biot number of the slab
c_p	specific heat at constant temperature
d	wall thickness
G	α_f/α_w ratio of thermal diffusivities
h	1 channel half-width
H	λ_f/λ_w ratio of thermal conductivities
K	$\sqrt{\rho_f c_{pf}\lambda_f/\rho_w c_{pw}\lambda_w}$ thermal activity ratio
L_1,L_3	streamwise and spanwise length of turbulent box
Nu	$(2h/\lambda_f)(q_w/\theta_{BULK})$ Nusselt number in channel flow
p	pressure
Pr	Prandtl number
q_w	wall-to-fluid heat flux
Re_τ	Friction Reynolds number
t	time
T_τ	$q_w/(u_\tau\rho_f c_{pf})$ friction temperature
u,v,w	velocity components in x,y and z directions
u_τ	$\sqrt{\tau_w/\rho}$ friction velocity
u_B	bulk mean velocity
x	streamwise distance
y	distance from the wall
z	spanwise distance
α	$\lambda/\rho c_p$ thermal diffusivity
θ	$(T_w - T)/T_\tau$ dimensionless temperature (difference)
λ	thermal conductivity
ν	kinematic viscosity
ρ	density
$\vec{1}_x$	unit vector in x direction (1,0,0)

Subscripts

$(\,)_w$	solid wall
$(\,)_f$	fluid

Superscript

$(\,)^+$	normalized by u_τ, T_τ, ν

models, i.e. a coupled problem of turbulent heat transfer and unsteady wall-side heat conduction has to be solved. Such conjugate heat transfer semi-analytical studies are actually even older than the first DNS studies. Polyakov considered the problem in 1974 [21] and Khabakhpasheva [22] performed experiments, which demonstrated variable penetration of the turbulent temperature fluctuations into the heated walls made of different materials and of various thickness. Similar study is described in [23]. Detailed study of conjugate heat transfer by Kasagi et al. [24] was performed with unsteady two-dimensional model of near-wall turbulence and simultaneous solution of fluid energy equation and equation for solid heat conduction. Kasagi et al. [24] has shown that the wall temperature fluctuations are usually negligible when the walls are being cooled with air, but might be measurable when thin metal walls are being cooled with water. The same analysis was repeated with improved accuracy of DNS by Tiselj et al. [25] for Prandtl numbers 0.7 and 7. Recently a similar analysis has been repeated by Tiselj and Cizelj in [11] for low Prandtl number Pr = 0.01, which roughly corresponds to the liquid sodium. Their study took into account two dimensionless parameters: thermal activity ratio $K = \sqrt{\rho_f c_{pf}\lambda_f/\rho_w c_{pw}\lambda_w}$ and ratio of thermal diffusivities $G = \alpha_f/\alpha_w$, as proposed by Brillant et al. [27,28]. Earlier studies of Kasagi et al. [24] and Tiselj et al. [25] used a single parameter K with a fixed value of $G = 1$. Some typical values of K and G parameters and fluid Prandtl numbers are collected in Table 1.

Present work couples DNS of turbulent heat transfer in the fully developed channel flow with unsteady heat conduction in the wall. Most of the simulations were performed at friction Reynolds number $\mathrm{Re}_\tau = 180$ and Prandtl number 1. Some additional runs were performed to test the influence of the Reynolds and Prandtl number. The original contribution of the present work is that the simulations are not focused only on single-side cooling of the heated wall, but on a more general case of the heated slab cooled with turbulent flow from both sides. Our first attempts in this direction have been performed with a numerical scheme that treated the solid wall heat conduction with a mixture of spectral and finite difference [26]. This scheme turned out to be of insufficient accuracy for very thin walls and was upgraded into fully spectral scheme for solid heat conduction. As described below, spectral scheme performed remarkably well under all circumstances.

Table 1
Parameters of typical fluid–solid systems: thermal activity ratio K, ratio of thermal diffusivities G and liquid Prandtl numbers.

	K	G	Pr
Water/iron	0.03	0.01	7
Water/aluminum	0.005	0.002	7
Air/iron	$4*10^{-7}$	2	0.7
Air/aluminum	$6*10^{-8}$	0.2	0.7
Liquid sodium/steel	1	10	0.005

2. Mathematical model

Equations describing passive scalar approximation of turbulent heat transfer in the channel can be found in most of the papers dealing with DNS that were mentioned in the introduction section. Dimensionless equations in fluid, normalized with channel half width h, friction velocity u_τ, kinematic viscosity ν, and friction temperature $T_\tau = q_w/(u_\tau\rho_f c_{pf})$ as given by Kasagi et al. [3] are:

$$\nabla\cdot\vec{u}^+ = 0, \tag{1}$$

$$\frac{\partial\vec{u}^+}{\partial t} = -\nabla\cdot(\vec{u}^+\vec{u}^+) + \frac{1}{\mathrm{Re}_\tau}\nabla^2\vec{u}^+ - \nabla p + \vec{1}_x, \tag{2}$$

$$\frac{\partial\theta^+}{\partial t} = -\nabla\cdot(\vec{u}^+\theta^+) + \frac{1}{\mathrm{Re}_\tau\cdot\mathrm{Pr}}\nabla^2\theta^+ + \frac{u_x^+}{u_B^+}. \tag{3}$$

Terms $\vec{1}_x$ (unit vector in streamwise direction) and u_x^+/u_B^+ appear in the Eqs. (2) and (3) due to the numerical scheme that requires periodic boundary conditions in the streamwise and the spanwise directions. They represent constant pressure and temperature streamwise gradients that are subtracted from the dimensionless pressure and temperature.

Dimensionless equation for heat conduction in the wall of thickness d with internal heating was used by Kasagi et al. [24] and improved by Brillant et al. [27,28]:

$$\frac{\partial\theta^+}{\partial t} = \frac{1}{G\cdot\mathrm{Re}_\tau\cdot\mathrm{Pr}}\nabla^2\theta^+ - \frac{K}{d\sqrt{G}}, \tag{4}$$

where the $-K/(d\sqrt{G})$ term represents dimensionless internal heat source with thermal activity ratio $K = \sqrt{\rho_f c_{pf}\lambda_f/\rho_w c_{pw}\lambda_w}$ and ratio

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