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► **To cite this version:**

Cédric Flageul, Sofiane Benhamadouche, Eric Lamballais, Dominique Laurence. DNS of Channel flow with conjugate heat transfer Budgets of turbulent heat fluxes and temperature variance. Engineering Turbulence Modelling and Measurements (10th ETMM), ERCOFTAC, Sep 2014, Marbella, Spain. hal-01162346

**HAL Id: hal-01162346**

**<https://hal.science/hal-01162346>**

Submitted on 11 Jun 2015

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# DNS OF CHANNEL FLOW WITH CONJUGATE HEAT TRANSFER - BUDGETS OF TURBULENT HEAT FLUXES AND TEMPERATURE VARIANCE

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## 1 Introduction

Conjugate heat transfer simulations are needed in industrial applications where fluctuating thermal stresses are a concern, in case of a severe emergency cooling or long ageing materials. High Reynolds RANS and LES simulations rely on wall-modeling as the viscous sub-layer is not resolved. DNS is a valuable tool for understanding the flow physics of such complex phenomena and to provide fine data in order to improve RANS and LES approaches.

Only few DNS dealt with conjugate heat transfer issues. To the authors' knowledge, Tiselj et al. (2001a) were among the first to investigate this more realistic situation in a channel flow using pseudo-spectral methods. Kang et al. (2009) performed Quasi-DNS with conjugate heat transfer of a heated cylinder in a channel using immersed boundaries and fully implicit LES solver based on unstructured collocated mesh using finite volume approach.

The present work provides budgets of turbulent heat fluxes and temperature variance for a channel flow with different thermal boundary conditions: an imposed temperature, an imposed heat flux and conjugate heat transfer.

## 2 Numerical approach

Our simulation is based on the open-source software Incompact3d developed at Université de Poitiers and Imperial College London (<http://code.google.com/p/incompact3d/>) by Laizet et al. (2009, 2011). Sixth-order compact finite difference schemes are used on a Cartesian grid. The pressure is computed on a staggered grid using a spectral approach while velocity's components and temperature are collocated.

The incompressible Navier-Stokes and energy equations of a fluid with constant properties (1) and (2) are solved using a projection method (eqs (3)

and (4)). The convective term in the momentum equation is computed using the skew-symmetric form.

$$\partial_t u_i = 0 \quad \partial_i u_i + \partial_j (u_i u_j) = -\partial_i p + \frac{1}{\text{Re}} \partial_{jj} u_i \quad (1)$$

$$\partial_t T + \partial_j (T u_j) = \frac{1}{\text{Re Pr}} \partial_{jj}^2 T \quad (2)$$

$$\frac{u_i^* - u_i^n}{\Delta t} = \frac{(\partial_{yy} u_i^* + \partial_{yy} u_i^n)}{2 \text{Re}} + F_i(u^n, u^{n-1}, p^n) \quad (3)$$

$$\frac{u_i^{n+1} - u_i^*}{\Delta t} = -\partial_i \delta p^{n+1} \quad (4)$$

The convective and the wall-parallel viscous terms are integrated through a 2<sup>nd</sup> order Adams-Bashforth time advancing scheme and the wall-normal viscous term through a Crank-Nicolson one. The numerical approach is exactly the same for the fluid temperature. While performing conjugate heat transfer, the diffusion equation for the temperature in the solid is given by eq.(5):

$$\frac{T_s^{n+1} - T_s^n}{\Delta t} = \frac{1}{G \text{Re Pr}} \partial_{yy}^2 T_s^{n+1} + F(T_s^n, T_s^{n-1}) \quad (5)$$

The thermal diffusivity ratio G is set to unity for the present computations. The solid domain is on top ( $y \geq L_y$ ) and on bottom ( $y \leq 0$ ) of the fluid domain. The thermal solver for the solid domain uses *finite differences* for the wall-parallel diffusion and a *spectral* method for the wall-normal one. Time-advancement is similar to the one used for moment equation except the wall-normal diffusion that is fully implicit.

The coupling between both thermal solvers is performed as follow. First, the fluid temperature is subjected to a Dirichlet condition:  $T^{n+1} = T^n + T_s^n / 2$ . Then, the solid temperature is subjected to a Neumann condition:  $\partial_n T_s^{n+1} = \alpha \partial_n T^{n+1}$ . The ratio of

thermal conductivity  $\alpha$  is set to unity for our preliminary computations. This coupling strategy is first order in time: the resulting temperature field is slightly discontinuous at the interface while the heat flux is continuous through the Neumann condition approximation.

### 3 Results

A channel flow with a Reynolds number based on the bulk velocity of 2280 is simulated with a Prandtl number of 0.71. The reference simulation are from Kasagi et al. (1992), which uses a Dirichlet boundary condition for the temperature and from Tiselj et al. (2001a) which uses a Neumann boundary condition for the temperature. Table 1 gives a comparison for the main parameters between our simulation and Kasagi et al. (1992). The flow rate is imposed with a source term in the momentum equation. The source term in the energy equation is similar to the one used by Kasagi et al. (1992). The simulation with a temperature imposed at the boundary is called isothermal. The one with a heat flux imposed is called isoflux. For the conjugate heat transfer case, the solid domain size is [12.8; 1; 4.26], located on top and bottom of the fluid domain. The external faces of the solid domain are subjected to a constant heat flux, equal to the one imposed in the isoflux case.

	Present	Ref
Domain	[12.8; 2; 4.26]	[ $5\pi$ ; 2; $2\pi$ ]
Grid	[256;193;256]	[128;97;128]
$Re_\tau$	148.8	150
$\Delta y^+$	[0.49; 4.8]	[0.08; 4.9]
$[\Delta x^+, \Delta z^+]$	[7.4; 2.5]	[18.4; 7.36]
$\Delta t^+$	0.02	0.12
Duration	16000	2100

Table 1: Simulation parameters. Ref: Kasagi et al. (1992)

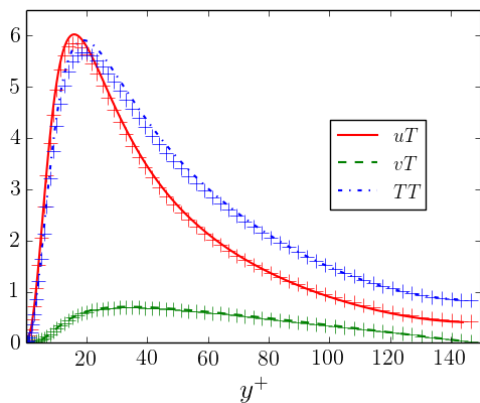


Figure 1: Isothermal b.c., turbulent heat fluxes and temperature variance. Line: present. Symbol: Kasagi et al. (1992).

As shown in Figures 1 and 2, our compact finite-difference simulation compares well with the

pseudo-spectral results of Kasagi et al. (1992), even for high order statistics such as the budgets of the turbulent heat fluxes.

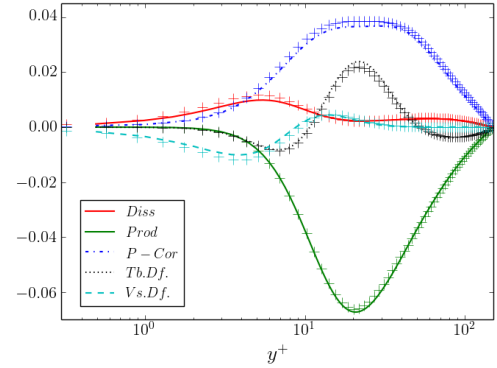


Figure 2: Isothermal b.c., budget of wall-normal turbulent heat flux. Line: present. Symbol: Kasagi et al. (1992).

In figure 3, the turbulent heat fluxes and the temperature variance are in good agreement with the results of Tiselj et al. (2001a).

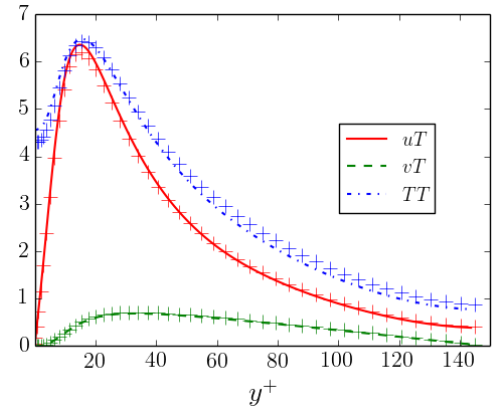


Figure 3: Isoflux b.c., turbulent heat fluxes and temperature variance. Line: present. Symbol: Tiselj et al. (2001a).

Budgets for the isoflux case are available for a flume configuration (Tiselj et al. (2001b)) and for concentric annular pipe flow (Chung et al. (2003)). However, to the authors' knowledge, no budgets are available in the literature when the heat flux is imposed or in case of conjugate heat transfer for the channel flow configuration. Therefore, budgets for our isoflux and conjugate simulations are compared with their isothermal counterpart.

In figure 4, the molecular diffusion and the dissipation behave differently at the wall where they have lower amplitude in the isoflux case compared with the isothermal case. The behavior is similar in the budgets of the temperature variance. Asymptotic analysis of the temperature variance ( $y \rightarrow 0$ ) leads to a non-zero molecular diffusion associated with the temperature variance at the wall for isothermal, isoflux and conjugate cases.

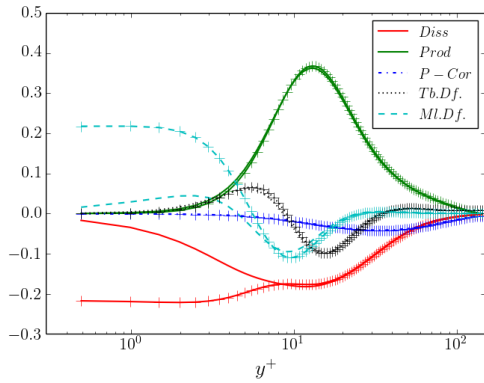


Figure 4: Budget of axial turbulent heat flux.  
Line: isoflux b.c.. Line+Symbol: isothermal b.c.

In figure 5, the situation is similar for conjugate heat transfer: the molecular diffusion and the dissipation have lower amplitude at the wall when compared with the isothermal case. In fact, for most of the statistics, a 3-way comparison puts conjugate heat transfer in-between isothermal and isoflux results, as expected.

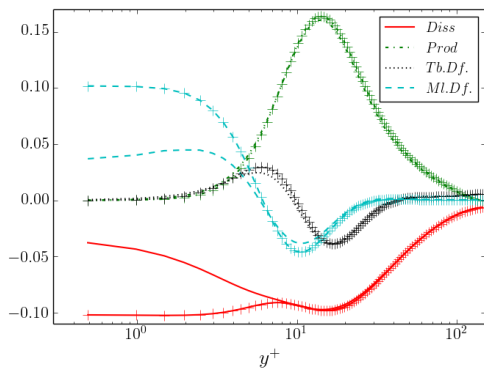


Figure 5: Budget of temperature variance.  
Line: conjugate. Line+Symbol: isothermal b.c.

#### 4 Conclusion

In figure 6, the temperature snapshot in logarithmic scaling shows that the solid conduction damps small scale fluctuations but large scale fluctuations penetrate the solid all the ways to the external face. The thermal fluctuation at the outer wall is close to the fundamental frequency in the spanwise direction.

This shows that the periodicity and the limited length of the domain in the streamwise direction artificially enhance the coherence of deeply penetrating streaky hot spots.

The objective behind the present DNS is to build a reference database focused on temperature-related statistics including conjugate heat transfer. DNS results reported here are currently available on request

and will be made publicly available on the user group of the Incompact3d website.

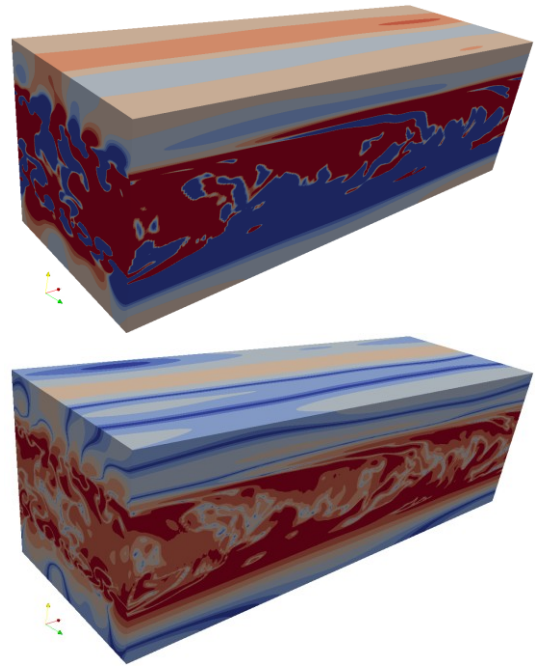


Figure 6: Instantaneous fluctuating temperature field in log scaling. Top: hot and cold spots. Bottom: absolute value.

#### Acknowledgements

The authors thank Prof. I. Tiselj for providing results and assistance. We also thank the French National Research Agency and EDF R&D for funding the study and providing computational time on Zumbrotta supercomputer (IBM – BlueGene/Q).

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