2nd Spanish Fluid Mechanics Conference
Barcelona (2 - 5 July 2023)

Turbulent Statistics and Coherent Structures in an Asymmetrically Heated Channel Flow

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A compressible direct numerical simulation (DNS) has been conducted to analyze the behavior of a fully developed turbulent channel flow. The problem has been set at mean friction Reynolds number $Re_{\tau m} = 400$ submitted to a high-temperature gradient between the two walls ($T_{hot}/T_{cold} = 2$).

1 Introduction

Wall bounded flows with large temperature gradients are present in many engineering applications. In recent years, fully-developed channels have been used to study the coupling effect of the temperature gradient and the turbulence. However, most of the studies of high-temperature gradient channels have been mainly limited to incompressible flows that transport the temperature as a passive scalar, or to low-speed flow limited to low mean friction Reynolds numbers (see for instance Nicoud (1998)).

Toutant and Bataille (2013) performed direct numerical simulations (DNS) with the same configuration as the present study, i.e., at mean friction Reynolds number $Re_{\tau m} = 400$ and temperature ratio $T_{hot}/T_{cold} = 2$. The mean Reynolds number is computed as the mean of the friction Reynolds number at each wall, $Re_\tau = \rho_w \delta u_\tau / \mu_w$, where $u_\tau = \sqrt{\tau_w/\rho_w}$ is the friction velocity, δ is the channel half-height, ρ_w and μ_w are the density and the dynamic viscosity, respectively. They studied the statistics of the flow with different scalings for the velocity and temperature profiles. However, the large-scale coherent structures were not discussed in detail.

The main goal of this work is to study the behavior of an asymmetrically heated channel flow and to analyze the large-scale coherence structures of the flow via two-point correlations and energy spectra of the channel. To do so, the mean friction Reynolds number has been imposed to $Re_{\tau m} = 400$ and the temperature ratio difference between the walls to $T_{hot}/T_{cold} = 2$.

2 Numerical methodology

A full compressible DNS of the channel has been performed considering thermo-dependent properties of the air (density, viscosity, and conductivity). The fluid has been defined as an ideal gas, with the ideal gas-specific constant set as $R_g = 286.86\ J\ kg^{-1}K^{-1}$ and the pressure heat capacity is considered constant $c_p = 1004\ J\ kg^{-1}K^{-1}$ for an initial thermodynamic pressure of $P_0 = 101325\ Pa$.

Sutherland's law has been used to solve the dynamic viscosity (μ) and the conductivity (λ) at Prandlt number $Pr = 0.71$:

$$\mu(T) = 1.458 \cdot 10^{-6} \frac{T^{2/3}}{T + 110.4} \tag{1}$$

$$\lambda(T) = \frac{\mu c_p}{Pr} \tag{2}$$

Simulations are carried out by means of a low-dissipation high-order spectral elements in-house code (*Sod*). The code combines a spectral-method version of Galerkin's finite-element continuous model with a modified version of Guermond's Entropy Viscosity stabilization tailored to work with this spectral elements approach. The scheme used, employs an operator split to the convective terms that counter the aliasing effects of the reduced order integration imposed by using a SEM model. For the time-advancing algorithm, a fourth-order Runge-Kutta method has been used. Simulations are performed on a computational domain of $2\pi\delta$x2δx$\pi\delta$ (where $\delta = 0.003\ m$). The domain is periodic in the streamwise (x) and the spanwise (z) directions and a 3rd-order numerical mesh with 74.6M grid points has been used. The grid spacings are $\Delta x^+ = 9.1$, $\Delta y^+_{center} = 6.6$, $y^+_w < 1$ and $\Delta z^+ = 4.0$, normalized by the hot wall properties as $y^+ = y\ u_\tau \rho_w/\mu_w$.

3 Results

In this study, the temperature at the cold wall is set at $T_{cold} = 293\ K$ and the hot one at $T_{hot} = 586\ K$. After averaging, the computed mean Reynolds number is

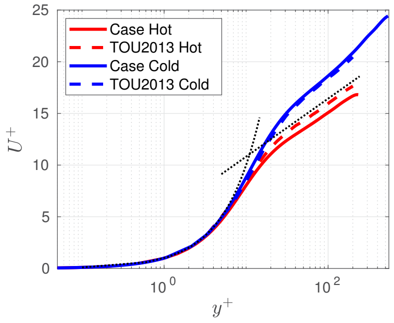

Figure 1: Mean velocity profiles at the cold wall (blue) and the hot wall (red) compared with Toutant and Bataille (2013) results (TOU2013).

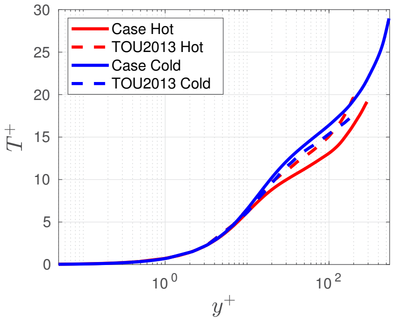

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Figure 2: Mean temperature profiles in wall units at the cold wall (blue) and the hot wall (red) compared with Toutant and Bataille (2013) results (TOU2013).

$Re_{\tau m} = 0.5(Re_{\tau cold} + Re_{\tau hot}) = 375.2$. At the walls, the friction Reynolds number is $Re_{\tau cold} = 514.6$ at the cold wall and $Re_{\tau hot} = 235.7$ at the hot one. The simulations are kept at subsonic Mach number ($M < 0.2$).

Figure 1 shows the mean velocity profile scaled by the local friction velocity u_τ along with the classical logarithmic law of the wall. Besides, in figure 2 the mean temperature profile can be seen scaled by the friction temperature T_τ computed with the wall heat flux $T_\tau = q_w/(c_p \rho_w u_\tau)$, where $q_w = \lambda_w(\partial T/\partial y)$. These results have been compared with the low-Mach computations of Toutant and Bataille (2013). In general good agreement with the results from the literature is observed, although some deviations in the mean temperature profile are observed.

The profiles expose the need of analysing new scaling methodologies for the cold and the hot wall to collapse. Compared with the isothermal channel (not shown here), the larger temperature gradient enhances the mixing, producing a decrease in the temperature inside the channel.

Figure 3 shows the instantaneous high and low-speed streaks, identified with contours at $u^+ = \pm 4$. They show significant differences between the walls. It can be seen that the fine-scale eddy structures in the cold wall are more than in the hot one. In fact, the flow regime at the hot wall exhibits a quasi-laminar flow regime.

In the final version of the manuscript, a detailed analysis of the effects of the temperature gradients on the turbulent structures will be performed via two-point correlations and energy spectra. In addition, the different coherent structures would be analyzed in detail.

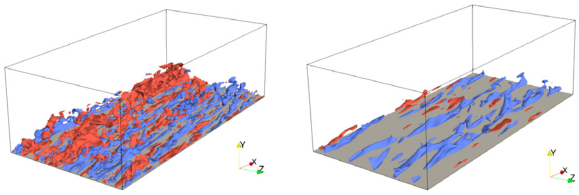

Figure 3: High and low-speed streaks ($u'^+ = 4$ in red and $u'^+ = -4$ in blue). Top for the cold wall and bottom for the hot one.

Acknowledgments

This work has been partially financially supported by the *Ministerio de Economía y Competitividad, Secretaría de Estado de Investigación, Desarrollo e Innovación*, Spain (ref. PID2020-116937RB-C21 and PID2020-116937RB-C22). We also acknowledge *Red Española de Supercomputación* (RES) for awarding access to the MareNostrum IV machine based in Barcelona, Spain (IM-2022-3-0010).

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