International Journal of Heat and Fluid Flow 50 (2014) 445–455

Contents lists available at ScienceDirect

International Journal of Heat and Fluid Flow

journal homepage: www.elsevier.com/locate/ijhff

DNS of turbulent droplet-laden heated channel flow with phase transition at different initial relative humidities

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ARTICLE INFO

Article history:
Received 29 July 2014
Received in revised form 8 October 2014
Accepted 15 October 2014
Available online 12 November 2014

Keywords:
Direct numerical simulation
Phase transition
Nusselt number
Turbulent channel flow
Relative humidity

ABSTRACT

In this paper a turbulent channel flow of a mixture of dry air and water vapor with water droplets is examined. Direct numerical simulation is used to quantify the importance of variations in the initial relative humidity. We focus on the droplet behavior along with the thermal properties of the system, such as the Nusselt number. During the initial stages of the simulations droplets evaporate more if the initial relative humidity is lower in order to reach the saturation condition. The difference in the Nusselt number between the cases of the lowest initial relative humidity and the saturation initial condition is on the order of 10% and this is connected with the different total heat capacity of the system. At the same time, we confront compressible and incompressible formulations comparing the results for both phases. A lower initial relative humidity leads to a larger difference in the mean gas mass density between the two formulations because of larger heat and mass transfer. Moreover, we find a larger relative difference in the Nusselt number between the two formulations in case of a lower initial relative humidity. These findings motivate the need to adopt the complete compressible flow model.

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1. Introduction

Turbulent flow fields laden with a large number of small relatively heavy droplets are present in many applications such as pollutant dispersion in the atmosphere and heat transfer in power stations. The dispersed phase not only undergoes momentum exchange with the carrier phase but, in addition, influences thermal properties of the flow. It is important to investigate how droplets affect the turbulent flow and in this paper we concentrate on the additional dynamic effects of coupling phase transitions to the flow. We compare the influence of differences in the initial relative humidity on the heat and mass transfer characteristics.

In the past fifteen years several studies were done in order to investigate droplet-laden turbulent flows. In Mashayek (1997) conducted an Euler–Lagrange simulation study of homogeneous turbulence with two-way coupling in momentum, mass and energy. A study of the mixing layer with embedded evaporating droplets was done by Miller and Bellan (1998). In Masi et al. (2010) investigated the interaction of a non-isothermal droplet-laden turbulent planar jet with a cloud of inertial evaporating droplets.

In this study we focus on wall-bounded turbulence by considering turbulent channel flow. We incorporate a dispersed droplet phase undergoing phase transition. As a point of reference we consider the flow of water droplets in air, in which the presence of water vapor is accounted for. Throughout the paper the carrier phase or carrier gas denotes a mixture of dry air and water vapor while water droplets will be referred to as the dispersed phase. The top wall of the channel is heated uniformly and the bottom wall is cooled in such a way that the total energy of the system is conserved (Bukhvostova et al., 2014). This leads to a temperature gradient in the wall-normal direction; in addition droplets tend to evaporate near the upper wall and water vapor tends to condense near the bottom wall. The key difference between the previously mentioned studies and the current is that we consider conditions in which not only evaporation of droplets but also their growth by condensation of the vapor phase is relevant.

The presence of phase transitions raises the question whether or not to include explicit compressibility of the carrier phase. The requirement of a constant mass density of the carrier phase in an incompressible formulation implies that the local changes in the mass density of pure air and of water vapor cancel each other precisely throughout the domain. An incompressible model was developed for this problem by Russo et al. (2014). In Bukhvostova et al. (2014) we confronted the incompressible model

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http://dx.doi.org/10.1016/j.ijheatfluidflow.2014.10.015
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with the fully compressible description, analyzing the consequences of non-constant mass density of the carrier gas and discussing for which quantities and under what conditions the full compressible formulation becomes essential. We performed a comparison between the results of the two formulations for "mild" initial conditions in the sense that mass transfer from the liquid to the vapor phase is modest. This was achieved by selecting room temperature, atmospheric pressure and 100% relative humidity (Bukhvostova et al., 2014). We consider the Nusselt number in order to quantify the heat transfer between the walls of the channel. For these initial conditions we found a difference on the order of 10% in the Nusselt number in the well-developed stages of the flow. We further investigated the agreement between the results from the two models increasing the value of the heat flux through the walls and increasing the mean temperature in the channel. More significant differences were observed: a difference on the order of 15% in the Nusselt number in the case of higher mean temperature and 5% difference in the mean droplet diameter near the walls in case of an increased heat flux.

These differences motivate us to investigate more "severe" physical cases of lower values of initial relative humidity varying it between 100% and 50%. The system tends to reach saturation independently of the initial relative humidity. The amount of water vapor in the carrier gas is increased by the evaporation of droplets and this process is more intense if the initial relative humidity is lower. In this paper we will study thermal properties of the carrier phase and droplet behavior for lower values of the initial relative humidity and also quantify the difference between the results from the two formulations for these cases.

The organization of this paper is as follows. In Section 2 the mathematical model is formulated for the coupled droplet-carrier gas system and the numerical methods for both formulations are presented. The initial conditions for the two models are described in Section 3. The differences in the behavior of the system under different initial conditions are discussed in Section 4. A comparison of the results from the two formulations is presented in Section 5. Finally, concluding remarks are collected in Section 6.

2. The governing equations and numerical methods

The models for the carrier phase are somewhat different in the compressible and incompressible formulations while the dispersed phase is modeled by the same set of ordinary differential equations in both models. The compressible model and the corresponding numerical method are the same as in Bukhvostova et al. (2014) and the incompressible model is the same as in Russo et al. (2014). For sake of completeness we briefly recapitulate them here.

2.1. The carrier phase in the compressible formulation

The carrier phase is a water vapor-air system in a channel, bounded by two parallel horizontal plates. In Fig. 1 a sketch of the flow domain is presented. The domain has a size of $4\pi H$ in the streamwise direction, which is denoted by x, and $2\pi H$ in the spanwise direction, z, where H is half the channel height. In addition, y is the coordinate in the wall-normal direction. We use periodic boundary conditions in the homogeneous directions (Kim et al., 1987) and no-slip conditions at the walls. A constant heat flux $\dot{Q}$ is applied through the walls. It is equal on both walls in order to conserve the total energy of the system.

The carrier gas is treated in the Eulerian manner as a compressible Newtonian fluid. We impose conservation of mass, momentum, total energy and water vapor. The equations can be written as (Miller and Bellan, 1998):

$$\partial_t \rho + \partial_j(\rho u_j) = Q_m \tag{1}$$

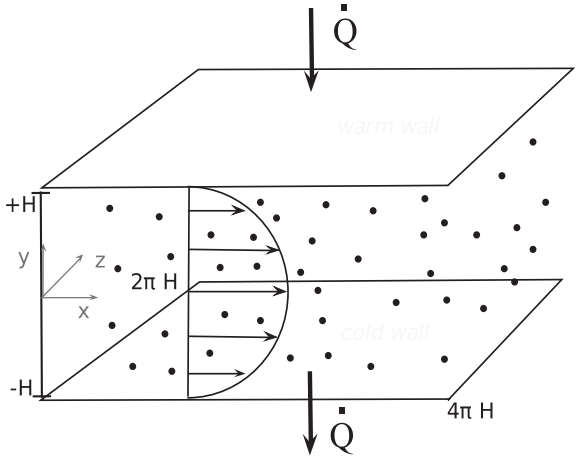

Fig. 1. The computational domain.

$$\partial_t(\rho u_i) + \partial_j(\rho u_i u_j) = -\partial_j \pi_{ij} + F_i + Q_{mom,i} \tag{2}$$

$$\partial_t e_t + \partial_j(\rho u_j e_t) = -\partial_j q_j - \partial_j(u_i \pi_{ij}) + Q_e \tag{3}$$

$$\partial_t(\rho Y_v) + \partial_j(\rho u_j Y_v) = -\partial_j J_{v,j} + Q_m \tag{4}$$

where $Q_m, Q_{mom,i}, Q_e$ are sink/source terms expressing the two-way coupling between the phases. These will be described later. In addition, (ρ, u_j, e_t, Y_v) are the carrier phase mass density, components of the velocity, total energy density and vapor mass fraction, respectively. Moreover, π_{ij} defines pressure and viscous contributions to the momentum flux; we use Sutherland's law to compute the dynamic viscosity. The term F_i is an external force density which is obtained from the conservation of total streamwise momentum. In addition, q_j denotes the components of the heat flux vector, which consists of heat transport by conduction and by diffusion and the vector J_v defines the diffusive mass flux of water vapor. The pressure and temperature of the carrier phase are denoted by p and T. The vapor mass fraction, p and T are connected by the ideal gas law for an air–water vapor mixture.

We incorporate the temperature dependence of thermophysical properties such as the thermal conductivity K, diffusivity D and dynamic viscosity μ. In fact, we adopt the Sutherland law for the dynamic viscosity μ (Sutherland, 1893) and keep the ratio of K and this dynamic viscosity constant. Similarly, the thermal diffusivity D is allowed to depend on temperature such that the ratio $\rho D/\mu$ is constant. Simulations reported in this paper are done under quite small temperature differences of up to approximately 3 K. It was found in Bukhvostova et al. (2014) that compared to a simulation in which the thermophysical properties are kept constant differences are less than 0.001% for several mean values. At larger temperature differences between the walls of up to 40 K), as studied, e.g., by Zonta and Soldati (2014), inclusion of the temperature dependency of K, D and μ was found to be more important, leading to changes of up to 30% in heat and momentum transfer coefficients arising from variations in local thermophysical properties on the order of 50%.

The system of governing Eqs. (1)–(4) is made non-dimensional using a set of reference scales of the system as will be further specified in Section 3. As a result, the final system of equations contains the Prandtl number Pr, the Mach number Ma, the Schmidt number Sc and the Reynolds number Re.

2.2. The carrier gas in the incompressible formulation

In the model by Russo et al. (2014) the carrier gas is also considered in the Eulerian way. The continuity equation and the

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