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Large-eddy simulation of thin film evaporation and condensation from a hot plate in enclosure: Second order statistics

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ABSTRACT

The archetypal case of a hot and wet plate surrounded by a cold and wet square enclosure is studied. Turbulent natural convection, the water evaporation/condensation, the heat exchange between the air and the enclosing solid bodies are simulated. The large-eddy simulation methodology is adopted in conjunction with a dynamic Lagrangian model for sub-grid scale viscosity and thermal-vapour diffusivities. Two statistical steady state simulations (maximum $Ra = 5 \times 10^8$) and three transitory drying-process simulations are carried out by deactivating and activating the air-solid heat transfer, respectively. The present work extends the companion study of Cintolesi et al. (2016), where first order statistics of the above mentioned cases were presented. Here, second order statistics are shown: first, the turbulent structures of the thermally uncoupled cases are analysed, along with the velocity root-mean square, the turbulent scalar fluxes and the turbulent kinetic energy budget. A few zones of negative production of turbulent kinetic energy are identified and discussed. The presence of splat and anti-splat events on the enclosure surface is detected and discussed. Subsequently, the evolution of the drying-process simulations is reported: the physical properties of the plate materials lead to different decays of the surface thermal fluctuations.

1. Introduction

The present work extends the analysis reported in the companion paper Cintolesi et al. [8] (henceforth CPA16). There a Large-Eddy Simulation (LES) methodology is used to study numerically the archetypal case of a hot and wet vertical plate placed in a cold and wet, square enclosure. In this context, it is worthwhile to mention that the use of LES allows exploiting flow features and high order statistics, not obtainable using standard methods relying on Reynolds-average simulations. The turbulent natural convection arising within the enclosure is reproduced, along with the evaporation from the water film onto the plate, the condensation on the enclosure surfaces and the conjugate heat transfer (CHT) between the air and the solid bodies. Different simulations are carried out. Initially, the CHT is switched off and two thermally uncoupled cases are investigated: the humid air case, where the evaporation-condensation process is allowed and the water film does not change its thickness; the dry air case, where air and solid surfaces are dry (hence evaporation and condensation do not occur). These cases are run until a statistical steady state configurations are reached. Subsequently, the final configuration for the humid air case is used to initialise three transient simulations, in which the CHT is switched on and the film thickness is allowed to vary in time. Such simulations are carried out changing the plate material: steel, PVC and porcelain are used for the plate, while the enclosure is always set of mild steel. The CHT reproduces the cooling of the plate and, consequently, the reduction of fluid motion and water evaporation. The thermal-physical properties of the plate materials strongly influence the cooling process and the overall evolution of the system.

Similar phenomena have been studied in literature, using different simulation techniques. Russo et al. [25] used Direct Numerical Simulations (DNS) to study a turbulent droplets-laden channel flows at low Reynolds, where the droplet distribution was simulated using a Lagrangian approach. Bukhvostova et al. [4] use the same Lagrangian approach, compering the performance of the incompressible and compressible solver. The discrepancies between the two were minimal for the fluid quantities, but relevant for the thermodynamic quantities. Overall, the compressible solver was more accurate. Laaroussi and Lauriat [18] investigated the thermosolutal convection and condensation of humid air in two-dimensional square cavity case, with thermal coupling between fluid and solid boundaries. A compressible low-Mach approach was adopted and compared with the incompressible

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Nomenclature

Symbols

ν	molecular kinematic viscosity
α	molecular thermal diffusivity
ω	water vapour concentration
ρ	space-time variable density
ρ^*	sum of air and vapour density
C_p	specific heat coefficient at constant pressure
k	thermal conductivity
g	gravity acceleration
β	expansion coefficient
L_h	latent heat of vaporisation
T	temperature
H	water film thickness
p	dynamic pressure
t	time
Γ	solid-fluid interface
η	second invariant of the normalised anisotropy tensor
S_{ij}	strain rate tensor
s_{ij}	fluctuation strain rate tensor
$S_{s/a}$	heat source/sink due to water change of phase
τ_{ij}	stress tensor
c_s^2	Smagorinsky constant for turbulent viscosity
c_t^2	Smagorinsky constant for turbulent diffusivity
k_e	turbulent kinetic energy (TKE)
P_K	TKE production
B_K	TKE buoyancy flux
ϵ_K	TKE dissipation
ϵ_{sgs}	SGS contribution to TKE
$\mathcal{T}_j$	TKE transport term

Non-dimensional numbers and parameters

Pr	Prandtl number
Ra	Rayleigh number
Re	Reynolds number
L	characteristic length
U_0	characteristic velocity
t_0	characteristic time
ω_{asy}	average vapour concentration in uncoupled case

Subscripts and superscripts

ψ	generic variable
ψ'	variation from mean value
$\overline{\psi}$	grid space filter
$\widehat{\psi}$	test space filter
ψ_a	air related quantity
ψ_s	solid related quantity
ψ_v	vapour related quantity
ψ_w	water related quantity
ψ_p	quantity evaluated on the plate
ψ_e	quantity evaluated on the enclosure
ψ_T	temperature related quantity
ψ_ω	vapour concentration related quantity
ψ_0	reference or characteristic value
ψ_{ij}	i,j-component, where $i,j = x,y,z$
ψ_{sgs}	sub-grid scale quantity
$[\psi]_{rms}$	root-main square

Vectors

$\mathbf{g}$	gravity acceleration
$\mathbf{u}$	air velocity
$\mathbf{U}_\omega$	evaporation/condensation velocity
$\mathbf{x}$	position vector

Boussinesq assumption: it resulted that both methods gave similar results when the initial temperature was uniform and equal to the average between the hot and cold isothermal walls. Iskra and Simonson [16] performed experiments on the three-dimensional rectangular ducts, both in turbulent and laminar regimes, while Raimundo et al. [24] analysed water evaporation and condensation across water free surface (simulations and experiments were performed). The problems related to conjugate heat transfer have been also studied: a review on this subject can be found in Dorfman and Renner [9]; a description of the main numerical coupling strategies is reported by Duchaine et al. [11,12]; while the technique herein used are described and validated by Sosnowski et al. [27].

In CPA16 the authors presented the first order statistics and the description of the gross motion for the aforementioned simulations. For the cases named *preliminary cases* in CPA16 and hereafter referred as *thermally uncoupled cases*, the mean flow was described along with the effects of the thermal-vapour stratification. The buoyancy force generated near the hot plate and near the vertical cold enclosure walls drives the flow. Three regions of high-speed motion were identified: (i) the ascending region, just above the plate where the hot air rises from the hot plate; (ii) the horizontal-flow region, near the enclosure horizontal wall (ceiling) where air flows towards the vertical walls of the enclosure; (iii) the descending region, near the upper part of the vertical walls of the enclosure where the buoyancy force pushes air downward. An additional region of low-speed flow, named the diagonal-flow region, appears when air is driven from the descending to the ascending regions across the cavity. A strong stable thermal-vapour stratification sharply splits the cavity into a hot-humid upper part and a cold-dry bottom part. Stratification inhibits air recirculation over the entire cavity height, thus the motions remain confined in the top-half of the cavity. Flow velocity and the stable stratification are more energetic for the humid case than for the dry one. Also, the heat transfer at the solid surfaces were studied, showing that evaporation and condensation rule the fluid–solid heat exchange. For the drying-process cases, the time evolution of active scalars (*i.e.* average temperature and vapour concentration) were reported, together with a description of the main flow characteristics all over the domain and along selected lines. The cooling process has been simulated for 60 s of physical time; this initial interval is crucial to investigate the different evolution of the systems associated to the variation of the plate materials. It was found that the thermal inertia ρC_p of the solid media controls the cooling process: when the average plate temperature is plotted in time, the steel plate (high ρC_p) exhibits a linear and moderate reduction of temperature, while the porcelain plate (low ρC_p) shows a higher cooling, with an initial strong and subsequent moderate decrease of temperature. PVC plate (moderate ρC_p) displays an intermediate behaviour. The evaporation of the water film on the plate and its condensation over the enclosure were also analysed: the water film thickness varies according with the temperature evolution. An estimation of the dew-point temperature showed that recondensation onto the plate is not allowed, for the materials considered in the study.