On the discrepancy of modelling the heat transfer
for pure natural convection

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ABSTRACT

In the present study, buoyancy driven flow fields in a bench scale apparatus are experimentally investigated by means of a non intrusive measurement technique, here the 2D-2C particle image velocimetry (PIV). An electrically heated copper block was used to induce a plume. As the temperature at the surface of the heating source is the main driver for the fluid dynamics, the surface temperature field was measured by thermography. The experimental results are compared to simulations with the Fire Dynamics Simulator (FDS). Simulations with a prescribed, homogeneous distributed surface heat flux and a 3D thermal heat conduction with a volumetric heat source were carried out. The impact of both approaches on the resulting flow fields showed no significant deviations, however, both do not agree with the experimental data. Whereas the surface temperatures of the heating source and its vicinity are in a good agreement for the 3D heat transfer approach. This leads to the conclusion that the modelled heat transfer mechanism does not provide valid predictions in this setup.

KEYWORDS

buoyancy driven flow, convective heat transfer, fire dynamics simulator (FDS), computational fluid dynamics (CFD), 3D thermal heat conduction, thermography, validation, particle image velocimetry (PIV), bench scale
INTRODUCTION

Convective heat transfer plays a major role in the cooling and heating of solids. In fires, the hot gases heat the ceiling and other structural parts of the building. If there is no forced convection, the dynamics of the flow is purely due to natural convection, which is driven by the heat flux in the near wall region, i.e. the boundary layer. As the boundary layer is in general not resolved in LES codes, like FDS [1], it needs to be modelled. The common approach is to use wall functions, which mimic the empirical correlations. In order to study the thermal convective flow, a basic experimental setup is chosen, which deliberately excludes phenomena like pyrolysis or combustion.

Meunders et al. performed a series of bench scale experimental investigations of buoyancy driven flows [2] in an enclosure. An electrically heated copper block was used as the heating source. The block induced turbulent or transitional plumes, depending on the heating power, and the velocity field was measured with the PIV technique. The comparison of the data from these validation experiments to the simulation results computed with FDS (version 6.2.0) showed deviations in the flow fields [3]. Initially, the deviations were suspected to be PIV measurement errors. The comparison of the PIV and additional laser Doppler velocimetry (LDV) results showed a good agreement and thus, an experimental error regarding the velocity measurements was excluded [4]. In the initial experimental setup, the block was just placed on the floor. Here, a significant influence of the geometry of the heating source on the turbulence induction, particularly at the upper edges, was observed [5]. To avoid the flow separation at the upper edges, the heating block was immersed in the floor.

For this new setup, preliminary FDS (version 6.5.1) simulations were run with a prescribed total heat flux. The results indicated the development of a plume with no indication of flow separation at the heating source and therefore the setup was physically implemented. Again, the focus was on the PIV measurements of the flow field and the thermal imagining of the surface of the heating source and its vicinity. However, the comparison of the experimental and the numerical velocity fields showed significant deviations. Furthermore, in contrast to the simulations the experimental plume was very unstable, which led to the assumption, that less heat is released at the surface than predicted by the model. This observation lead to the here presented experimental and numerical investigation of the surface temperatures.

In this paper we outline the comparison of the experimental results, i.e. velocity fields and surface temperatures, to FDS simulations. The error induced by the initially assumed boundary conditions in the preliminary simulations is discussed. Furthermore, the validation experiments and simulations for the 3D thermal heat conduction model in FDS are presented.

EXPERIMENTAL AND NUMERICAL SETUP

In figure 1 (left column) the technical drawing of the heating source and the surrounding insulation is presented. The copper block is heated by an electrical resistor, which is placed in a borehole in the symmetry axis. To avoid heat losses the block is insulated with a 20 mm thick calcium silicate plate, which has a low thermal conductivity. For mounting purposes the copper block and the insulation are pasted in a wooden shell. The material properties are listed in table 1. The heating source is mounted to a 760 mm × 10 mm × 760 mm stainless steel plate (X6CrNiMoTi17-12-2), which is the floor of the enclosure. For the PIV measurements the flow is to be seeded with particles. In this experimental setup DEHS particles were used, which is a synthetic oil, and are supposed to have good gas flow following properties [6]. The particles were generated by an atomizer, which is connected to an outlet at the floor plate of the enclosure. The floor plate has a second connection to the environment, which is used to remove the particles by a vacuum cleaner at the end of an experimental. During the run the connection is open, thus ambient pressure can be assumed in the enclosure. Both connections are located at the outer edge of the floor, so the influence on the flow field is assumed to be negligible. The experimental volume is enclosed by PMMA walls (width × height: 760 mm × 1440 mm). These walls are insulated with an extruded polystyrol (XPS) plate to reduce the impact of the environment on the boundary conditions.
Figure 1: Technical drawing of the heating source (upper left), the floor plate of the experimental setup (bottom left), and the thermocouple instrumentation at the bottom and side of the wooden shell (middle column). A photo of the full experimental setup is on the right side.

One of the enclosure walls is equipped with optical openings for the IR-camera measurements. In the opposite side wall a small slot for the laser sheet inlet is added. Further openings in the insulation were created for the PIV cameras, which were mounted perpendicular to the light sheet. The top of the enclosure is designed as a plate heat exchanger, which is supplied by water at a constant temperature. So, the temperature at the ceiling is assumed to be constant. The full setup is depicted in figure 1 (right column).

Table 1: Material properties of the heating source and insulation.

| Material         | \(c_p\) / kJ kg\(^{-1}\) K\(^{-1}\) | \(\lambda\) / W m\(^{-1}\) K\(^{-1}\) | \(\rho\) / kg m\(^{-3}\) |
|------------------|--------------------------------------|--------------------------------------|--------------------------|
| Copper           | 0.385                                | 400                                  | 8920                     |
| Insulation (Promat) | 0.95                                  | 0.083                                | 450                      |
| Wood             | 1.7                                  | 0.3                                  | 1350                     |
| Stainless steel  | 0.5                                  | 15                                   | 8000                     |

As the top of the enclosure is basically an aluminium plate, it could not be used as an optical access for the IR camera, the laser and the CCD cameras. Thus, the IR camera had to be mounted with an inclined view on the centred heating source. To keep the emissivity of the materials constant, chalk spray (\(\varepsilon = 0.9\)) was applied on the floor surfaces. The thermography’s measurement uncertainty is stated with ±1.5% or ±1.5°C. The light sheet optics and the CCD camera of the PIV system were mounted on a traversing unit, so the velocity field could be observed in different heights. The measurement errors of the PIV measurements depend on various factors, that were studied by multiple authors [6, 7, 8]. Following these studies, we assume a measurement uncertainty of 6% in the near field of the heating source, where the flow is assumed to be laminar.

Further temperature measurements, i.e. at one side of the wooden mounting device and the bottom of the wooden shell, have been instrumented with thermocouples (CO3-K). Four locations at the side and four at the bottom were selected. Figure 1 (middle column) shows the according positions (M00-M07). The thermocouples have a measurement uncertainty of ±0.75% but at least ±2.2°C. The sensor
itself has a circumference of 0.25 mm, leading to a radius of 0.04 mm, which is the assumed positioning uncertainty.

Two kinds of FDS simulations were carried out: with a prescribed net heat flux at the surfaces and a 1D heat transfer in the solid (1DHT, full enclosure with the active cooling at the ceiling, 4 mm grid resolution), as well as with a prescribed volumetric heat source inside the copper block and a 3D heat transfer using FDS version 6.6.0 (3DHT, vicinity of the heat source, no enclosure, 2 mm grid resolution). The choice of these grid resolutions is based on sensitivity studies.

Regarding the 1DHT case, only a part of total heating power contributes to the heat flux through the floor into the enclosure. To quantify this fraction, detailed heat conduction simulations in ANSYS CFX (Version 18.1) have been computed. E.g. at an input power of $P_{el}= 75.4 \, W$ the net heat flux is at least 60% of the total electrical power.

In the 3DHT case, the modelling of the electrical resistor is based on a volumetric heat source. For this purpose, a centrally located region, extension of $4 \, mm \times 4 \, mm \times 38 \, mm$, is defined inside the block. It delivers the previously defined heating power directly to the copper block. The dimensions of the thermal insulation as well as the wooden shell are identical to the experimental setup, as shown in figure 1 (left column). The material properties are set according to table 1. Although all relevant solid structures in the floor are modelled, the gas phase is modelled only up to a height of 70 mm.

COMPARISON OF EXPERIMENTAL AND SIMULATION DATA

The experiments have been performed within a heating power range of 1.4 W to 75.4 W, but in this contribution only a part of the experiments is presented. The experimental and modelled velocity fields are compared along horizontal lines, cutting the plume, at three selected heights and for an experiment with an input power of $P_{el}= 75.4 \, W$. In figure 2a the profiles of the averaged velocity component in the z-direction (vertical) are shown, where the simulation data is based on computations with the 1D heat transfer model (solid line) as well as with the 3D (dashed line) one, but in a reduced computational domain. The data shows, that inside the simulated plume the peak velocity is at least twice as high as observed in the experiments (dotted line). As the convective heat transfer is the driver for the gas dynamics, this suggests, that the convective heat transfer in the simulation is overestimated. Therefore, the thermal boundary conditions, i.e. surface temperatures, are investigated more in detail.

Motivated by the discrepancy in the flow field, the thermocouple and thermographic measurements are compared to FDS simulations, where the 3D thermal heat conduction is taken into account. The block’s top and its vicinity temperature are compared to the simulation results in figure 2b. Figure 3 compares the localised thermocouple measurements at the side and bottom of the block for three input power settings. Taking the measurement uncertainty into account, the modelling results are in good agreement with the experimental data. This observation leads to the conclusion, that the 3D thermal heat conduction model provides valid results in this setup. The small deviations are due to measurement uncertainties as well as the uncertainty of the exact material properties. However, small temperature differences at the solid surface may lead to significant changes in the gas phase temperature, as the ratio of the heat capacities of the solid and the gas is large. Which in turn results in a different gas dynamics.

CONCLUSION AND OUTLOOK

The temperature measurements at the surfaces of the copper block are in a good agreement with the simulations in FDS, taking the 3D thermal heat conduction model into account. This is also the case for the not presented power inputs. However, the comparison of the flow field shows significant deviations. As the convective heat transfer at the surface of the copper block is the main driver for the fluid dynamics, it may lead to the conclusion, that the modelling of the near wall phenomena by means of standard temperature and velocity wall functions may not be valid here. Hölling already showed, that the standard wall functions are not valid for natural convection at a vertical heated wall [9]. Experiments
40
20
0 20 40
position / mm
0.0
0.1
0.2
0.3
0.4
0.5
0.6
0.7 velocity / m/s
exp, 50 mm
exp, 100 mm
exp, 165 mm
sim, 1DHT, 50 mm
sim, 1DHT, 105 mm
sim, 1DHT, 165 mm
sim, 3DHT, 50 mm

(a) Vertical velocity profiles through the plume at three heights: 50 mm, 100 mm, and 165 mm. Shown are experimental and simulation (1D and 3D heat transfer) results with an input power of $P_\text{el} = 75.4$ W.

40
20
0 20 40 60
position / mm
50
100
150
200
250
300
350 temperature / $^\circ$C
experimental, 75.4 W
simulation, 75.4 W
experimental, 21.0 W
simulation, 21.0 W
experimental, 10.0 W
simulation, 10.0 W

(b) Temperature profiles along the x-axis on the surface of the heating source for various heating powers. The vertical lines indicate the boundaries between the copper block ($|x| < 30$ mm), insulation ($30$ mm $< |x| < 50$ mm), wooden shell ($50$ mm $< |x| < 60$ mm) and the steel plate ($60$ mm $< |x|$).

Figure 2: Comparison of experimental and simulation results.

by Hundhausen et al. determined increasing deviations of the dimensionless velocity profile for mixed convection (buoyancy aided) conditions from the standard profile with increasing buoyancy effects, i.e. increasing Grashof number [10]. In the presented experimental setup, pure natural convection at a horizontal flat plate occurs. But the standard wall functions were developed for forced. Considering the transport equations, the main drivers for the convective heat transfer are the temperature and velocity gradients in the viscous sub-layer ($0 < y^+ < 5$). With increasing gradients the convective heat transfer rises. For natural convection the velocity gradient $\partial u/\partial y$ is lower than for forced conditions. Furthermore, the temperature profile is coupled to the velocity profile. Thus, the heat transfer for pure natural convection is overestimated, which might be an explanation for the deviation in the observed velocity fields. Considering the Nusselt correlation for a flat plate confirms the assumption, that FDS overestimates the convective heat transfer. The transfer coefficient $h$ is computed in LES mode as a combination of natural and forced convection correlations [1]

$$h = \max \left( C |T_g - T_w|^\frac{1}{2}, \frac{k}{L \text{Nu}} \right),$$

with the gas $T_g$ and wall $T_w$ temperature, a constant $C$ [11], the gas thermal conductivity $k$, a characteristic length scale $L$ and the Nusselt number Nu.

In the next steps, the validity of the above stated correlation in the presented setup will be investigated. Additionally, a direct numerical simulation (DNS) for a detailed analyses of the near wall phenomena at the heating source will be carried out and compared with the LES simulations. Furthermore, a sensitivity analyses regarding the influence of minor deviation of surface temperature on the quantities of the velocity field is planned.

OPEN ACCESS DATA

The experimental and simulation data presented in this contribution is publicly available [12].
Figure 3: Thermocouple measurements at the surfaces of the wooden shell (bottom, side). The measurement uncertainties are indicated by the red error bars.

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