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INERT WALL THERMAL FEEDBACK VALIDATION STUDY

Lucie Hasalová, Vojtěch Šálek, Petr Hejtmánek & Milan Jahoda

Technical Institute of Fire Protection
Prague, 143 00, Czech Republic
e-mail: lucie.hasalova@hzscr.cz
University of Chemistry and Technology
Prague, 166 28, Czech Republic
e-mail: vojtech.salek@vscht.cz, milan.jahoda@vscht.cz
Czech Technical University in Prague
Prague, 166 28, Czech Republic
e-mail: petr.hejtmanek@cvut.cz

ABSTRACT

Thermal feedback from the burner to the inert wall was studied experimentally and compared to the FDS model prediction. Emphasis in this study was put mainly on repeatability of the experiments for validation purposes and a grid cell size sensitivity study for a problem of inert wall fire. The presented contribution is a work in progress with the long-term goal to provide material properties for the problem of engineered wood products (EWPs) vertical wall flame spread validated over real scale experiments.

INTRODUCTION

The long term goal of this project is to provide a pyrolysis model input parameter data set of engineered wood products e.g. oriented strand board or medium density fiberboard to be used in CFD flame spread models. Methods of obtaining these parameters from the micro scale and bench scale experiments are well discussed in the literature (Chaos et al, 2011; Kim & Dembsey, 2015; McKinnon, 2016). However, these parameters are largely model dependent and are rather calibrated to experimental conditions during measuring or inferring the parameters than validated (Bal & Rein, 2015). In order to validate the pyrolysis model input data set derived from these experiments, experiments of burning 2×1 m vertically placed EWP´s boards were proposed.
The problem of a vertical flame spread over a combustible material is a complex modeling task. Turbulent buoyant flow is coupled with gas phase kinetics, heat transfer and last but not least solid phase thermal decomposition (Wong et al., 2013). In such simulation, a source of discrepancy between model and experiment when judging either the model predictive capacity or quality of model input parameters can be compensated by many mutually influencing factors. In order to decouple pyrolysis from the model process and focus only on the remaining aspects of the CFD flame spread simulation, flame spread over an inert wall was first studied. A grid sensitivity study was performed comparing the inert wall temperature, net heat flux to inert wall and gas phase temperature in order to accurately predict thermal feedback from the burner to the wall and surroundings before pyrolysis will be introduced to the model.

EXPERIMENTAL PART

Experimental set-up

The layout of the experiment can be seen in Fig. 1. All experiments were carried out in the Room corner test (RCT) facility (ISO 9705-1:2016) consisting of a room (2.4×3.6×2.4 m) with one door opening (0.8×2 m) and adjacent extraction hood. A fiber cement panel (Fermacell Powerpanel H_2O)

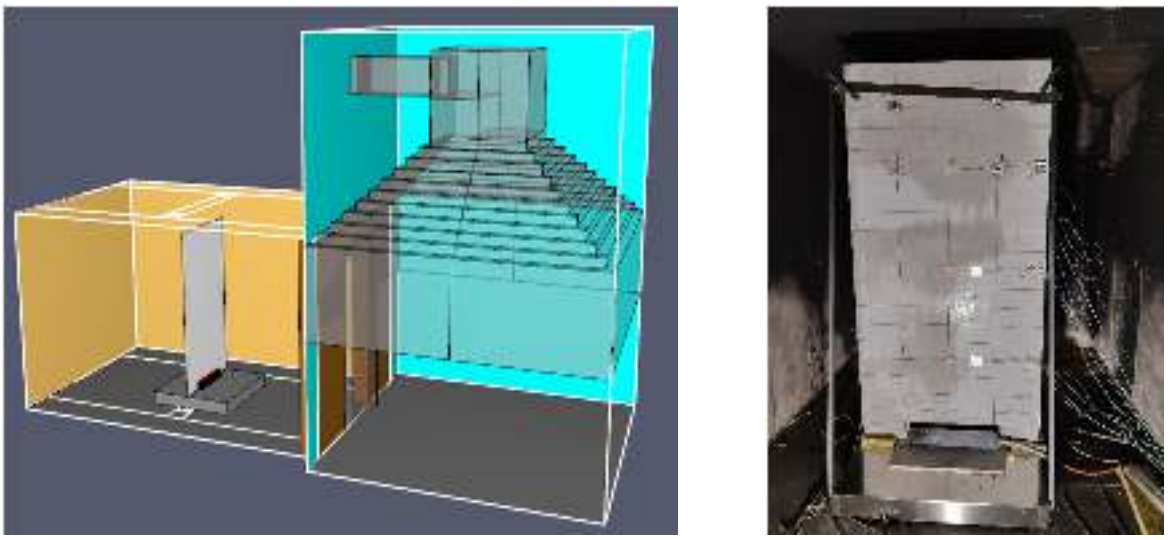

Figure 1: Schematic diagram of the test room with an inert panel. Picture of the panel fixed in position with a metal frame and with a sand burner at the lower edge of the panel.

of dimensions 2×1 meter and 12 mm thickness was placed in the middle of the test room. The panel sat on a scale approximately 15 cm high and was fixed in position with a metal frame attached to the panel 10 to 15 cm under its upper edge. A sand burner of dimensions 50×5×10 cm was positioned right next to the front side of the slab in the center of the panel. The back side of the panel was exposed.

Test scenario

Propane was used as a fuel. The gas flow of propane was set to achieve a constant heat release rate (HRR) of 30 and 50 kW. Two values of propane burner power were tested to ensure that model parameters calibrated over experiments at 30 kW burner power can be extrapolated to different conditions. In this paper, only 30 kW burner power results are presented. The exhaust hood was set to a flow of 0.8 m^3/s in the exhaust duct and was turned on 120 seconds before the propane burner was ignited. The duration of the test was 1200 seconds. The test was terminated immediately after the burner was shut off.

Measured quantities

The location of the measuring devices used in all tests is depicted in Fig. 2. Front and back side panel temperature were measured using bare wire K-type thermocouples with a 0.5 mm bead (standard accuracy - greater of ±2.5 °C or 0.75 %). To shield the bead from the flame and radiative heat and to ensure good contact with the panel, thermocouple beads were covered and pressed to the panel with 3 mm thick 3×3 cm pieces of insulation sheet (Nefalit) manufactured from mineral wool fibers. Front side temperature was measured along the center line of the slab and 10 cm from the right side at three heights – 50, 100 a 150 cm (T1-T6). The temperature at the back side of the slab was measured only along the centerline at the same location as on the front side (T7-T9).

Net heat flux was measured at four locations marked R1 to R4 on the front side of the panel with water cooled Schmidt-Boelter type heat flux gauge. A hole was drilled in the panel and the gauge was stuck through the panel from the back side so the sensitive part of the sensor is in the plane with the front wall. The measuring range of the gauges was 20 kW/m^2 for R2 and R3, 10 kW/m^2 for R4. R1 measuring range was 50 kW/m^2 for three tests and was replaced by 20 kW/m^2 range heat flux gauge for the remaining two tests.

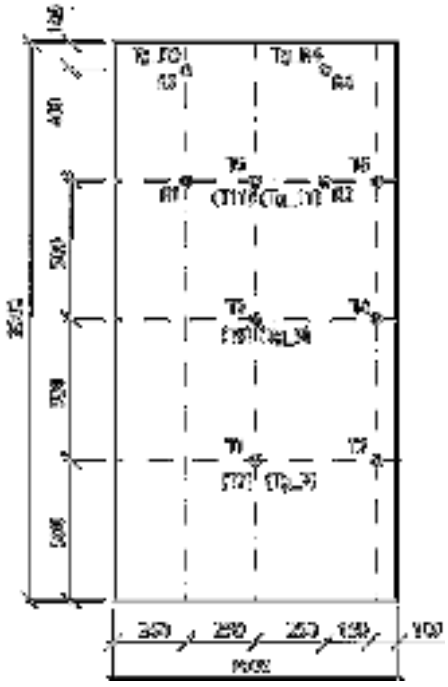

Figure 2: Location of heat flux gauges - R and thermocouples – T on the panel. Values in brackets denote devices on the back surface of the slab. T_g denotes gas phase temperature in the vicinity of the surface.

Gas phase temperature was measured at the location close to the heat flux sensor surface at positions R3 and R4 and at the back side at locations T7, T9 and T11 to monitor conditions close to the panel. For monitoring the room conditions, insulated 1 mm diameter K-type thermocouples (standard accuracy - greater of ±2.5 °C or 0.75 %) were used. Room temperature in time was measured in the front left corner of the room (30 cm from side and front wall) at seven positions along the height of the room and at five positions under the ceiling over the entire cross section of the room according to ISO 9705-1:2016. A video camera was placed on the ground in the door opening to monitor the flame height and evolution of the neutral plane.

Repeatability of experimental runs

When comparing model predictions to experimental data to validate model or material parameters, conclusions need to be drawn in the context of the experimental dataset uncertainty. Uncertainty is connected to each measured quantity mainly due to the instrument used and repeatability of results but also human limitations as well as the factor of randomness.

Repeatability tests and measuring instruments uncertainty are especially important when conclusions are made whether a particular model or material parameter improves model prediction. Five experiments with the same geometrical set up and same measuring devices (except heat flux gauge R1) were performed. The only difference between the experiments were the initial conditions in the room, namely the ambient temperature. After each experiment, the room and the panel were left to cool but if more experiments were performed over a course of the day, heat accumulated in the room makes it time consuming to ventilate and the initial room and panel temperature slightly varied. The same panel was used for all experiments.

Experimental results

Fig. 3 shows selected experimental results to demonstrate the main important observations about the experimental data set. Net heat flux is presented as an average value with error bars depicting

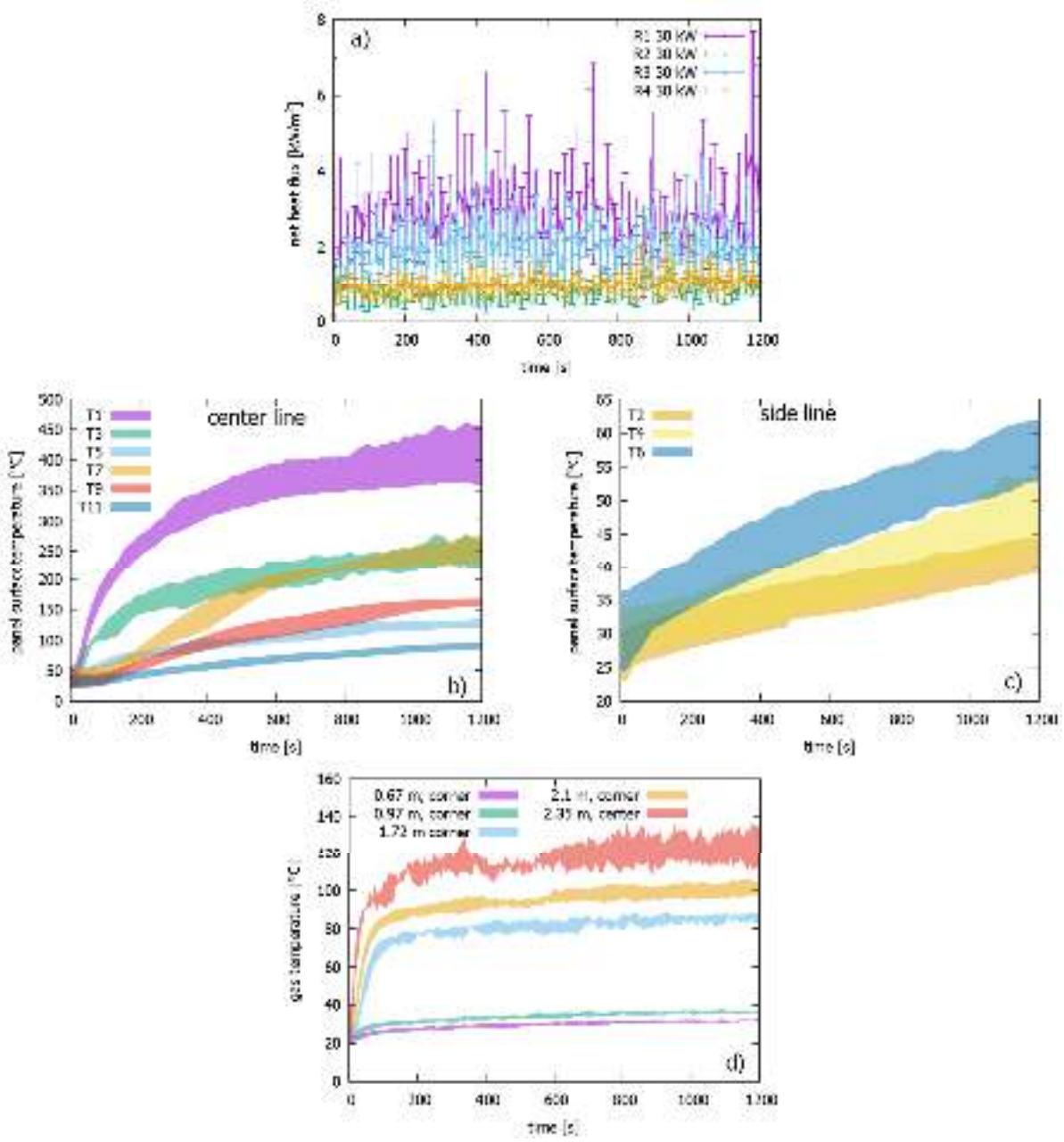

Figure 3: Experimental results a) net heat flux to the wall, b) center line front and back wall panel temperature, c) side line front wall panel temperature d) gas phase temperature in the left front corner at various heights and in the center of the test room.

the minimum and maximum values. According to the manufacturer, the calibration accuracy is ± 3 % at full scale, however, the accuracy of measurement may be significantly influenced by the so called non-linearity error. It is recommended to use a heat flux sensor at a flux level higher than 50 % of the full scale range. The data presented in Fig. 3a include neither the calibration accuracy nor non-linearity error. The largest uncertainty is expected for the R1 sensor, where the non-linearity error may lie around 50 %.

The smallest uncertainty is expected for the R4 sensor, around 10 %. The data acquisition frequency was 1 Hz. Every 10th data point was printed in the plot. Measured net heat flux was strongly oscillating, however, the trend can be clearly seen. Net heat flux was relatively constant over the whole duration of the experiment. Depending on the location, net heat flux ranged between 1 to 3 kW

on average. Heat flux on the right side of the panel (R2, R4) is lower compared to the left side of the panel (R1, R3). This behavior was observed also at the 50 kW burner power suggesting that the heat flux is not evenly distributed over the panel area although the set-up is symmetrical. The soot patterns on the panel after the experiments were also not symmetrical and tilted to the left side. However, for conclusions to be drawn, measurement uncertainty for each sensor in the experiment has to be firstly carefully evaluated.

Panel surface temperatures are presented in Fig. 3b) and 3c) as filled lines with boundaries being the minimum and maximum dataset value. Thermocouple measuring uncertainty is not included in the plotted data and lies at maximum around ± 3.8 °C (0.075 % of 450 °C). 600 seconds from the ignition, the front wall centerline temperatures continue to grow very slowly, at 1000 seconds from ignition steady state seems to be reached. The largest span in measured wall temperatures between five experiments was measured in the flame region (thermocouple T1) and reached 90 °C. The back wall centerline temperatures grew very gradually after 600 seconds from burner ignition. Front temperature 10 cm from the panel edge (Fig. 3c) did not reach a stable state and was continuously rising over 1200 seconds. Fig. 3c also demonstrates the span of the initial wall temperature.

Gas temperature in the room at 0.67, 0.97, 1.72 and 2.1 meters above the floor in the left front corner of the room is plotted in Fig. 3d. Very slight and gradual increase may be observed after 200 seconds to the end of the experiment. Both hot and cold gas layer temperatures have fairly good repeatability between the experiments. Due to the equipment malfunction, room gas temperature was measured only in three runs. The largest span in the measured gas temperature was recorded on the thermocouple in the center of the room under the ceiling and was around 25 °C. The span of initial gas temperature in the room under the ceiling is 20 °C. The thermocouple tree measuring accuracy was tested using the thermocouple calibration furnace. In the temperature range up to 600 °C, the difference between the reference and thermocouple tree reading fell within ± 3 °C. The standard accuracy declared by the manufacturer is 2.5 °C.

FDS MODEL

The experimental set-up and test scenario described in the previous section were modeled using FDS software version 6.7.7. Parameters regarding the gas flow solver and radiation solver were kept default. Emphasis was put, in the first step, on the sensitivity of the inert Fermacell panel model on the grid cell size. Front and back panel temperature, net heat flux on the wall and gas phase temperature were outputted for various mesh sizes and compared to experimental results.

Model set-up

Geometry and materials

The geometry of the model copies the geometry of the room. The walls, floor and ceiling of the test room are modeled as layered surfaces with assigned thermal properties. Except the front wall of the room that creates the separation between the room and the exhaust area, all three walls, floor and ceiling of the test room were modeled as vent, lying on the mesh boundary. Walls were modelled as inert. The thickness of the floor and ceiling for heat transfer calculations equaled the physical thickness with particular layers of fiber cement panel, mineral wall insulation and aerated concrete blocks on the ceiling and concrete on the floor (both in the test room and exhaust area).

The Fermacell panel was modeled as the 12 mm thick one-layer, one cell thick material with BACKING set to exposed. Properties of fiber cement panel were taken from the manufacturer - density equals 1000 kg/m^3, specific heat and thermal conductivity independent of temperature equals 1 kJ/(kg·K) and 0.17 W/(m·K) respectively. Emissivity was left default 0.9. The only setting of the FDS solver that was modified from the default was related to the heat conduction in solids. As the thermal feedback from the surroundings to the wall is the main interest of this study and the back wall surface is exposed, the quality of the solid phase grid is crucial. CELL_SIZE_FACTOR parameter of the panel

was set to 0.5 reducing the solid grid cell size compared to default and STRETCH_FACTOR was set to 1 generating a uniform mesh within the panel.

Boundary conditions

The exhaust area around the hood, except the floor, was modeled as open to the surroundings. The exhaust volume flow of 0.8 m^3/s was set as a vent on the exit part of the exhaust duct. Burner with prescribed HRRPUA (Heat Release Rate per Unit Area) 1200 kW/m^2 (equals to 30 kW propane burner HRR) was applied as a vent on the upper face of the inert sand burner block. The exhaust hood and duct were modeled as inert surfaces as well as the balance the Fermacell panel was placed on.

Gas phase reaction

Propane, as the only gas fuel in the domain, is a pre-defined FDS tabulated specie. Its thermos-physical properties are calculated by FDS based on the specified formula of propane. Complete combustion of propane was assumed, zero CO and soot yields were specified.

Output quantities

Wall temperature for comparison with the thermocouple readings was outputted as a solid phase quantity "WALL TEMPERATURE" and "BACK WALL TEMPERATURE". Gas phase temperatures measured on the thermocouple tree were modeled as a quantity "THERMOCOUPLE" with specified bead diameter. For comparison with experimental heat flux gauges reading, the quantity "NET HEAT FLUX" was outputted. It is important to consider when evaluating the model and experimental data agreement that the net heat flux measured by the heat flux gauges is a heat flux to the cold wall due to gauge water cooling compared to the model, where the wall temperature is increasing with time.

Grid sensitivity study

Correct thermal feedback to combustible material is a key factor that influences the thermal degradation and thus the prediction of the flame spread over the combustible material. Temperature and flow conditions close to the wall need to be realistically captured with a model to predict the vertical flame spread. The thickness of the boundary layer for the problem of vertical wall flames is estimated to be in order of 1 cm. The grid resolution to fully resolve boundary layer and wall gradient in LES simulation of vertical wall flame spread is in order of 1 mm (Brown et al., 2018). In this work, five different grid cell size meshes were created. The goal was to quantify the difference between heat feedback predicted by very fine millimeter grid cells at the wall surface with model predictions in a case of a grid that is typically used in the FDS applications, in order of centimeters. All model predictions are then compared to experimental results which allow us to further calibrate the model parameters.

Five different mesh cases are summarized in Tab. 1. The second column of the table states the grid cell size in the axis perpendicular to the panel starting from the panel front surface. In the bracket is the number of cells of the particular size. The grid cell size was gradually enlarging further from the panel. It wasn´t possible to maintain the same size of meshes. Number and location of meshes for a very fine grid resolution were proposed to maintain on average 800 000 grid cells per mesh and were limited to 80 meshes in total due to computational resources. This also resulted in different pattern of coarsening the meshes between the five cases. However, couple of rules were adhered to when creating the meshes. The back side of the inert wall was never located at the mesh boundary and the whole burner was placed within meshes of the same grid cell size. The smallest number of grid cells in each direction was 10. Compared to the experiment, the burner in the model did not sit right on the panel. That is due to the FDS restriction on building geometry and solid material heat conduction. 1 cm gap between the inert wall and burner was created in the model. Thus, for the 2, 5 and 10 cm grid size mesh in the domain, the 1 cm grid region has to be set to stretch over the panel and burner. In the 10 cm mesh case, this resulted in a large volume ratio on the mesh boundary. The required

Table 1: The mesh characteristics for variously refined mesh in the model.

Mesh ID	Grid cell size in direction perpendicular to the panel (number of cells in this direction)	Total number of cells	Number of meshes	Simulation time (s)
1 mm	1mm (12) - 2mm (26) - 4mm (9) - 2cm (95)	65 902 500	80	10
2 mm	2 mm (20) - 4 mm (10) - 8 mm (10) - 4cm (46)	22 222 800	36	40
1 cm	1cm (20) - 2cm (10) - 10 cm (17)	804 084	11	600
5 cm	1 cm (20) - 5 cm (38)	1 070 496	6	394
10 cm	1 cm (20) - 10 cm (19)	637 812	6	585

simulation time is 600 seconds. Tab. 1. shows actual simulation time calculated up to date. 10 mm grid simulation time is only 10 seconds which is too short to a drawn conclusion in comparison to coarser grids. 2 mm grid simulation time is 40 seconds which is sufficient to show trends in comparison to coarser grids.

Model results – gas phase

Fig. 4 shows predicted and measured gas temperature in the corner of the room in the cold layer. Fig. 5 shows the gas phase temperature in the hot layer in the corner of the room and in the center of the room above the inert panel. The fit of the model to experiment is, in general, satisfactory. Model slightly underpredicts the gas temperature in the hot layer (around 10 to 15 °C) in the corner of the

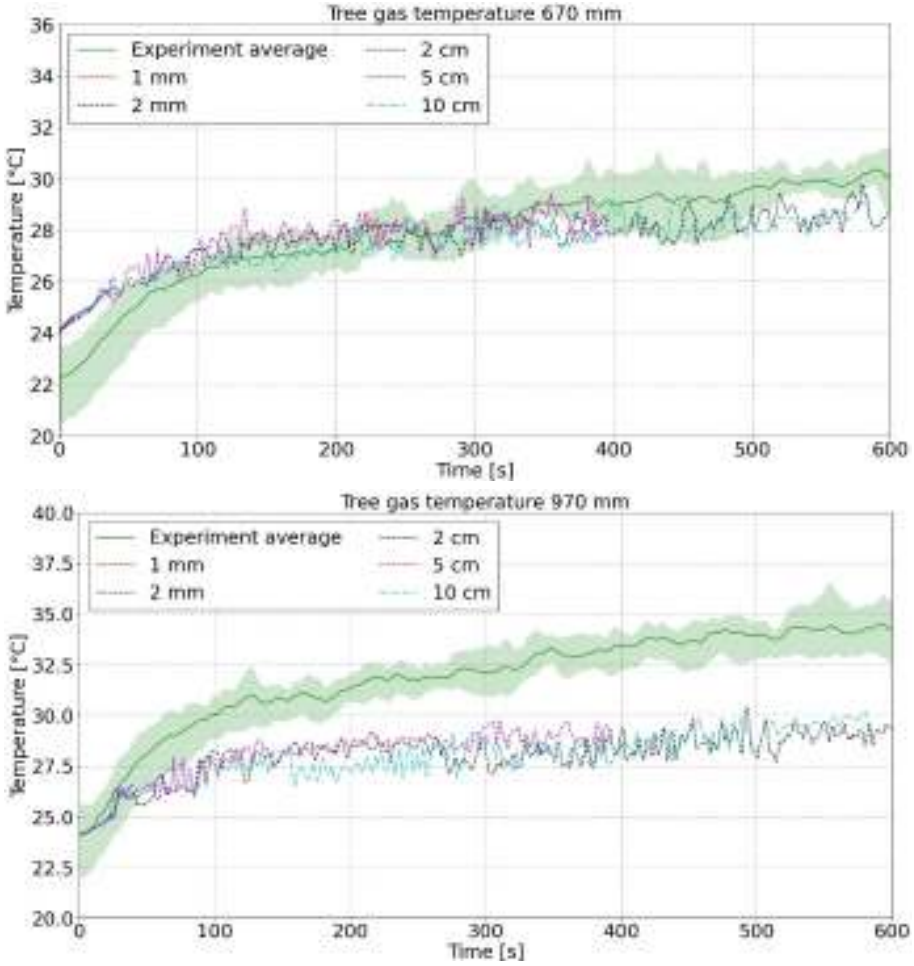

Figure 4: Gas phase temperature in the corner of the test room 670 and 970 mm above floor.

room. Under the ceiling in the center of the room, however, is fit to experimental data very good. In the cold layer at 670 mm the fit is very good, at 970 mm model underpredicts the cold layer temperature about 5 °C. There is no clear trend between various mesh sizes in the cold layer temperatures. In the hot layer, the temperature from the 2 cm mesh simulation is fitting the experimental values in the steady state closer compared to coarser grids.

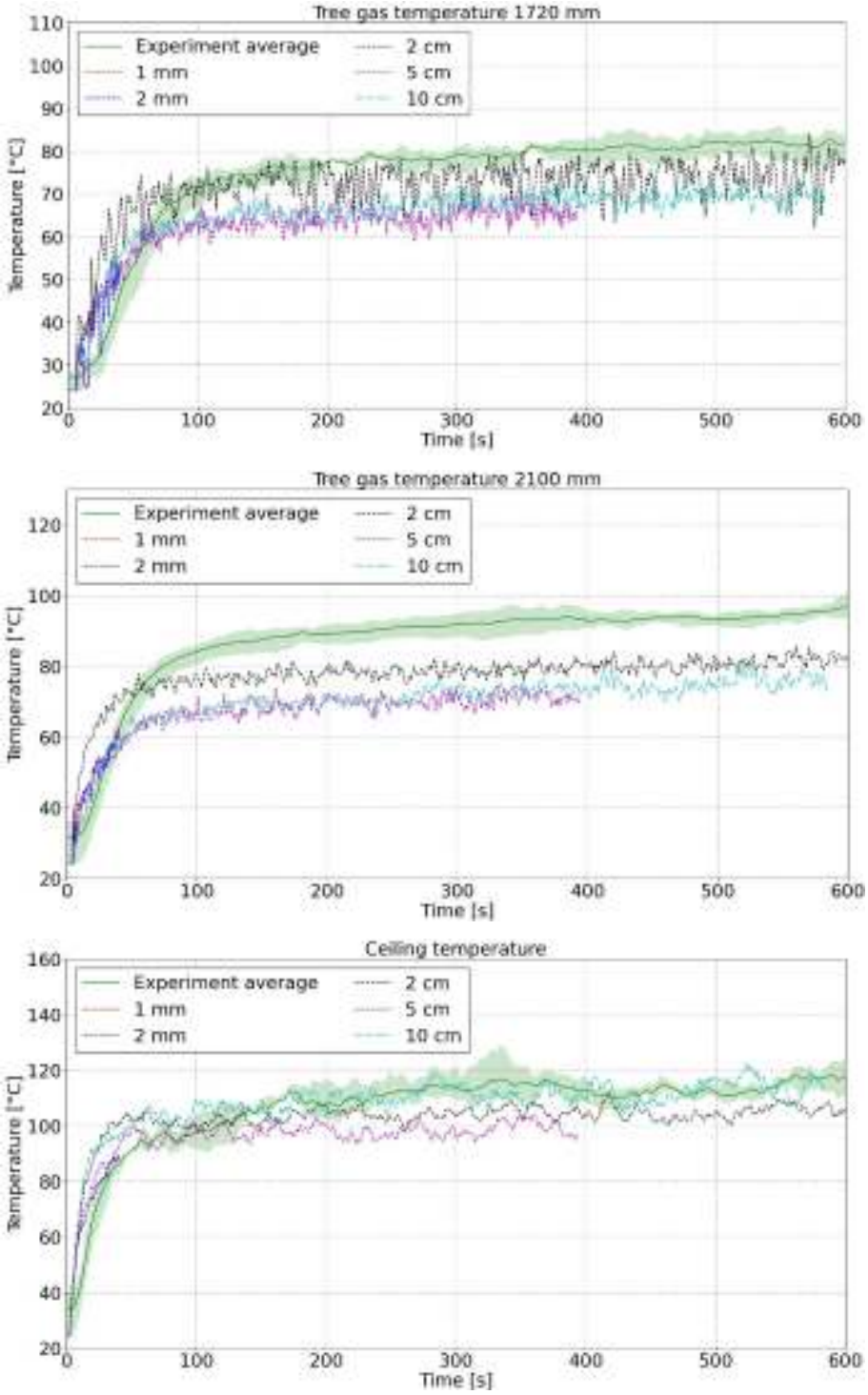

Figure 5: Gas phase temperature in the corner of the test room 1720 and 2100 mm above floor and under the ceiling (2300 mm) in the middle of the test room (above the panel).

Gas phase temperature near the panel surface on the front side 10 cm below the panel upper edge (Fig. 6) is strongly oscillating and, compared to experiment, is the same in the model on the right and left side of the panel. The same applies for net heat flux to the panel (Fig. 7). This is probably caused by the incorrect boundary condition in the hood area. In the model, the area under the hood is considered open to the surroundings from all three sides. In the test facility, however, only left side is open to the surroundings. There is a wall on the right side of the exhaust area and the glass wall to the control room in the front. The air entrainment to the room through the doors is, thus, not uniform from all sides. This results in an asymmetrical flow pattern close to the panel in experiment. In the model, however, the flow is symmetrical around both sides of the panel. The net heat flux values (Fig. 7) are strongly underestimated by the model compared to experimental results on the left side of the panel (position R3 and R1). The larger difference between the model and experiment on the left side of the panel can be explained by the unsymmetrical flow conditions in the experiment. However, the heat flux is slightly underpredicted also on the locations R4 and R2. It is important to notice, that the experimentally measured heat flux is a heat flux to the cold wall compared to the model, where the panel temperature is increasing in time. This can lead to higher values measured in the experiment compared to the model. Also, as a result of the panel increasing its temperature, the net heat flux to the wall in the model slightly decreases with time. It is also important to notice,

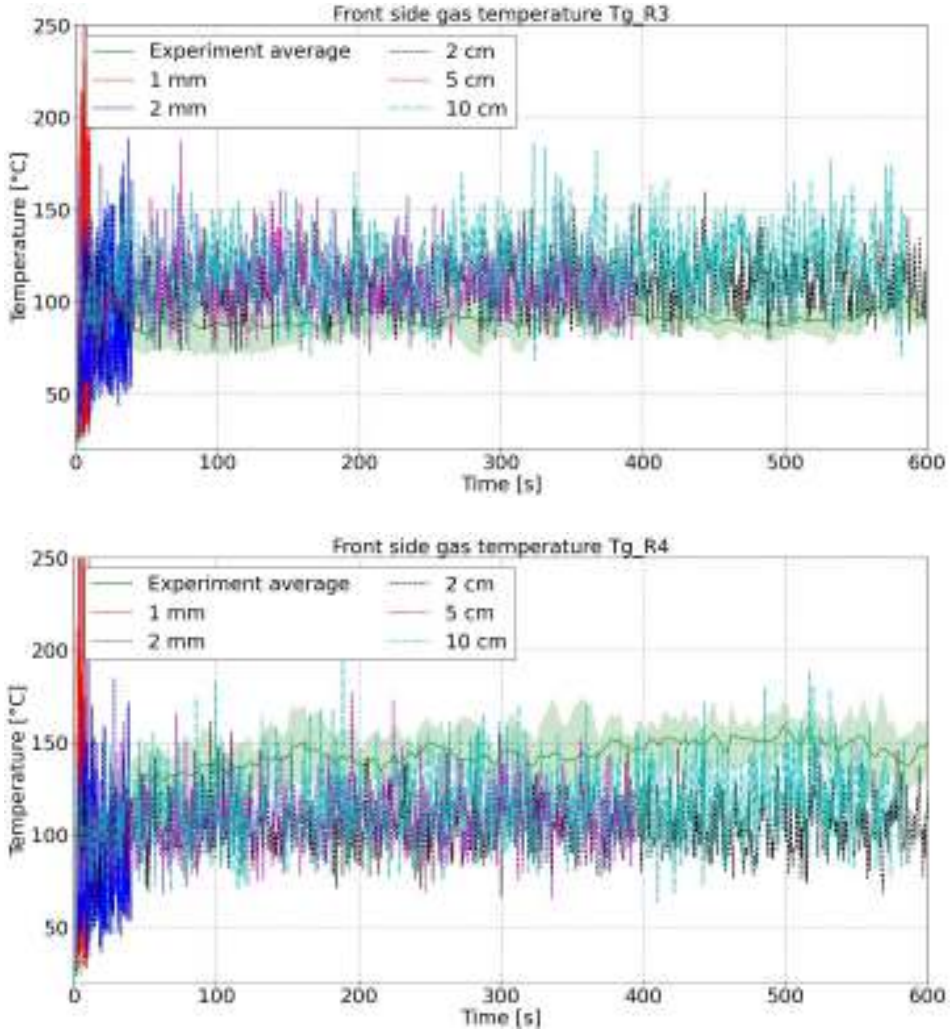

Figure 6: Gas phase temperature near the inert panel at location R3 and R4.

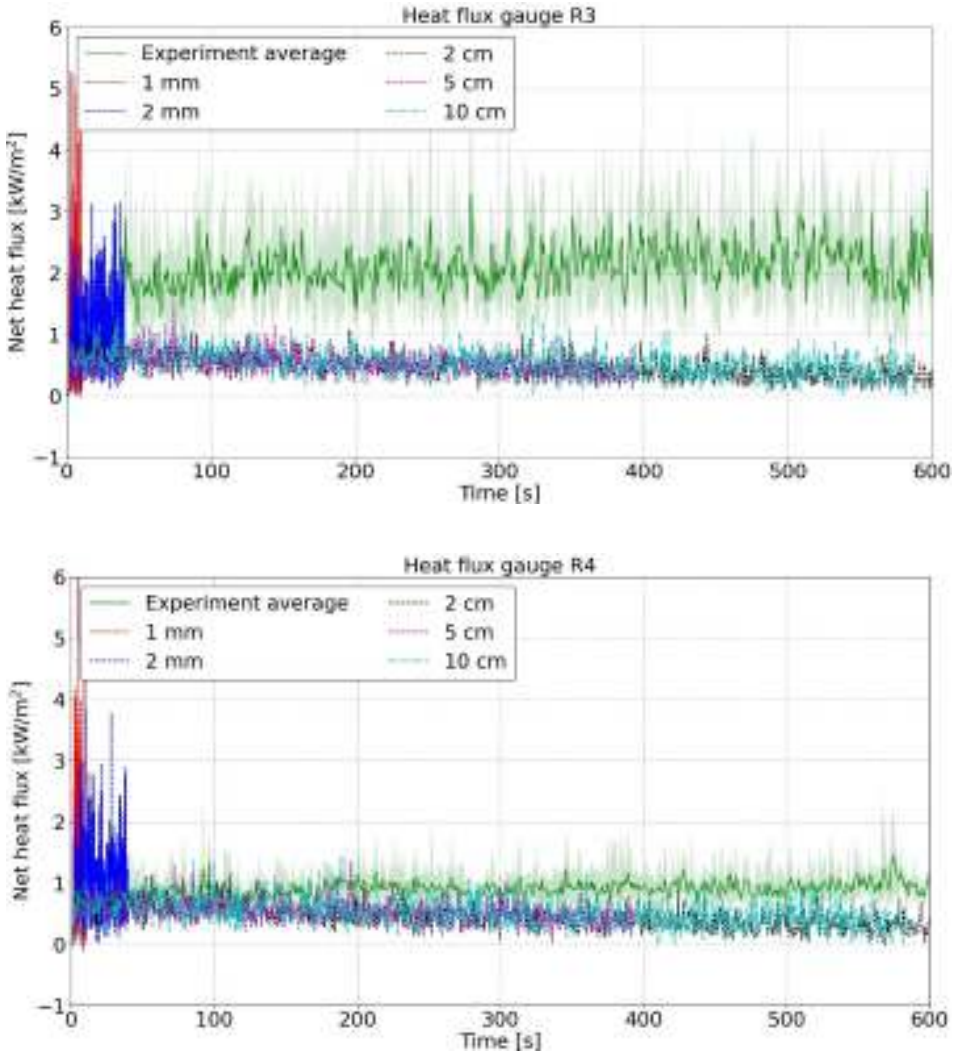

Figure 7: Net heat flux predicted by the model for five different mesh simulations compared to experimental results.

that the experimentally measured net heat flux value does not include the gauge uncertainty as discussed in the "measured quantities" section.

There is no difference between 2, 5 and 10 cm coarse grids. These three meshes, however, had all 1 cm cells in front of the panel. Looking on the first 40 seconds of the simulation, 1 and 2 mm mesh results show slightly higher net heat flux fitting closer to experimental results.

Gas phase temperature close to the back side of the panel along the centerline (Fig. 8) is fitting the experimental data very well at locations T7 and T9. At location T11, there is an increase in the gas temperature in the first 200 seconds predicted by the model but not present in the experiment. This is caused by too coarse grid between the panel upper edge and the ceiling and the large volume ratio of the cells, especially in the case of 2 cm grid. This case has a space above the panel filled with 2 cm grid, compared to 5 and 10 cm grid cases. These two simulations have a 1 cm grid above the panel. As a result, the hot gases in the model are "pushed down" behind the wall before the hot gas layer develops. Although the short simulation time, this behavior seems not to be observed with 2 mm grid, because the mesh refinement in the ceiling area above the panel is less coarse and cells had a smaller volume ratio.

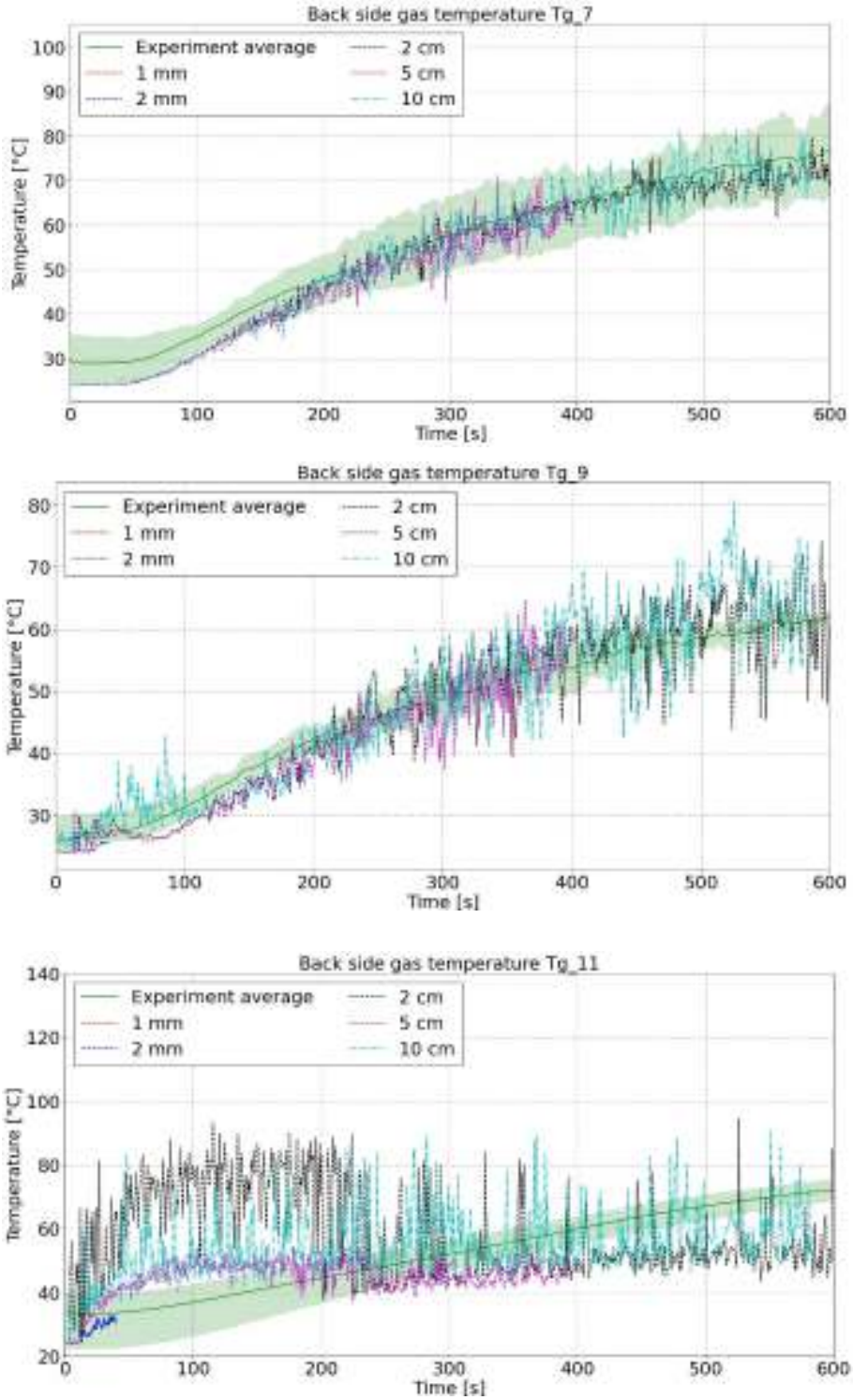

Figure 8: Gas phase temperature close to the back surface of the panel along the centerline at heights 500 (T7), 1000 (T9) and 1500 (T11) mm.

Model results – solid phase

Fig. 9 and Fig. 10 show the front panel temperature at three heights along the centerline and on the right edge of the panel. The model significantly overpredicts the increase of the surface temperature in the region of the flame (Fig. 9, T1) in the first two hundred seconds of the simulation. On the contrary, the gas phase temperatures and the net heat flux predicted by the model are rather underpredicted. Overprediction of the panel temperature can be seen even for the very fine 2 mm and 1mm grid. With this fine resolution, the heat transfer to the wall in the region of the boundary layer should be resolved and shouldn't be the reason for the discrepancy between the model and experiment. One of the possible explanations for the model and experiment discrepancy in terms of solid state temperature evolution is the presence of humidity in the panel. More heat is required to heat up the wall when humidity is present. Although the panel is inert from the point of view of thermal degradation, it contains a natural amount of moisture. In three out of five runs, mass was recorded during the experiment. There was a mass decrease ranging from 500 grams to 150 grams recorded over the whole duration of the test. The humidity was largest with the first experiment in the whole data set as well as with the first experiment on the respective day.

The other possible contribution to this discrepancy is the dependency of the thermal conductivity and specific heat capacity on temperature, which is not accounted for in the model.

Steady state value of the surface temperature at location T1 is, however, the same in the model and experiment. Getting higher along the centerline of the panel moving outside the burner flame region, panel surface temperature gets underpredicted. Besides the panel thermal properties, this may be caused by the 1D heat transfer assumption. Heat is conducted only in the direction perpendicular to the plane of the panel. No heat is conducted upwards or to the sides.

Looking at the front panel temperature 10 cm from the panel edge, at all three heights, the fit of the model to experiment is very poor. The model significantly overpredicts the temperatures. Besides the 1D heat transfer "error" and the thermal properties "error", there is also an influence of the unsymmetrical flow pattern in the experiment that leads to lower heat flux to the wall and lower temperatures in the vicinity of the wall compared to the model. At locations T4 and T6 outside the flame region there is a clear trend between mesh resolutions. The 2 cm mesh provides better fitting results to experimental data compared to 5 and 10 cm mesh resolution.

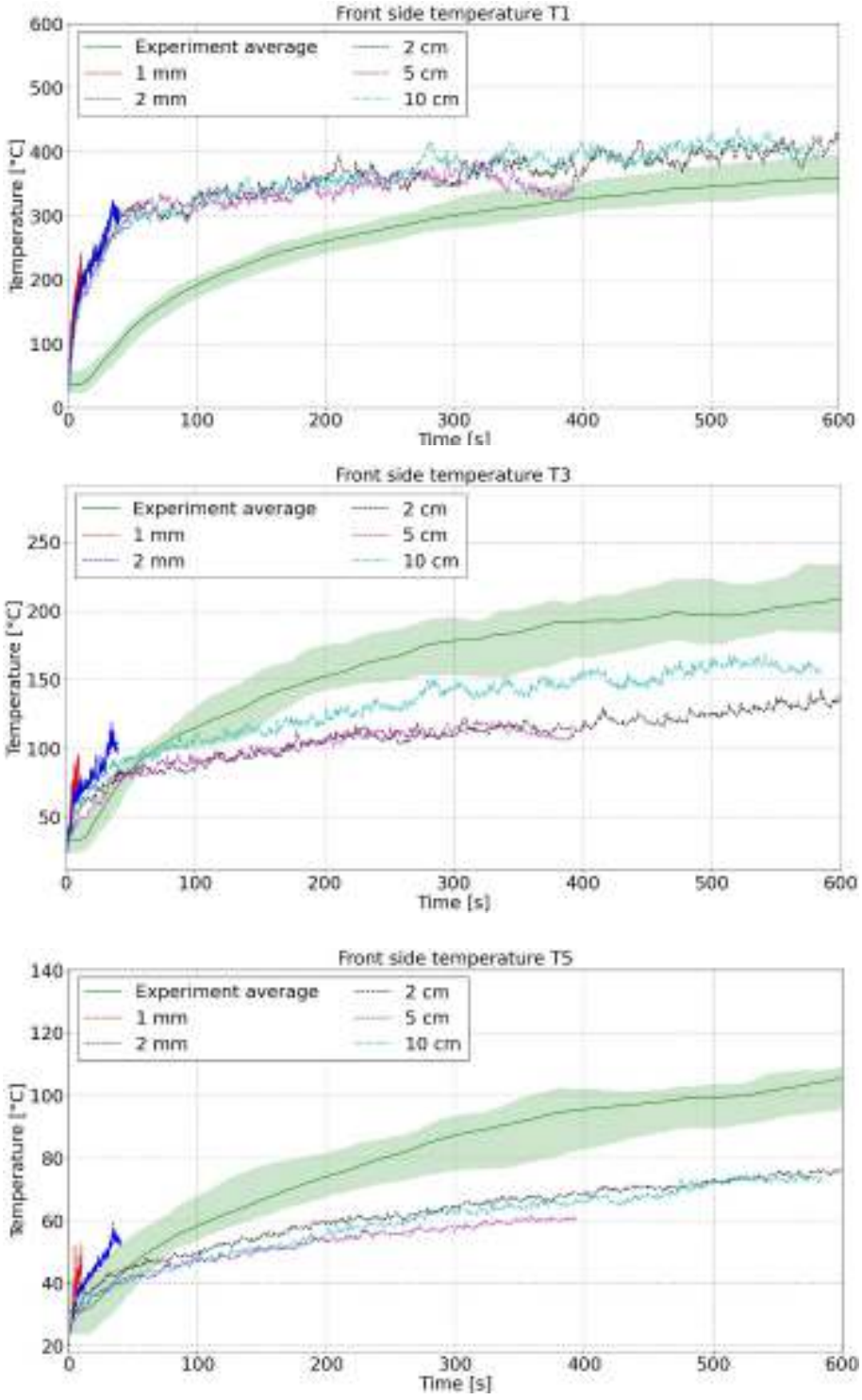

Figure 9: Front side panel wall temperature at 500 (T1), 1000 (T3) and 1500 (T3) mm front panel lower edge along the centerline of the panel.

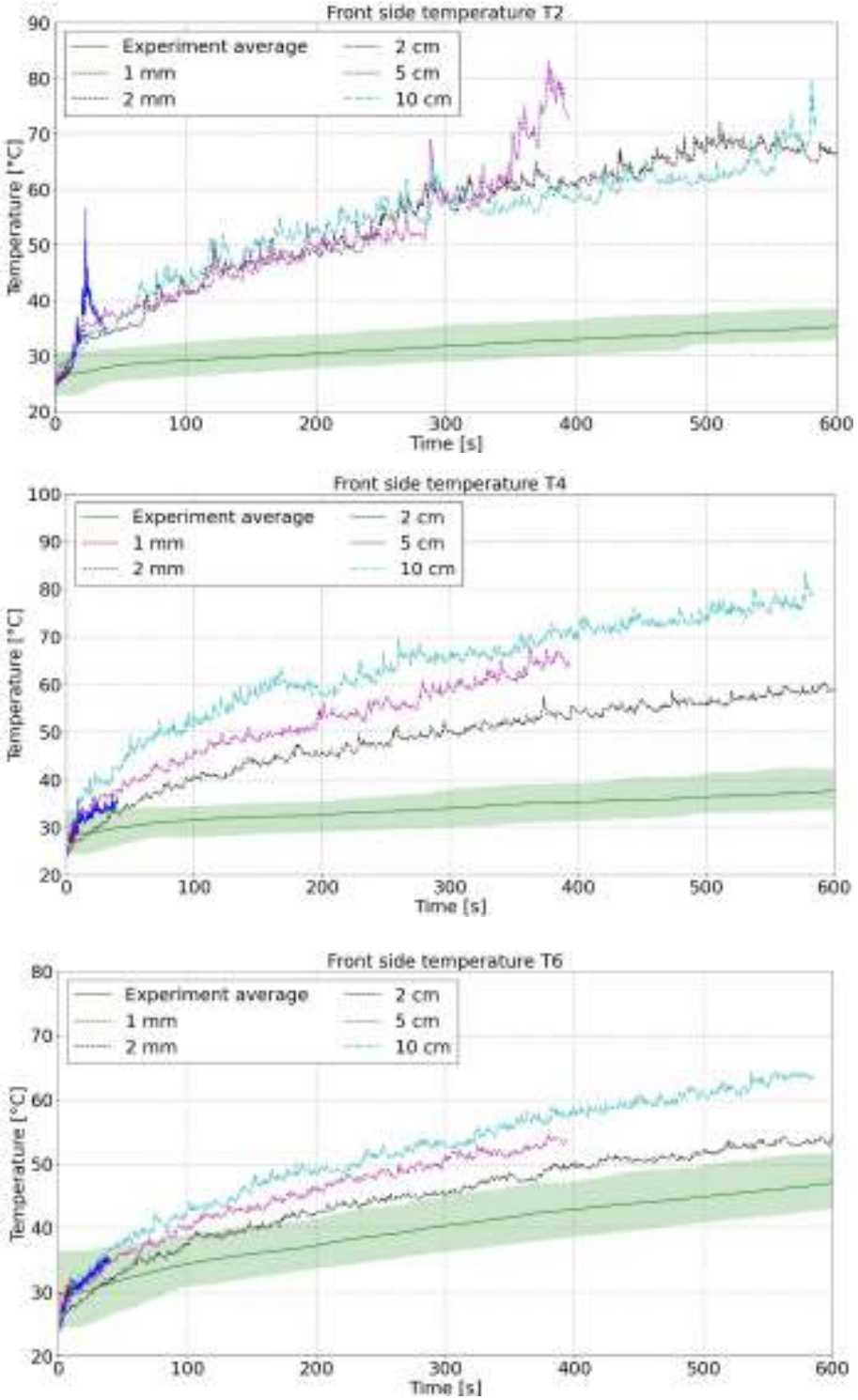

Figure 10: Front side panel wall temperature at 500 (T2), 1000 (T4) and 1500 (T6) mm from panel lower edge on the right side of the panel (10 cm from the right panel edge).

CONCLUSIONS

From the point of view of the experimental data set, five repeats proved to be the minimum number of repeats for the case of the inert panel model to get a representative uncertainty for all the measured values. Measured net heat flux to the wall needs to be carefully reevaluated to incorporate the measuring device uncertainty. In addition to the measured values, the velocity profile in the doors to the test room would be a beneficial variable to be added to the data set.

The coarse 5 and 10 cm mesh in the test room is sufficient to represent the evolution of the neutral plane in the room and the cold and hot gas layer temperatures. 2 cm grid needs to be refined in the region above the panel to 1 cm to fit the 5 and 10 cm grid simulations.

The simplification of assuming the test room walls to be inert has no significant influence on the model prediction in the first 600 seconds of the simulation. Looking at the experimental data, however, there is a slight increase in the gas temperature after 600 seconds. This is probably caused by the walls reradiating the accumulated heat into to space of the test room. If the simulation time should be prolonged, the room walls need to be modeled as layered material with appropriate thermal properties.

Looking at the values close to the panel and the solid temperatures, for further calculations, it is necessary to change the boundary condition in the hood area.

To better understand the influence of the flow conditions adjacent to the wall on the heat transfer to the wall, the 1 and 2 mm grid case needs to reach at least 200 seconds of the simulation time. Velocity profiles adjacent to the wall need to be compared.

It is necessary to add thermal properties dependency of Fermacell on the temperature to the model. Thermal heat conductivity and specific heat capacity need to be measured experimentally up to 400 °C.

A simplified smaller scale test model should be performed using the 3D heat transfer feature of FDS software to evaluate the "loss of heat" through heat conduction upwards and to the sides of the panel. Understanding the influence of the panel properties and the 1D heat transfer assumption will significantly improve the interpretation of the model results.

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