

ABSTRACT

Title of Dissertation: **ESTIMATING THE HEAT TRANSFER
FROM AN ELECTRIC VEHICLE FIRE
USING FIRE DYNAMICS SIMULATOR**

Oliva Izquierdo Hunt, Master of Science, 2026

Thesis Directed by: **Professor Arnaud Trouvé and Adam Barowy
Fire Protection Engineering Department**

In this thesis guidelines for modeling a vehicle using Fire Dynamics Simulator (FDS) and its relevant parameters are explored and documented through an estimation and comparison to experimental data. Vehicle construction consists of determining fire spread and subsequent burner delineation from analysis of the experimental data, of which two geometries evolve. One with fire spread across multiple burners located under the vehicle, on the hood, in the cabin, and at the rear of the vehicle shaped geometry. The second, a single rectangular burner used to represent the burning vehicle. These two geometries are compared with each other and the experimental data to determine model validation.

The selected vehicle for this study was a 2020 electric Hyundai Kona which was experimentally free burned by the Fire Safety Research Institute (FSRI) in February of 2024. The heat flux, mass loss rate, and video evidence from this experiment was used to build a FDS model of the performed experiment. The model is used to create guidelines to simulate vehicle fires, compare the single burner model to the multi-burner model, and compare the simulated lateral

heat flux against experimental data. It is found that the single burner model's lateral heat flux was comparable to the multi-burner model and the experiment.

ESTIMATING THE HEAT TRANSFER FROM AN ELECTRIC VEHICLE
FIRE USING FIRE DYNAMICS SIMULATOR

by

Oliva Izquierdo Hunt

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Advisory Committee:

Professor Arnaud Trouvé, Chair/Advisor
Mr. Adam Barowy, UL FSRI
Dr. Jonathan Hodges, UL FSRI
Dr. Ali Tohidi

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Dedication

To my Father Dr. Alan Hunt whose patience has been unwavering since he taught me how to ski at 2 years old.

Acknowledgments

Much to my dismay it is nearly impossible to thank everyone who accompanied me in the successful completion of this thesis.

First and foremost, this work would not have been possible without the support of the Fire Safety Research Institute and their fellowship program. Adam Barowy and Nate Sauer for mentoring me through the process of research and challenging me to problem solve from every angle. Special thanks to Jonathan Hodges for coming onto the project halfway into the process, thank you for the expertise and fervor you brought to our meetings. I would also like to thank my advisor and committee member from the University of Maryland Dr. Trouvé for showing me the ropes to fire modeling and Dr. Ali Tohidi for graciously accepting to be a committee member.

On a more personal note, I would like to thank my mother whose unwavering confidence in me is astounding. To my family in the US who taught me the world is really just one big physics problem and always willing to geek out about research. To my family in Spain who never fails to make me laugh or figure out the best way to translate ‘fire protection engineering’.

Thank you to my best friends for always being my greatest cheerleaders, and best conversationalists, whom I fondly refer to as “The Marias” (yes plural). And to my fire protection engineering friends who supported me through undergraduate and graduate school. Lastly, I would like to thank the Catholic Student Center for showing me that life is a victory lap to the greatest sacrifice made for mankind.

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List of Abbreviations

CFD	Computational Fluid Dynamics
EPUMO ₂	Energy per Unit Mass Oxygen
EV	Electric Vehicle
FDS	Fire Dynamics Simulator
FSRI	Fire Safety Research Institute
HRR	Heat Release Rate
ICEV	Internal Combustion Engine Vehicle
NIST	National Institute of Standards and Technology
RMSE	Root Mean Square Error

Chapter 1: Introduction

The global transition toward electric mobility has accelerated dramatically in recent years. Electric vehicle (EV) sales surged from approximately 2.2 million units in 2019 to 13.7 million in 2023, representing nearly one in five vehicle sold worldwide [1]. This rapid adoption, driven by environmental imperatives and government incentives, has introduced new fire safety challenges that differ from those associated with internal combustion engine vehicles (ICEVs). Lithium ion batteries present unique hazards including thermal runaway, jetting flames, prolonged burning durations, and re-ignition risks. These characteristics complicate suppression efforts and raise concerns about fire spread in confined environments such as parking garages, tunnels, and residential structures.

Despite these concerns, full scale EV fire experiments remain limited due to their high cost, logistical complexity, and proprietary restrictions from manufacturers. Comparable data for modern ICEVs is similarly scarce, requiring fire protection engineers to use alternative methods to resolve vehicle fire heat transfer for safety analyses. Computational fluid dynamics (CFD) modeling offers a practical solution to bridge the gap between limited EV fire experiments and the insight they provide on heat transfer from a flaming vehicle. Computer modeling has long been used to simulate fire scenarios where full-scale experiments may be infeasible due to cost, time and other practical challenges. This practice is often employed by fire protection engineers

to provide design recommendations for unique events, particularly involving thermal analysis and heat flux to a target. More specifically to the EV concerns is the analysis of vehicle-to-vehicle fire spread, vehicle-to-structure damage, and human tenability during egress and emergency response. Fire Dynamics Simulator (FDS) is a general purpose CFD program which is designed to estimate and model these concerns in any environment, such as parking garages, tunnels, or outdoors. However, existing literature on vehicle fire modeling suffers from two major shortcomings. First, many designers base their analyses on data from older ICEV experiments which may not be representative of the hazard posed by modern EV fires. Second, there is often little documented rationale regarding model input parameter choices, making replication and adaptation by other modelers difficult.

The absence of clear methodologies for EV fire modeling, combined with the growing need to evaluate vehicle heat transfer, highlights a gap in current practice. This thesis addresses these deficiencies by developing guidelines for modeling EV fires using FDS and PyroSim to estimate vehicle heat flux to a target based on experimental data. The proposed methodology emphasizes reproducibility and clarity by providing detailed guidance on parameter selection and its influence on lateral heat flux. To ensure technical relevance, the simulation is created using full scale experimental data from the Fire Safety Research Institute (FSRI), specifically focusing on a free burn experiment of a Hyundai Kona SEL 2020 EV [2]. This process enables a direct comparison between simulated and experimental results, establishing confidence in the model's predictive capability.

The research objectives are fourfold:

1. Develop a structured approach for EV fire modeling in FDS that includes geometry con-

struction, burner placement, material assignment, and combustion parameter definition.

2. Evaluate the impact of modeling complexity on predictive accuracy by comparing a multi-burner vehicle model with a simplified single-burner fire representation.
3. Assess the simulation outputs against experimental heat flux measurements.
4. Identify best practices for achieving agreement while balancing computational efficiency.

Addressing these objectives, provides actionable guidance for fire protection engineers tasked with assessing EV fire hazards in performance-based design contexts. The methodology enables construction of models that are both technically sound and adaptable to diverse scenarios, including those where full scale testing is impractical. Furthermore, the comparative analysis of single and multi-burner approaches informs decisions about model complexity, offering insights into when simplification is acceptable and when detailed geometry is necessary; particularly for scenarios involving ceiling impingement or structural heat exposure.

Ultimately, this research aids in bridging the gap between experimental findings and practical modeling applications. This work enhances understanding of EV fire behavior, promotes the development of modeling guidelines, and establishes a foundation for future studies involving different vehicle types, alternative fuels, vehicle heat transfer, and complex built environments. By evaluating a modeled EV fire through the comparison of experimental measurements, this thesis contributes to improved fire safety strategies for infrastructure where electric vehicles are prevalent.

Chapter 2: Literature Review

2.1 Growth of EV Market

Internal combustion engine vehicles have been the standard since Nikolaus Otto's invention of the internal combustion engine in 1876, and the evolution of regular long distance travel after World War I [3]. However, heightened environmental concerns in the 1990's and 21st century led to the mass production of electric vehicles whose purpose was to reduce carbon emissions [4]. Electric vehicle popularity and sales rapidly increased in the 2020's due to environmental benefits, government subsidization, tax incentives, and consumer consciousness. In 2019 the number of electric vehicles sold worldwide was about 2.2 million, the value reported for worldwide electric vehicle sales in 2024 skyrocketed to over 17 million, accounting for nearly one in four vehicles sold [1, 5]. EV sales in the United States slowed in 2024 increasing by 10% compared to 40% in 2023, nevertheless these EV sales boosted the vehicle market as ICEV sales stagnated. In 2025 electric vehicle sales are expected to increase 25% worldwide with sale shares doubling in emerging markets such as Asia, Latin America, and Africa [5].

2.2 Comparison of ICEV and EV Heat Release Rate

Differences between power train technology from internal combustion engines to lithium-ion battery vehicles, the rapid increase in electric vehicle sales, and initial challenges managing EV fire incidents have led to uncertainty regarding their safety [5, 6]. News reports often allege increased fire risk with electric powered vehicles attributable to lithium-ion battery thermal runaway and burning behavior. These concerns include battery induced ignition, faster fire growth, larger and longer lasting fires, jetting flames, re-ignition from minutes to months later, fire suppression difficulty, toxic gas and particulate emissions, and contaminated suppression water runoff [7, 8].

Limited full-scale EV fire experiments have been conducted to examine these hazards due to significant costs, few laboratories have adequate capacity for the full scale testing, and manufacturers that have conducted experiments desire to protect proprietary information [9]. Comparable data assessing ICEV fire behavior is similarly limited [10]. Newer ICEVs merit further investigation of fire behavior due to a higher load of combustible materials than older vehicles [9]. This holds true for both ICEVs and EVs as the percent plastic by mass used in vehicle construction has increased from 6% in the 1970's to an expected 18% in 2020 [11]. The Fire Protection Research Foundation's assessment on changes to vehicle composition found that increased combustible load due to plastics had relatively insignificant effect on the heat release rate (HRR). Nevertheless, increased plastic use was identified as a contributing factor in vehicle-to-vehicle fire spread, potentially due to the replacement of metal components with plastic components [6].

The following studies focus on the comparison between EVs and ICEVs through large-scale testing of vehicles. EVs and ICEVs are often contrasted, with the expectation that an EV

burns significantly more intensely than the other. A French study by Lecocq tested two small passenger EVs with two ICEVs of comparable size and power. This series of tests found that the peak heat release rate, heat of combustion, and effective heat of combustion were all relatively similar between both kinds of vehicles [12]. Another study by Hynynen et al. corroborates this finding as both the ICEVs and EVs had similar peaks in HRR with the ICEV peak at about 5,750 kW and the two EVs peak at 7,000 kW and 5,250 kW respectively [13]. These experiments demonstrate significant similarities between ICEVs and EVs.

Despite these similarities, the recent attention on modern vehicles has highlighted that historical guidance on passenger vehicle fire size may be an underestimate for current vehicles on the market. This has caused fire protection engineers to revisit assumptions on the likelihood of a single vehicle fire transitioning to a multi-vehicle fire.

2.3 Fire Spread Between Vehicles

Fire spread between vehicles is related to the fire size of the initial vehicle, separation between the vehicles, and the material properties of the target vehicle. While limited data exists on fire spread from older vehicles, flame spread between modern vehicles has not been well characterized to date. However, these dangers can be evaluated through the use of hand calculations with models of a flame shape and view factors or with detailed computational fluid dynamics modeling. Either method can be used to provide a technically sound assessment of fire risks with well developed inputs for a specific fire scenario. However, view factor methods are generally limited to open burning scenarios where the fire behavior is not dependent on local interactions with ceilings and walls.

Although the view factor method has limitations, it is necessary to understand its function in practice and how recent studies have assessed its reliability. The view factor method is used to estimate radiation heat transfer from the burning vehicle. The determination of the view factor between the flame and surrounding objects arises from the analysis of the heat release rate, heat flux, and the radiative transfer to potential targets [14]. Assumptions regarding the flame shape, object shape, and orientation allow for the final view factor to be implemented. Hu et al.'s results were compared to measured values of two heat fluxes in the lateral direction and found to be within a 20% deviation, therefore in good agreement according to the author [14]. The work of Gavilán and Gong demonstrates a conically shaped view factor as opposed to a rectangular shape used in Hu et al [7, 14]. Gavilán and Gong also detailed the challenges in specifying model inputs for EV fires [7]. Additionally, Gavilán and Gong's study used video analysis to measure flame height and width [7]. To simulate flame spread multiple fire curves were allocated to different portions of the vehicle throughout the duration of the burn, serving as input for the measuring devices [7].

Halada et al. used the CFD method and successfully modeled an ICEV fire spread with an air flow based on an experimental set up [15]. There is also a discussion regarding flame spread within the vehicle, however, it was contained to the cabin or the hood rather than spreading between the two. Of these three studies conducted, only the one by Gavilán & Gong was conducted using an EV [7]. Other studies specific to EVs such as Brzezinska & Bryant, Wang et al., and Barré et al. discuss EV fires but only within a structure or unique scenario, as opposed to free burning [16, 17, 18]. Nevertheless, the process in determining view factor and building models within Fire Dynamics Simulator (FDS) is similar amongst these four studies.

2.4 Research Gaps

The rapidly changing field of lithium-ion batteries has led to few methodologies regarding the simulation of EV fires. Although the previously mentioned modeling efforts advanced the topic, further explanation regarding the model inputs is necessary to make widespread use of a modeled vehicle fire. An example of a study where both species inputs and model parameters are clearly listed is a recent paper by Voigt et al. [8]. Species included are soot yield, thermal conductivity of the battery, specific heat, and model velocity. The model parameters include time, HRR, ignition temperature, and onset temperature of the reaction, as well as some mesh refinement details. Although the study aimed to predict the hazards induced by heat and gas release during a battery fire, it lacked greater explanation as to why those parameters were chosen and in large part focused on the lithium-ion battery, as opposed to the entire vehicle.

Compared to a view factor analysis, CFD allows for more expansive examination of fire hazards in various design scenarios since its calculations include plume characteristics, combustion product dispersion, and heat effects to adjacent objects [19]. Thus, by modeling a specific EV with good agreement to experimental data, the model can then be implemented in untested environments. Testing a large spectrum of environments would be near impossible to implement with large-scale testing due to the previously mentioned constraints. The application of EV modeling in various scenarios allows for a broader spectrum of users to access fire testing results, data, and conclusions.

The uncertainty regarding EV fire behavior has driven concerns that EVs pose an increased fire risk in comparison to ICEVs. In the absence of experimental testing in scenarios such as parking lots, parking garages, tunnels, etc. a CFD simulation of a full-scale EV free burn can be used

to enable fire risk analyses in various scenarios. This thesis develops a framework to characterize an individual EV fire with flame spread throughout the vehicle with inputs and model parameters based on the Fire Safety Research Institute's (FSRI) full-scale EV free burn experiments [2]. This model will be used to demonstrate a successful comparison to large-scale test data, thus allowing the model to be used in untested scenarios or the application of this methodology to model other vehicle fires.

Chapter 3: Experimental Setup and Procedure

3.1 Laboratory Setup

The large-scale tests completed by FSRI were conducted in an enclosed fire testing facility and focused on electric vehicle burning characteristics [2]. The space included an exhaust rate of $28.3 \text{ m}^3 \cdot \text{s}^{-1}$. Using figure 3.1 as a guide, the vehicle was situated on a large metal tray to catch drippage during burning while sitting on 4 mass balances, each placed underneath one of the wheels. Each mass balance was a Mettler-Toledo model PFD779 US11 with a IND570 indicator with a capacity of 1500 kg, resolution of 100 g, and repeatability of 90 g [2]. Type-K thermocouples were located within the centerline of the vehicle cabin at passenger head height in the trunk, and front and rear seats. The hood thermocouple was placed in the middle of the hood from either direction directly beneath the hood surface, inside the engine compartment. Steel panels painted black are centered on the vehicle and used to measure a two dimensional heat flux field [20, 21]. These panels have dimensions of 1.83 m by 3.3 m with 4 Medtherm 64-series Schmidt-Boelter water-cooled heat flux gauges spaced 1.04 m apart and 1.3 m above the lab's ground. The method of ignition was a nominally 30 kW propane diffusion burner that was 7.6 cm tall, with a square opening where each side length was 30.5 cm. The burner was placed beneath the location of the vehicle's battery. Once battery thermal runaway was confirmed, the burner was removed. One camera was placed at each corner of the vehicle to observe the

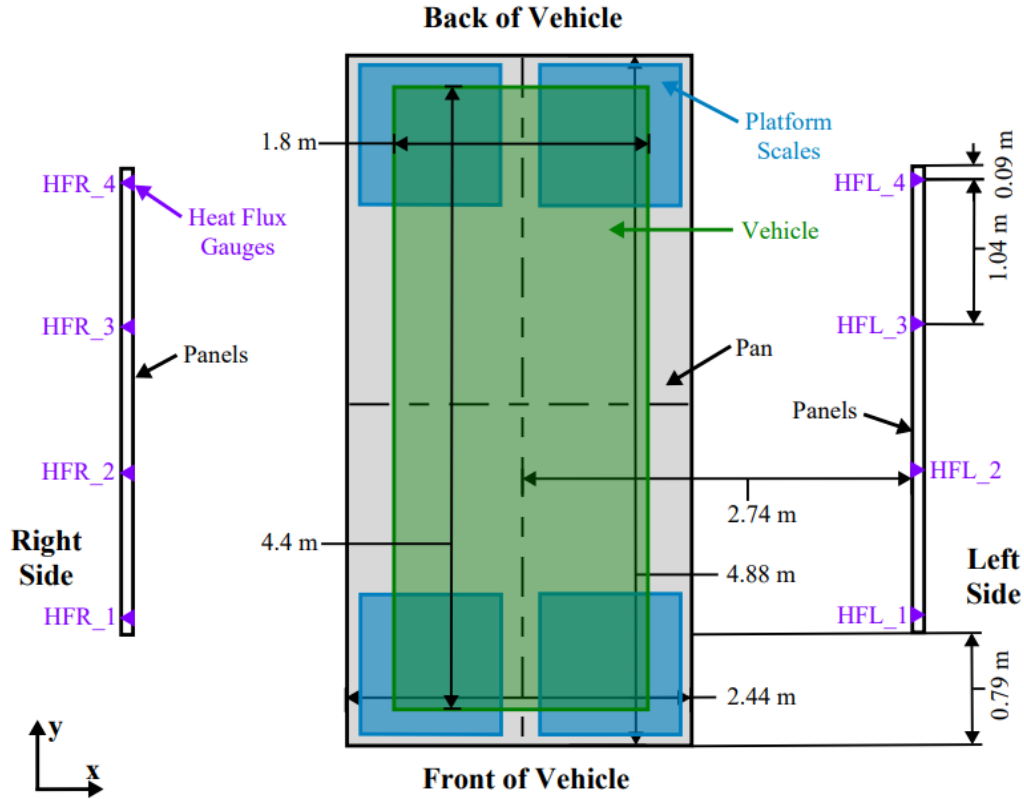


Figure 3.1: The top down view of the experimental setup.

burning behavior throughout the experiment. Sound was incorporated into the videos to gather information on the burning lithium-ion battery and its connections to jetting, explosions, and thermal runaway. A picture of the set up is seen in figure 3.2 and figure 3.2. A photograph of the heat flux gauge placement relative to the panels is shown in figure 3.3, how the gauges were mounted is in figure 3.4.

3.2 Vehicle Selection

This experimental setup is the basis for the model procedure as the model aims to replicate the experimental findings. Therefore, of the 7 vehicles that were tested, two were considered to



Figure 3.2: A front view of the experiment showing the vehicle and panels.



Figure 3.3: The red circles indicate the heat flux gauge placement on the panels.

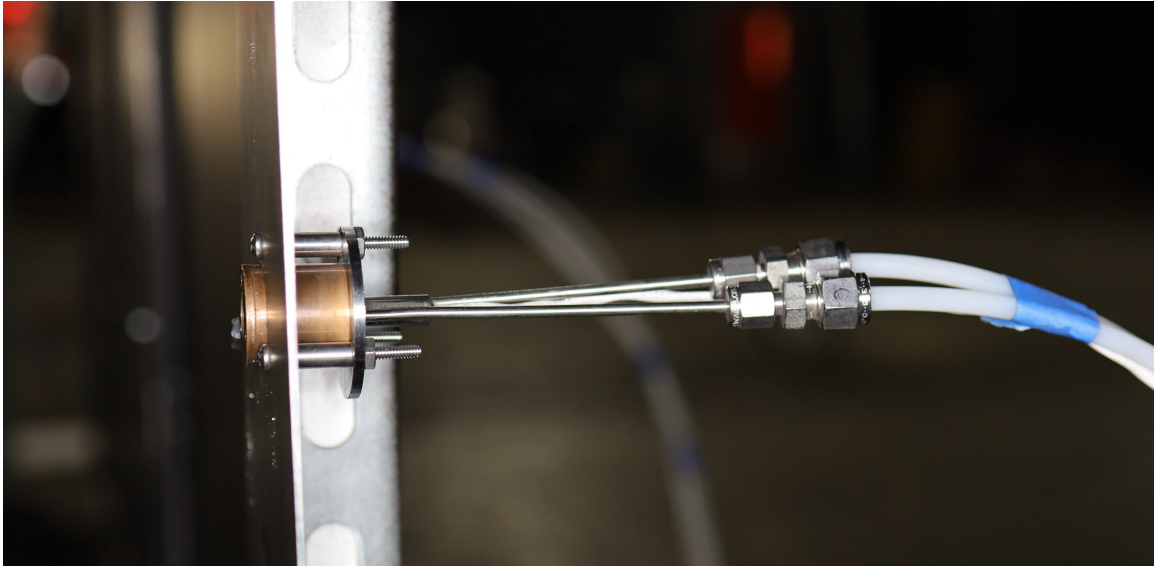


Figure 3.4: Heat flux gauge mounting between panels, demonstrates how flush the gauges is to the panel surface.

be taken into the process of modeling, table 3.1 lists the 7 vehicles tested and the reason why they were not selected. For more details about the vehicle experiments themselves, Sauer et al. provides that information [2]. Considerations that were taken were the consistency of the mass loss rate curve, unusual battery behavior, jetting, and vehicle repeatability. Some vehicles experienced the battery melting out of the vehicle into the drip pan, which rendered it less practical for modeling purposes. Other vehicles were extinguished prior to burn completion resulting in a truncated data set. Many of the vehicles experienced jetting, which although unique to the problem of electric vehicle modeling is outside the scope of this thesis. The final two vehicles considered were the Chevrolet Bolt and the Hyundai Kona. The Bolt offered a very steady mass loss rate but had jetting from the battery occurring in the back half of the vehicle. The Kona had no jetting with a mass loss rate curve clearly attributable to various sections burning. Since the Kona was ignited beneath the “engine compartment” of the vehicle the burning progression was easier to track due to the fire development throughout the vehicle facilitating modeling.

Table 3.1: Table summarizing vehicles that underwent free burn experimentation. All vehicles except G-HK-F were battery EVs.

Experiment ID	Vehicle Make & Model	Decision Justification
G-HK-F	Hyundai Kona SEL	Was a ICEV
E-CB-F	Chevy Bolt EV LT	Jetting from battery
E-NL-F	Nissan Leaf S	Jetting from battery
E-HI-F	Hyundai Ioniq Limited	Less delineated burn
E-TM-F	Tesla Model 3 Long Range	Terminated before full burn
E-FM-F	Ford Mustang Mach E GT	Battery melted out
E-HK-F	Hyundai Kona SEL	Chosen

The Kona and Chevrolet are both popular and representative of a common consumer's electric vehicle. Additionally, the Kona was ranked highly by Car and Driver in the Best Electric Subcompact Sports Utility Vehicle category, particularly for its quality to price ratio, as well as being one of the best sellers in 2021 [22, 23]. Indicating it as a popular choice among consumers and representative of common electric vehicles particularly for the make of the Kona being from 2020. For these reasons the Hyundai Kona SEL 2020 electric vehicle burn was chosen to create an FDS validation model.

3.3 Vehicle Geometry

Preliminary steps to modeling were required for proper implementation of the vehicle into FDS. The dimensions of the vehicle were found [24], and the schematic shown in figure 3.5 was used to determine additional vehicle dimensions [25]. The vehicles primary dimensions are a length of 4.2 m, a width of 1.8 m (not including rear view mirrors), and a height of 1.6 m, all of

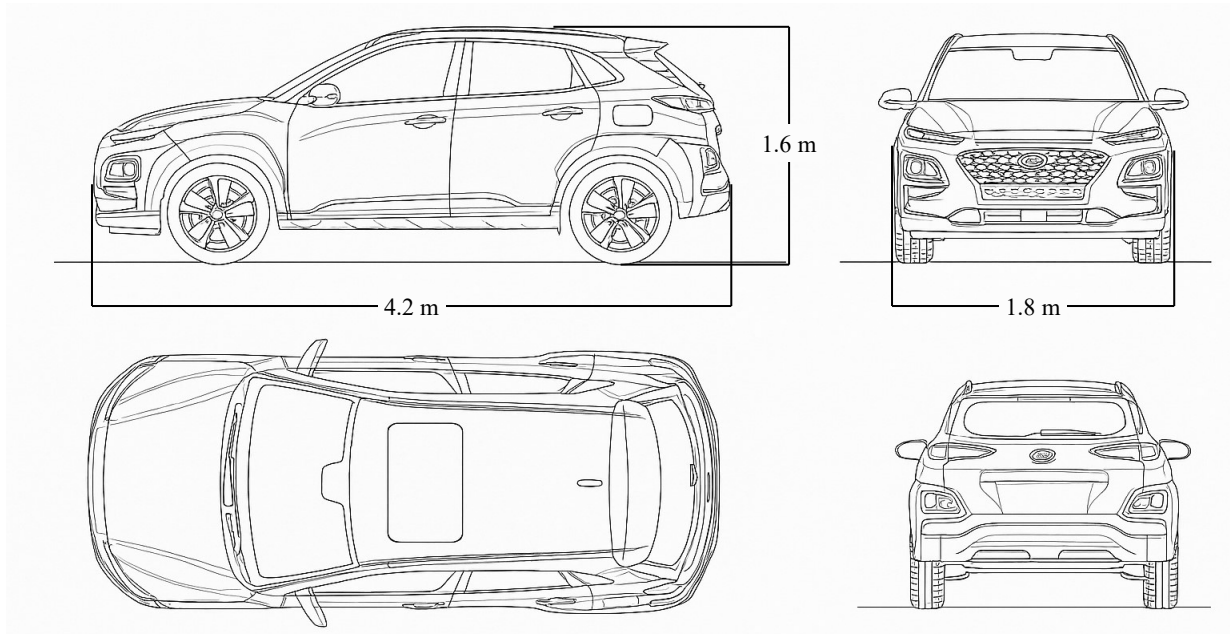


Figure 3.5: Scaled visualization of vehicle used to build model with accurate and proportional dimensions [25].

which are sketched in figure 3.5. The vehicle was a 4 door hatch back with rolled down windows and a fully charged lithium-ion battery (100% state of charge). The battery had a capacity of 64 kWh, and a voltage of 356 V. Within the vehicle the battery was located under the floor, in the mid-rear section. The interior of the vehicle had two front seats and a rear bench seat with a capacity to hold 5 people in total. Four air bag systems were within the vehicle, two air bags are facing the driver and passenger with an additional side air bag deploying on the left side of the driver and right side of the passenger. A large curtain air bag spanning the upper left and right side of the vehicle is also present particularly for the rear seat passengers. These air bags were undeployed at the beginning of the experiment but deployed during the burn.

3.4 Experimental Fire Spread

Experimental video evidence was used to categorize how the fire spread throughout the vehicle. The burning of the vehicle could be classified into two fundamental levels of complexity.

1. Taking the vehicle as simply "on fire," where the vehicle would be a single burner.
2. Splitting the burning areas into major vehicle components where multiple burners would be present.

The levels of burning complexity are later used in chapter 4 with the assumption that multi-burner combustion would improve agreement amongst experimental heat flux measurements. Through observation of the video evidence, portions of the vehicle were delineated regarding when they began burning, figure 3.6 shows the experimental burn at these times. Then a comparison was made to the mass loss rate graph to ensure agreement between a new portion of the vehicle burning to an increased mass loss rate. To corroborate between the video and mass loss rate, the heat flux and thermocouple data were compared and contrasted. The increase in the heat flux and thermocouple data matched increases in the mass loss rate data, as well as visual indication from the video that an additional portion of the vehicle was burning. The spread was ultimately defined as starting under the vehicle, spreading to the hood, and subsequently the cabin or body of the vehicle, and lastly the trunk of the vehicle burns. Figure 3.6 shows the progression of vehicle burning at the times where burning delineation was defined.



Figure 3.6: Still images of the experimental free burn at ignition onset of each major component. Image (1) is when burning under the vehicle begins, image (2) is hood ignition followed by image (3) cabin burning, lastly image (4) with the rear portion of the vehicle igniting. Each quadrante of images displays one section of burning from four distinct angles with the corresponding time in minutes located at the center of the quadrant.

3.5 Experimental Data

3.5.1 Mass Loss

Figure 3.7a-c, and 3.8 are experimental data measured by the setup detailed in section 3.1.

The data presented is not an exhaustive representation of the data gathered by FSRI's free burn vehicle experimental series. However, the presented data is most relevant to the simulation outputs and measurements needed for performance-based design objectives. These objectives are “(1) agreed upon fire safety goals and objectives, (2) deterministic and/or probabilistic analysis of fire scenarios, and (3) quantitative assessment of design alternatives against the fire safety goals

and objectives using accepted engineering tools, methodologies, and performance criteria” [26]. These criteria are based on engineering and scientific judgment from experimental data. Particularly, points (1) and (3) where the goals may be evaluated on a deterministic measurement from data and the quantitative assessments often use modeling approaches in conjunction to experimental data to determine the efficiency or failure of a design. Hence, discussing the experimental data and ultimately its comparison to the simulation is of great value.

Vehicle mass loss was taken by adding the recorded mass of each scale placed beneath the vehicle. Figure 3.7a shows the vehicles mass decreasing as a function of time due to combustion. Ultimately, the vehicle lost 15.6 % of its total mass, which is equivalent to 153.8 kg. More commonly seen and used is the mass loss rate, figure 3.7b, which is the derivative of figure 3.7a times negative one. Found in the form of the finite difference method where the difference in mass loss is divided by a selected time delta. The mass loss rate is the more commonly used measurement as it is used to determine the heat release rate of a fire. The heat release rate is taken as the mass loss rate multiplied by the heat of combustion of the vehicle, where the heat of combustion is $25 \text{ MJ}\cdot\text{kg}^{-1}$ [2, 9, 10]. As a result, the heat release rate and the mass loss rate have identical shapes. However, heat release rate is used as a measure of fire size and is often used as a criterion or goal to evaluate a performance-based design. In figure 3.7c, the maximum heat release rate seen to be approximately 5 MW, this value is an example of a commonly used performance-based design input or evaluation metric.

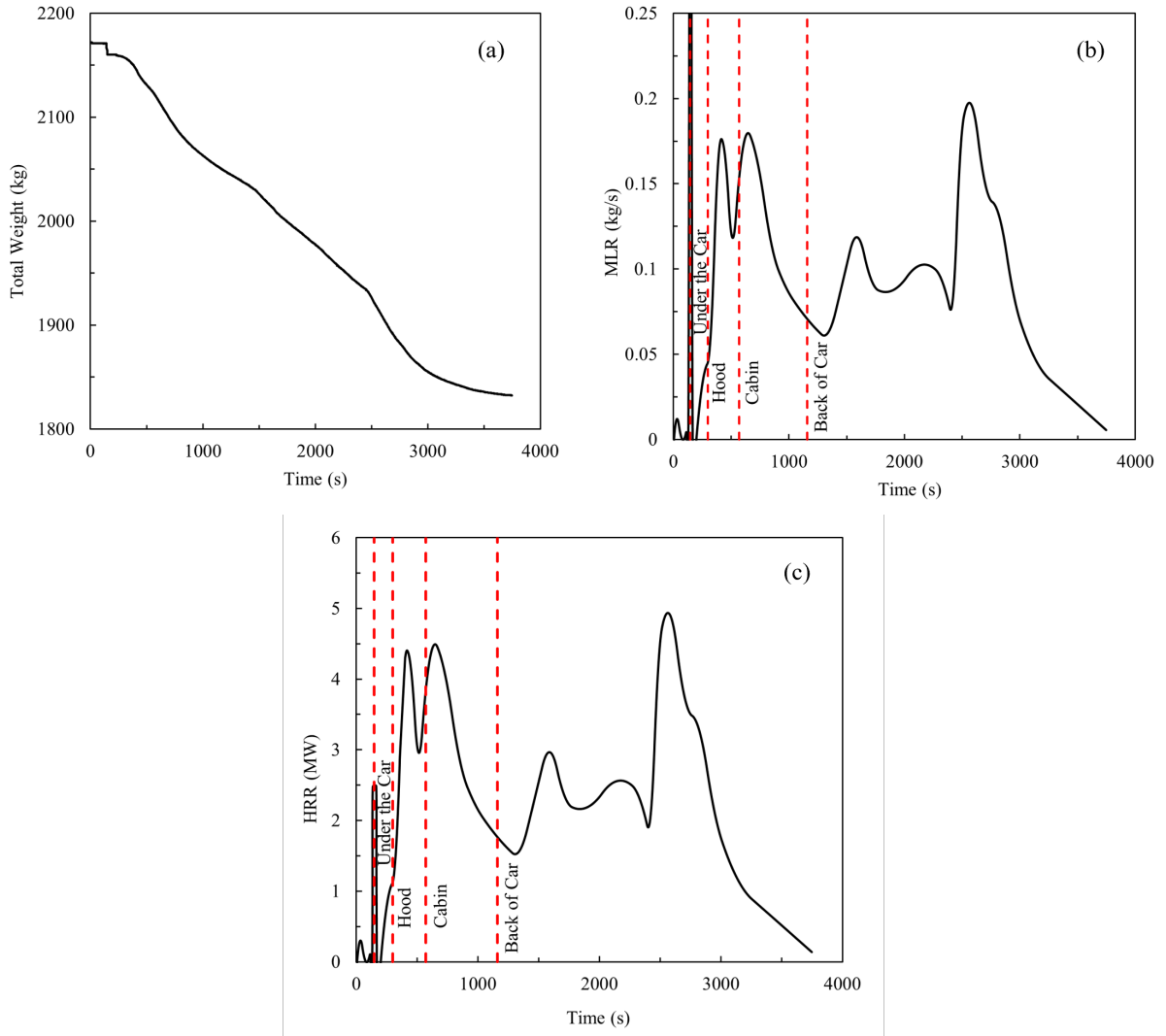


Figure 3.7: (a) The experimental mass loss is the mass of the vehicle lost due to burning over the duration of the experiment. (b) The experimental mass loss rate as the finite difference of the mass loss from figure 3.7a. This experimental mass loss rate is a spline of the mass loss rate such that the data is more legible. (c) The experimental heat release rate calculated from the mass loss rate shown in figure 3.7b.

3.5.2 Heat Flux

Another common value used as performance-based criteria is the heat flux. The heat flux is used to determine how much energy is transferred per unit area to nearby objects. This is used to determine the potential for ignition, tenability, or fire spread beyond the origin of the fire.

Figure 3.8 shows the heat flux as measured by the heat flux gauges placed along the panels (see figure 3.1 and 3.3). It is worth noting the patterns and inconsistencies that occur within the heat flux measurement. The right and left heat flux does not mimic either side, as would be expected in an axisymmetric fire. This indicates more intense burning on one side of the vehicle or tilt present in the flames emanating from vehicle. There is also a tendency for the left heat flux to be higher in the first 1500 s than the right heat flux. After 1500 s the right side becomes the dominant heat flux source. The only time this is not true is on the 4th heat flux where the inverse occurs.

Within the first 300 s the heat flux gauges do not begin at zero. This can be due to a few reasons, one is that the Kona free burn was the last experiment of the series which could have heated the pan beneath the vehicle and potentially heated the air. Very long cables were used making them prone to a voltage offsets affecting the zero reading. This non-zero heat flux is not a concern as the experimental team checked the measured heat transfer with the inverse heat transfer model and found agreement. Concluding that once heat flux significantly increases the heat flux is accurate. The experimental heat flux prior to 300 s is excluded from the comparison to simulated heat fluxes.

Thermal runaway was confirmed within the experimental burn of the Hyundai Kona SEL 2020. Seen in figure 3.8 as the dashed red line at 1500 s. The confirmation of thermal runaway was done through comparing temperature data within the vehicle cab, floor of the vehicle, and observation of increased heat flux without visible changes to burning.

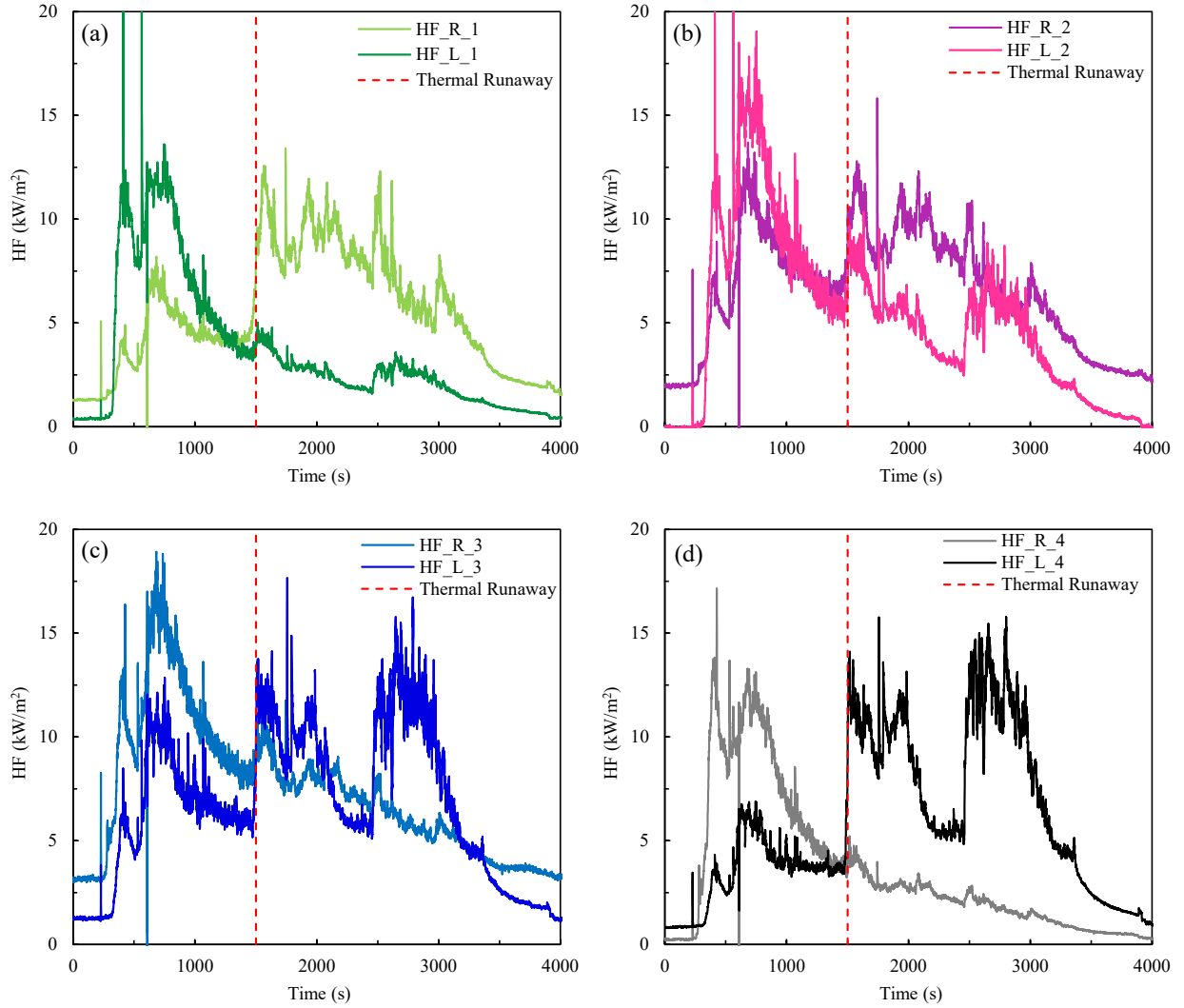


Figure 3.8: The experimental heat flux plotted compares the right and left heat flux gauges individually such that the differences from the first to fourth heat flux gauge could be extrapolated.

Chapter 4: Modeling Methods

CFD models were developed in this work to predict the heat transfer measured in the fire experiments. The research discussed in chapter 1 is used as context to the problem and illustrates the gaps to be filled where previous studies have not cleared documented model setup. FDS input parameters are discussed with some remaining fixed to best replicate experimental setup while others are varied to determine the best match to experimental data. The sensitivity of the model to these parameters was investigated through a parametric study. An iterative process was used as the most influential parameters were identified. The results from this study were analyzed to develop recommendations for best practices in modeling the heat transfer from vehicles.

4.1 Introduction to FDS

FDS was created by the National Institute of Standards and Technology (NIST) as a code based large-eddy simulation for low-speed flows which prioritizes smoke and heat transfer from fires [27]. FDS does this by numerically solving a form of the Navier-Stokes equation suitable for low speed Mach Numbers ($Ma^1 < 0.3$) of thermally-driven flow. These capabilities enable a user to create high-fidelity fire spread models in different scenarios [28]. All models presented in this work use FDS version 6.10.1 release g618ac07. An overview of FDS features are discussed below. Any unstated parameter throughout this chapter is assumed to be set at its default value.

4.1.1 Hydrodynamics

By numerically solving a form of the Navier-Stokes equation appropriate for low speed, thermally driven flow, FDS captures the smoke and heat transfer from fires. The algorithm for this flow is a second order predictor-corrector method accurate in space and time. Flow turbulence is resolved by means of large-eddy simulation which is derived by applying a low-pass filter to the transport of mass, momentum, and energy equations [28].

4.1.2 Combustion

A single-step, mixed-controlled chemical reaction with the simple chemistry model was applied for combustion specification. This treated air, fuel, and products as lumped species [28]. The fire modeling for this project treated all combustible components as a single fuel source delivered through defined burner surfaces. The lithium ion battery materials were not individually characterized. Logistically, the battery burner is located in the same region as the cabin burner, and assigning separate material properties or defining an additional burner specific to the battery was not necessary. The fuel source for the vehicle was an altered propane with oxygen added to the fuel molecule to align with Thornton's rule for energy per unit mass oxygen (EPUMO₂) of 13.1 MJ·kg⁻¹ with the heat of combustion, soot yield, and CO yield selected for a vehicle [29]. Hence, C₃H₈O_{1.55} was used as the fuel. Further discussion regarding rationale is in section 4.2.3.1.

4.1.2.1 Pyrolysis

There are three pyrolysis methods implemented in FDS including fully prescribed pyrolysis, prescribed pyrolysis after an ignition temperature is reached, and detailed Arrhenius kinet-

ics [28]. Prescribed burning requires a ramp function to define the fire growth as well as the mass loss rate per unit area, or the heat release rate per unit area. The next method, temperature threshold, is similar to the first, but pyrolysis begins when the local surface temperature of fuel in a grid cell exceeds the defined threshold. This method is commonly used for modeling fire spread and delayed ignition. The final method is a more realistic representation of burning behavior, but requires detailed inputs on reaction properties that are not typically available [30].

The given case could use either prescribed burning, fire spread through an ignition temperature, or the Arrhenius kinetics based model. The HRR of the fire is predicted rather than prescribed in the latter two methods. Models with these methods would be interesting but hard to match with experiments as FDS determines the location of fire spread. This makes it difficult to isolate the impact of individual parameters on the predicted heat transfer for direct comparison with the experiments. To create the most analogous model, the mass loss rate in all the models was prescribed using experimental data for the duration of the simulation. The mass loss rate was used in two different ways, 1) in a multi-burner vehicle geometry, and 2) in a single burner representation. Prescribed burning was chosen to consistently portray pyrolysis between the multi- and single burner models for comparison to experimental data.

4.1.3 Heat Transfer

4.1.3.1 Radiation

Radiation transport in FDS is resolved using the finite volume method and absorption coefficients calculated with RADCAL [31]. The finite volume method uses radiation angles as the discretized angular directions used by FDS to solve the Radiation Transport Equation. These an-

gles represent the directions in which thermal radiation is emitted, absorbed, and scattered within the simulation's domain. To discretize the space, the angular domain (4π steradians) is divided into a finite number of solid angles, each of which is defined by a direction vector representing the orientation of radiation propagation. These vectors are used to compute the radiative intensity in each direction. The radiation angles are relative to each gas-phase cell in the domain where every cell emits and receives radiation from all directions. The radiation angles then define in which direction the energy is exchanged. The number of radiation angles can be an important input in FDS, and is varied in this work (see Section 4.4.3). The thermal radiation emitted from the fire has a range of wavelengths. Physically, the absorption coefficient of the gas mixture to each wavelength is different. However, FDS implements a gray gas assumption where a single effective absorption coefficient is computed with RADCAL. The absorption coefficient calculation requires a user to specify the radiative `PATH_LENGTH`. A single path length was used in this work (see Section 4.2.4.2). FDS uses the radiative fraction to set a lower bound on the fraction of thermal radiative energy released due to combustion. The background humidity assumed in the model has an impact on the absorption coefficient which can affect the radiation predictions (see Section 4.2.4.1). Liquid droplets can also affect the radiation transport due to the droplets ability to absorb and scatter thermal radiation [28]. However, this impact was not considered in this work.

4.1.3.2 Convection

A sub-grid model is needed for convection when using the large-eddy simulation method. The default approach in FDS is to compute a convective heat transfer coefficient, h , based on

the larger of the Nusselt numbers for natural and forced convection. The default convection correlations for horizontal and vertical flat walls was used for all simulations [28].

4.1.3.3 Conduction

Conduction can be modeled in either either a one or three dimensional solver. For many fire scenarios a one dimensional solver provides reasonable agreement as the surface heat transfer does not vary significantly between adjacent grid cells. The default one dimensional solver was used for all simulations. Conduction required obstructions to be one gas phase cell length greater than the prescribed surface thicknesses as an artifact of FDS ignoring `BACKING= 'EXPOSED'` and applying `BACKING= 'VOID'` (which assumes the solid is well-insulated) [28].

4.1.4 Computational Meshes

The domain and grid size of the simulation are foundational to the model setup since they are the backdrop to the geometry and combustion calculations. The domain is the overall three dimensional volume where calculations are performed within the model. This can be thought of as a “room” where exterior impacts to the flow field do not significantly affect the predictions. This allows for the combustion calculations to be completed for the defined domain. The mesh or grid size is the dimension of the cells that make up the domain. Each cell is a location where local conditions are resolved and used to determine the conditions of the surrounding cell, thus making up a grid. For the purposes of this thesis a 10 cm grid was used in the “near field” and 20 cm in the “far field”. The determination of the “near field” grid is discussed in the appendix section [A.1](#).

4.2 Fixed Input Parameters

4.2.1 Geometry Construction

Careful consideration is needed on the level of detail required on the vehicle geometry. The boundaries of the vehicle and the placement of the burners with respect to the vehicle will have an impact on the heat transfer predictions. The vehicle measurements and blueprint were obtained and used to approximate the dimensions of the FDS vehicle to a 10 cm grid size [24]. Various iterations of vehicle shapes were constructed using different obstructions and hole arrangements. Each of which had benefits and drawbacks. The fewer obstructions used, the quicker the vehicle could be made. However, this results in errors when assigning a burner to an area with a hole, or delineation of burner placement. Additionally, obstructions that were one gas phase cell length greater than the prescribed surface thicknesses failed to conduct heat as an artifact of FDS ignoring `BACKING= 'EXPOSED'` and applying `BACKING= 'VOID'` (which assumes the solid is well-insulated) [28]. The vehicle shape is constructed using 10 cm slabs. This allows for the wheel wells to be open to the exterior and allow flame through. Side windows were purposely left out since during experimentation they were fully rolled down. Burner placement is also more customizable as it can be set to a multitude of locations within the vehicle. Of course, this method is more time consuming and becomes a question of priority.

The model setup also included the steel panels, which were part of a system used to measure heat flux over a large area. These are included in the model because of any airflow impacts they might have on the vehicle fire. Similarly, the drippage pan is also present as it may affect the fire direction and re-radiation back onto the vehicle. The construction of the model vehicle can

be seen in figure 4.1. Additionally, figure 4.1(c) demonstrates the interior compartments of the vehicle as well as the wheel wells.

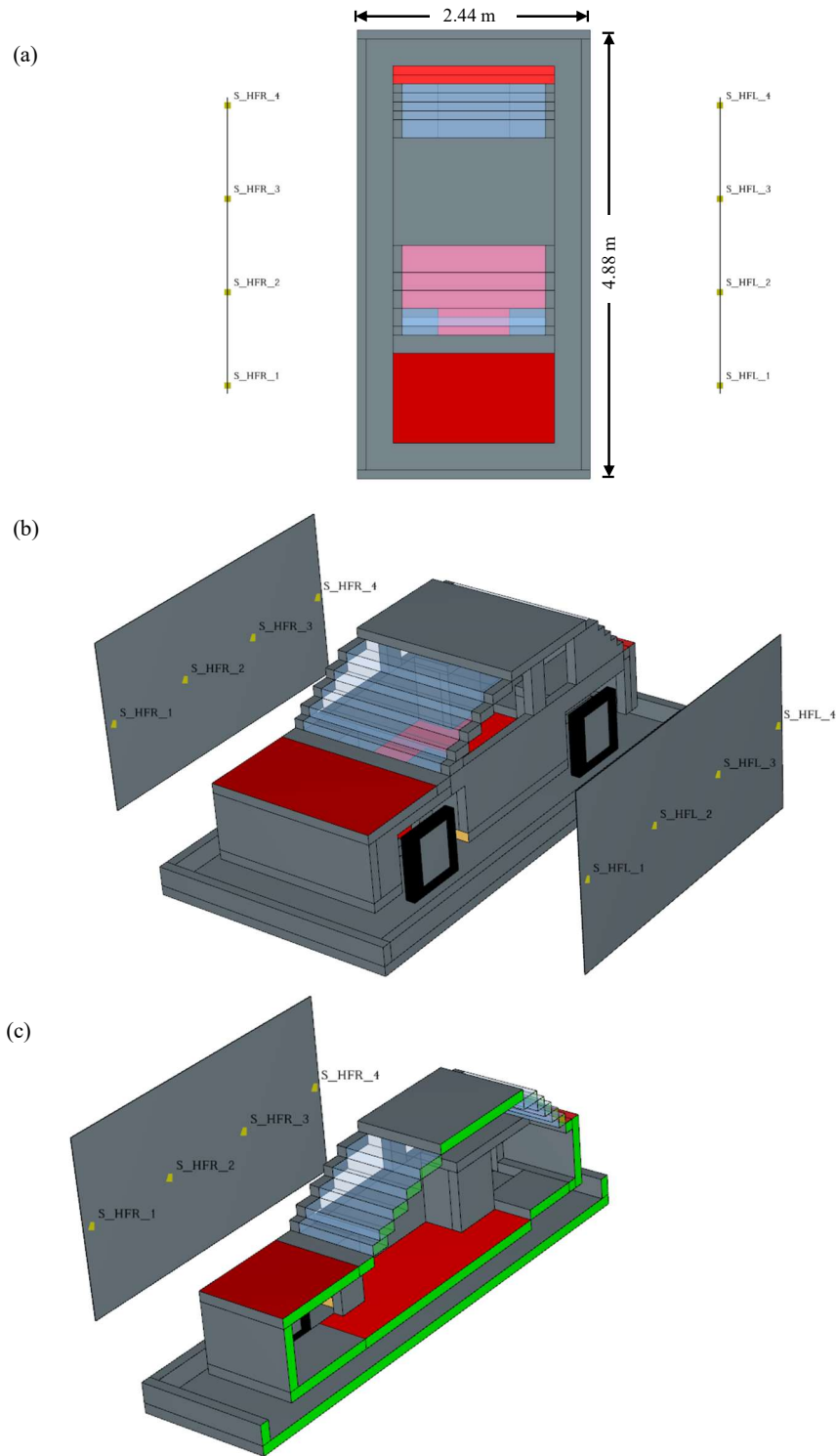


Figure 4.1: Multi-perspective views of the model in FDS. Sub-figure (a) is a Top down view for comparison to figure 3.1. (b) Isometric full vehicle view (c) Cross-section isometric view showing the interior burner and shape of the vehicle.

4.2.2 Materials

The materials chosen were dictated by the major components of the vehicle. Particularly, the components that will be prone to heat transfer onto areas of interest such as next to the vehicle and above the vehicle. This also includes structurally important members, such as the hood, fenders, ceiling, windshield pillars, center pillars, and rear pillars. These members are critical to every vehicle and why they were emphasized when choosing the materials. Vehicles are often comprised of a variety of materials including steel, high-strength steel, aluminum, magnesium alloys, carbon-fiber, polypropylene, polyvinyl chloride, and of course, rubber [11]. The materials listed are often layered or part of a larger component of a piece of the vehicle, such as bumpers, seating, dashboards, and internal and external trim [11]. Due to how intertwined these materials are within the vehicle complexity can rapidly increase if all are considered. For the purposes of this project, only the largest percentage and base materials were assigned to the model. Thus, there is no layering of materials but rather the primary component. Some emissivities were altered to reflect the paint on the vehicle. Most of the constructed vehicles is comprised of 1.2 mm steel which is typical for a laminate steel used in vehicle construction. Other materials include 3 cm steel for the rims of the wheels, rubber for the tires, and glass for the front and back windshield. A list of the materials used and the component to which it was assigned can be seen in table 4.1, where the columns are as follows, C_p is specific heat, λ is conductivity, ρ is density, ϵ is emissivity, and α is the absorption coefficient [27, 32, 33, 34, 35].

Table 4.1: Thermophysical properties of materials used as model inputs

Parameter	C_p (kJ·kg ⁻¹ ·K ⁻¹)	λ (W·m ⁻¹ ·K ⁻¹)	ρ (kg·m ⁻³)	ϵ	α (m ⁻¹)	Components
Concrete	1.04	1.80	2280	-	-	Domain grade level
Rubber	2.15	0.09	910	-	-	Vehicle tires
Steel	0.46	45.8	7850	0.95	-	Vehicle body
Glass	0.84	1.05	2500	0.92	0.5	Windshields

4.2.3 Fire Spread and Combustion

The fire spread throughout the vehicle is one of the compelling characteristics to recreate in FDS. The vehicle must appear to be burning in a similar pattern to match video and heat flux measurements. This of course is somewhat arbitrary to where the burners are placed for this particular vehicle. Nevertheless, the scope is to resolve the experimental results in FDS. Figure 4.2a shows the multi-burner vehicle representation developed in this work. Figure 4.2b shows the single burner representation of the vehicle fire with the red rectangular surface which is typical of engineering design practice.

Fire spread in the multi-burner representation was determined by breaking up burning into different critical sections of the vehicle. The fire started under the vehicle, which was expected since that was the location of the ignition burner. From there it spread to the hood, then into the cabin, and lastly igniting the rear portion of the vehicle (see §3.4). Figure 4.2a depicts the vehicle with multiple burners (shown in red) to replicate the described fire spread. There were more locations in which the fire was seen, such as its emergence from the wheel wells causing the tires to ignite. Within the cabin there is additional spread, such as the seating vs. the ceiling within the cabin ignition. These fine details can be modeled, but for the purpose of this study, a simplified

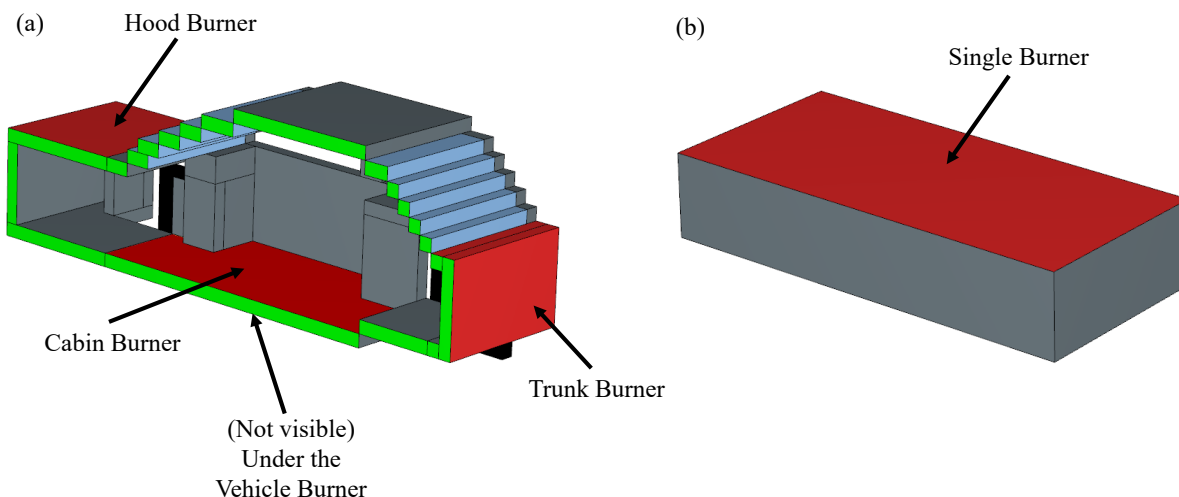


Figure 4.2: Figure (a) shows the locations of the various burners on a vehicle geometry. Figure (b) represents the vehicle as a single burner. All burners are red in color.

approach was chosen as introducing additional burner complexity becomes a matter of increasing quantity and time investment rather than enhancing analytical capability. The delineation of under the vehicle, hood, cabin, and trunk burning allows for the observance of the percent of burning which occurs in each of these areas (i.e., how much burning a singular burner experiences in comparison to its counter parts such that all burners must add to 100%). This allows for burner spread to reflect the real case-scenario as well as simplified changes to which burner is allowed more emphasis. (i.e. changing the amount of burning one burner experiences by lowering or raising the percent burning of a different burner.)

4.2.3.1 Fuel Species

The fuel species makeup of a vehicle is complex given the amount of different materials within it. In this work, the composition of the vehicle is considered in two parts. First, the composition of the lithium-ion battery in the propulsion system. And second, the contribution of

non-battery combustible materials in the vehicle. The objective of this section is to describe the basis of the gas phase reaction used in this work.

Lithium-ion batteries are comprised of different elements such as lithium, nickel, cobalt, manganese, and graphite which form the cathode, and anode, respectively. Electrolytes and a separator fill the space between the cathode and anode [36]. Ultimately, these electrodes and separators are layered to construct the cell configuration. A complex electrical system comprised of high-voltage wiring interconnects the cells. The battery also has its own cooling system and protective case to preserve structural integrity and increase fire resistance [36].

Although the battery is a significant mass of fuel and unique to the construction of the vehicle a large amount of the burning can be attributed to the polymers [6, 37]. As established in chapter 2 the increased quantity of polymers in modern vehicles has increased the similarities between ICEVs and EVs resulting in vehicle cabins which should have comparable burns. Since the battery is located in the vehicle's cabin area, assumptions to the FDS representation of a battery were made to make the simulation more practical to accomplish. Specifically simulating a fire due to thermal runaway is out of scope for this effort. It is also of interest to determine if the model can have reasonable agreement with the experiment without modeling the intricacies of the battery. Literature is available concerning how lithium-ion batteries can be modeled, often times as previously mentioned it is a very complex process [8]. However, further research into vehicle modeling where the lithium-ion battery is present would be necessary to further corroborate the accuracy of the model's comparison to the experimental case.

As discussed in section 4.2.2, there are many components within a vehicle [11]. The non-battery combustible material in the vehicle is in large part polyvinyl chloride, polypropylene, and polyurethane, consisting of over two thirds of the plastic in the vehicle [37]. All of these are solid

materials. Therefore the pyrolysis based model could be used for spread, but as established in section 4.2.3 the simplified mode of fire spread was chosen to be prescribed burning. Hence, an approximation for these materials had to be developed. Multiple hydrocarbon fuels are available for prescribed burning, since the FDS combustion model allows adjustment of the combustion reaction based on the amount of carbon and hydrogen [28]. Ethylene and propane were two hydrocarbons that were considered as possibilities to substitute polyvinyl chloride, polypropylene, and polyurethane. The heats of combustion were compared, both have similar heats of combustion at about $43.7 \text{ MJ}\cdot\text{kg}^{-1}$ for propane and $41.5 \text{ MJ}\cdot\text{kg}^{-1}$ for ethylene [26]. Polyvinyl chloride, polypropylene, and polyurethane had heat's of combustion of $10 \text{ MJ}\cdot\text{kg}^{-1}$, $37 \text{ MJ}\cdot\text{kg}^{-1}$, and $18 \text{ MJ}\cdot\text{kg}^{-1}$, respectively [26]. Given the commonalities between the proposed fuels and how they compared to the actual fuel it was difficult to gauge which fuel would be a best fit. However, given propane's common use in modeling and soot yield being a parametric parameter in this study, propane was chosen as the fuel species. Propane's chemical formula is C_3H_8 , with a heat of combustion of $43.7 \text{ MJ}\cdot\text{kg}^{-1}$ [26]. However, this heat of combustion is not an accurate representation of the heat of combustion of the vehicle which was found to be $25 \text{ MJ}\cdot\text{kg}^{-1}$ [2, 9, 10]. Multiple values for the heat of combustion have been reported most ranging from $13 \text{ MJ}\cdot\text{kg}^{-1}$ to $30 \text{ MJ}\cdot\text{kg}^{-1}$ [9]. Averaging these heats of combustion from relevant experiments results in a heat of combustion of $23.7 \text{ MJ}\cdot\text{kg}^{-1}$. Miechówka et al. calculated an average heat of combustion based on total burning time, mass loss rate, and heat release rate, this resulting value was $25 \text{ MJ}\cdot\text{kg}^{-1}$ [10]. During FSRI's experimental burns a heat of combustion of $25 \text{ MJ}\cdot\text{kg}^{-1}$ was used [2]. Given the minimal numerical difference between $23.7 \text{ MJ}\cdot\text{kg}^{-1}$ and $25 \text{ MJ}\cdot\text{kg}^{-1}$, a heat of combustion of $25 \text{ MJ}\cdot\text{kg}^{-1}$ was chosen to best match the measurements and calculations conducted by FSRI [2]. FDS calculates the EPUMO_2 based on the fuel's stoichiometry, however,

since the fuel has a different heat of combustion than the vehicle the resultant EPUMO₂ does not align with Thornton’s observation which maintains an EPUMO₂ of 13.1 MJ·kg⁻¹ [29]. This would result in a low flame temperature and affect the heat transfer throughout the vehicle and would ultimately lead to a heat flux that is too low. Additionally, any experimental vehicle heat of combustion calculated with calorimetry would use an EPUMO₂ of 13.1 MJ·kg⁻¹. Hence, to align with Thornton’s rule and the vehicle heat of combustion 1.55 oxygen atoms were added to the fuel species. Resulting in the fuel’s chemical formula being C₃H₈O_{1.55} [29, 38, 39].

4.2.3.2 CO Yield

The CO_YIELD is the mass of CO that is produced per unit mass of the fuel burned, i.e. units of (kg-CO·kg-fuel⁻¹). The quantity of CO produced by the fire impacts the tenability of occupants exposed to the smoke. Although tenability is not within the scope of this project, the CO_YIELD was applied for accuracy of burning. The averaged CO yields of polyvinyl chloride, polypropylene, and polyurethane were used since these materials are representative of the non-battery combustible load of the vehicle as discussed in sections 4.2.2, and 4.2.3.1. The resultant CO yield was 0.04 and was used in the FDS input file [26].

4.2.3.3 Background CO₂ Concentration

The amount of CO₂ within the domain can be altered depending on local conditions. However, the concentration of CO₂ in the laboratory was not measured in these experiments. The default background mass fraction of CO₂ was maintained in FDS (0.06%) [26].

4.2.4 Heat Transfer

The parameters below affect the heat transfer through air from the vehicle to the target. This is not an exhaustive list, but rather a selection which were used as fixed parameters in the model. Other parametric heat transfer parameters are later discussed in section 4.4. These include soot yield § 4.4.1, radiative fraction § 4.4.2, and the number of radiation angles § 4.4.3. The results of each of these analyses are presented in chapter 6.

4.2.4.1 Humidity

Humidity has an effect on the radiation transport and attenuation throughout the model. The more humidity that is in the air the less radiation is transmitted to nearby obstructions. This is due to the water content in the air absorbing the radiation. FDS' default humidity is 40%. To replicate experimental conditions, historical weather data was drawn for the day of the test and the approximate time. The humidity was found to be 45% which is similar to the default but did mildly alter the measured gauge heat flux by 3%. To best match the experimental data the experimental humidity was used [40]. For comparison, when the humidity was set to 40% the measured heat flux rose by $2\text{kW}\cdot\text{m}^{-2}$.

4.2.4.2 Path Length

The thermal radiation emitted from the fire and how quickly it is absorbed by background species is determined by the absorption coefficient as discussed in Section 4.1.3. The FDS default path length is 0.1 m which is designed for heat fluxes near the fire. Since the distance from the vehicle fire to the heat flux gauges was 2 m the `PATH_LENGTH` was set as such.

4.2.4.3 Burner Surface Temperature

TMP_FRONT specifies the temperature of a surface and all fuel species entering the domain from the surface. Artificially low values would underestimate the enthalpy in the plume due to the sensible heating of the fuel species and the convective exchange with the gas phase. This was set to 300°C in this work which was representative of pre-ignition temperatures observed in testing of polyvinyl-chloride, polypropylene, and polyurethane (see section 4.2.3.1)[39]. A value slightly below the ignition temperatures of these polymers was chosen since the vehicle had not yet ignited. This also is used to account for the sand burner which was placed beneath the vehicle for ignition. The sand burner would have preheated some of the vehicle surfaces prior to their ignition.

4.3 Mass Loss Rate Modeling

The mass loss rate is one of the most important parameters to ensure accuracy with the experimental setup because it defines the energy input that represents the fire. FDS requires two inputs to calculate the mass loss rate: 1) the mass loss rate per unit area or the heat release rate per unit area and 2) a ramp function. The ramp function is the time dependent fraction of burning relative to the maximum heat release rate. The number of burners present in the model impacts how the mass loss rate is calculated. However, only a single mass loss rate measurement could be derived from the four scales on which the vehicle was placed. There is no direct measurement describing the mass loss rate of individual portions of the vehicle such as the hood, battery, or cabin.

Two different approaches considered to represent the burning vehicle were to split the burn-

ing across multiple burners and a single burner fire. When simplifying the vehicle as one single burner, the mass loss rate per unit area was calculated as follows. First, the area of the burner was determined, i.e. the rectangular area the vehicle would occupy, next the maximum mass loss rate is found in the experimental data. This value is necessary to calculate the mass loss rate per unit area, $MLRPUA$ ($\text{kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$), the maximum experimental mass loss rate is MLR_{max} ($\text{kg}\cdot\text{s}^{-1}$), and the area of the burner over which combustion will occur is A_{burner} (m^2),

$$MLRPUA = \frac{MLR_{max}}{A_{burner}}. \quad (4.1)$$

The mass loss rate per unit area value is then multiplied by the RAMP function and the area of the burner to obtain the calculated mass loss rate at any discrete time step. To determine the ramp function, the mass loss rate at relevant times was determined from the experimental data and divided by the maximum mass loss rate. These times are determined through observation of experimental data particularly the peaks and troughs are chosen as well as some others for best fit to the experimental data. This process can be as detailed as needed but at a minimum should include all peaks and troughs if using a prescribed mass loss rate.

Calculating the mass loss rate for multiple burners is similar but requires video analysis and engineering assumptions as to how much and where the burning in the vehicle is occurring. The detailed explanation on how this is completed is in section 3.4. Once the direction of spread and percent burning is determined then the time-dependent specification of the ramp function can continue. The percent burning for each burner is estimated from video analysis. Then percent burning is iteratively multiplied by the experimental mass loss rate, and divided by the maximum experimental mass loss rate. Equation 4.2 shows the ramp function calculation for one burner,

which is completed at each time step of the chosen experimental mass loss rate:

$$RAMP(t) = \frac{MLR(t) \cdot \alpha(t)}{MLR_{max}}, \quad (4.2)$$

where $MLR(t)$ is the experimental mass loss rate at chosen times ($\text{kg} \cdot \text{s}^{-1}$), $\alpha(t)$ is the percent burning of one burner in comparison to the rest of the vehicle (%), and MLR_{max} is the maximum mass loss rate of the experimental data ($\text{kg} \cdot \text{s}^{-1}$). This process is repeated for all burners such that each burner has a unique ramp function due to $\alpha(t)$ (percent burning).

The resultant output of equations 4.1, and 4.2 when input into FDS is shown in figure 4.3. Where the experimental mass loss rate is compared to the simulation's multi-burner output mass loss rate. The single burner MLR is identical to the multi-burner option. The experimental data as discussed in section 3.5 is a spline of the original data to remove noise completed by experimentalists [2]. Figure 4.3 demonstrates how the simulated data accurately reaches the peaks and troughs of the experimental mass loss rate. The remaining curvature found in the experimental is not necessary to replicate since FDS will conduct those calculations as a linear approximation. If a more detailed simulated mass loss rate is required, it is a matter of inputting the sufficient time steps with corresponding mass loss rate. This would produce a simulated mass loss rate graph that is more round. Nevertheless, by seeing figure 4.3 it is clear that the peaks and the troughs are being achieved, while still maintaining the shape of the experimental mass loss rate curve.

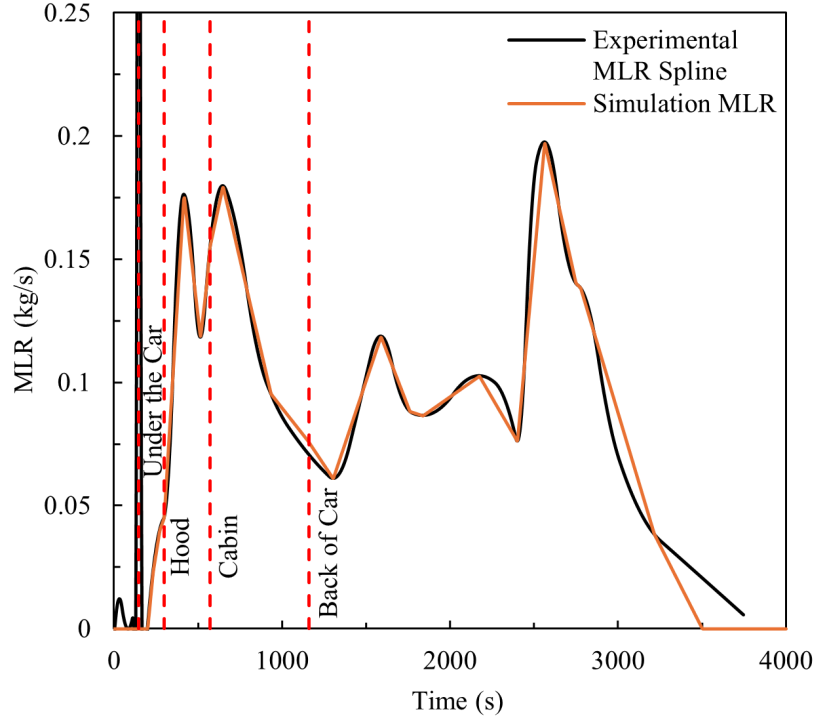


Figure 4.3: The experimental mass loss rate is plotted with the multi-burner simulation mass loss rate. The goal of the simulation was to match the peaks and troughs of the experimental mass loss rate. The detail based on curvature can be adjusted by adding more points.

4.4 Parametric Input Parameters

4.4.1 Soot Yield

The soot yield prescribes how much soot is produced by the defined fuel as the kg of soot produced for every kg of fuel burned. This soot primarily affects the soot concentration in the gas phase, the smoke obscuration, and the radiative heat transfer. For the purposes of this project the soot's greatest affect is the radiative heat transfer and its concentration in the gas phase. The higher the soot value the more black bodies fill the gas and radiate out to the heat flux gauges. Hence, the soot yield controls the local emissivity and absorption of radiant

energy, with the radiative fraction controlling the radiant energy released by burning [28]. Thus, the radiative fraction should have a higher impact on the heat flux. Nevertheless, the soot yield must be considered as a potential factor affecting the measured heat flux. The initial soot yield used was 0.015 as this value was considered plausible for the combination of materials present in the vehicle [17]. Upon further review of [26] it became apparent that a higher soot yield would also be accurate. Table 4.2 shows common vehicle components, their corresponding soot yield, and in which vehicle component they could be found [26, 41]. Hence, multiple soot yields were chosen to determine the impact of a low soot yield. Chosen soot yield values were 0.001, 0.015, 0.05, and 0.1, with many of the corresponding soot yields found ranging between 0.05, and 0.1. Particularly of interest is the NIST vehicle burns which tested three vehicles on a pine straw bed and found soot yields between 0.03 and 0.06.

Table 4.2: Table summarizing soot yields of some vehicle components

Material	Soot Yield	Component
PVC-Nylon	0.115	Fabric components
PVC (LOI=0.5)	0.098	Dashboards and paneling
Polypropylene	0.05	Bumper and paneling
Polyurethane	0.131	Seat Cushion
NIST Vehicle Burn	0.03 - 0.06	Full Vehicle

4.4.2 Radiative Fraction

The radiative fraction defined on the &REAC line is the energy emitted as radiation divided by the total energy release. Hence, this term defines how much heat is released via thermal

radiation rather than convective heat. Therefore, this value has a large impact on the radiation that reaches the heat flux gauges and therefore the corresponding measurement. The higher the radiative fraction the more heat is transferred via radiation due to the σT^4 dependence of the flame's radiated energy, in turn affecting the surface heating of the vehicle, and potential for material ignition. The FDS default radiative fraction is 0.35 which is considered typical for hydrocarbons such as the fuel defined in section 4.2.3.1 [28]. Additionally, literature also indicated other instances where the radiative fraction was set to 0.35 for vehicle modeling [7, 17]. To quantify the impact of radiative fraction a range of values was tested including 0.2, 0.35, 0.5, and 0.7.

4.4.3 Number of Radiation Angles

FDS's implementation of the number of radiation angles is discussed in section 4.1.3.1. An illustration on how the radiation angles are interpreted from the heat flux gauge is shown in figure 4.4. Each arrow emitting from the heat flux gauge represents a discrete radiation solid angle, a full set of these would cover the 4π of a sphere in a gas-phase cell, for clarity only a subset of angles are depicted. The arrow-head, demonstrates the orientation in which the heat flux gauge and the radiation angle “look” for radiation. The solid angle Ω demonstrates a single “view angle”.

The more radiation angles there are, the higher the resolution of the radiative intensity is, particularly in the far field. This, of course, requires more computing power and/or time. Too few radiation angles, results in a non-physical “star-like” pattern in the radiation fields. These can be observed in a heat flux boundary applied to the FDS code. Since the computational cost may be

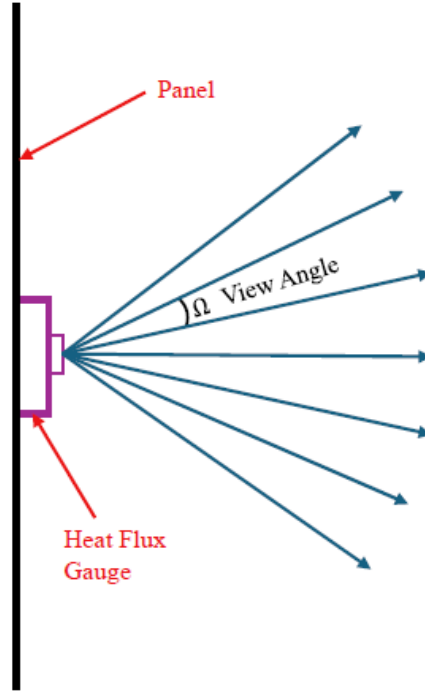


Figure 4.4: Depiction of radiation angles where the solid angle Ω demonstrates the view of the radiation angle.

high, the radiation angles should only be added until there is no "star-like" pattern, and there is no longer a significant change in the measured heat flux. The default number of radiation angles is 100. To determine the critical angular resolution equation 4.3 can be used [28, 42],

$$NRA_{critical} = N_{\Omega} \cdot \frac{4\pi H^2}{S}, \quad (4.3)$$

where N_{Ω} is the number of angles per steradian, H is the separation distance (m), and S is the emission source surface area (m²). The number of angles per steradian, N_{Ω} , is chosen as 5 and 10 to bracket $NRA_{critical}$ [42]. For the setup shown in figure 3.1, the radiation angles are particularly relevant to the heat flux gauges mounted on the panels and the radiation they measure from the

vehicle. Hence, the separation distance, H in this scenario, is the distance from the side of the vehicle to the heat flux gauges. Since none of the burners are on the side of the vehicle, the surface area was varied ranging from 0.1 m^2 to 15 m^2 . At 2 m^2 the number of required radiation angles was approaching the default. Hence limiting the reasonable surface areas from 0.1 to 2 m^2 . Along with varying surface area, the calculation was completed using both 5 and 10 as N_Ω . Considering the outputted values and radiation angles, it was deemed that a surface area of 1 m^2 was the most realistic. This resulted in the number of radiation angles ranging from 213.4 to 426.8 . Focus was placed on testing radiation angles in the range of the calculated results, 200 , 300 , 400 , from equation 4.3. A few values outside of that range 50 , 100 , and 500 were tested to ensure convergence of the number of radiation angles.

4.4.4 Flame Extinction

The `&COMB` line is part of the global combustion parameters that affect how fire and combustion are modeled throughout the simulation, particularly regarding how the oxygen concentration within the burning enclosure is driven by combustion. These conditions can be defined by different modes such as setting `SUPPRESSION= 'FALSE'` and `EXTINCTION_MODEL= 'EXTINCTION 2'`. `SUPPRESSION= '.FALSE.'` allows combustion to continue under mixing limitations until the local O_2 reaches zero rather than having the lower limit set as a function of the local cell temperature [28]. Alternatively, `'EXTINCTION 2'` allows for combustion extinction if the O_2 drops below a critical value dependent on local cell temperature [28]. Hence, the burning would migrate to cells with higher oxygen concentration; i.e. moving burning towards the outside of the vehicle. This may potentially increase heat transfer

and raise the heat flux, as the flames would be in more direct sight to the gauges than with interior burning of the vehicle. These parameter settings were compared to each other to determine the resulting effects and which would be the most appropriate fit.

4.4.5 Vehicle Geometry

Previously in section 3.4 it was stated that a vehicle fire could be viewed with two levels of complexity.

1. Taking the vehicle as simply “on fire,” where the vehicle would be a single burner.
2. Splitting the burning areas into major vehicle components where multiple burners would be present.

These two states were established with the assumption that a multi-burner vehicle fire would out perform a single burner vehicle representation and therefore improve agreement amongst experimental heat flux measurements. However, a multi-burner geometry requires more time and consideration to construct with more material assignment, ensuring conduction throughout the vehicle components, and constructing the vehicle geometry to the proper dimensions. After which all the burners must be placed and tuned to the proper percent burning as discussed in section 4.3, where 4 burners were chosen to represent a multi-burner vehicle fire (§3.4, §4.2.3. This poses the important question of “can a vehicle fire be simplified to a single burner?” This idea is relevant to this project since the methodology aims to assist performance-based fire protection engineers in their work which is often time restricted. Hence, a single burner version of the vehicle was evaluated.

All surrounding obstructions and parameters discussed in chapter 4 remained the same

while only the shape of the vehicle was altered. The new set-up can be seen in figure 4.5. The dimensions of the single burner were maintained as the length and width of the Hyundai Kona 2020 that was used for experimentation. With only one burner to model the FDS input slightly alters as discussed in section 4.3. Where equation 4.1 remains unchanged and $\alpha(t)$ within the ramp function 4.2 is removed. However, height requires consideration since the location of the fire relative to the gauges could heavily impact the lateral heat flux, posing the question “if the gauges are obstructed how much does that impose on the gauges readings?”. Additionally, it is unrealistic to portray the burning as occurring at roof height considering the most combustible material in a vehicle is contained within the cabin rather than on the roof. To test this concept the burner height was altered to 4 different levels in increments of 0.4 m. The varying single burner fire heights began at 0.4 m above wheel base height (i.e., ground). Lower values were not tested since they approached the wheel geometry such that it would limit the believability of where significant combustible vehicle components are located. Figure 4.6 indicates the various heights tested relative to the heat flux gauges and to the vehicle geometry. To be clear, the single burner simulations were not run with vehicle obstruction geometry, see figure 4.5, but rather geometry is shown in figure 4.6 to demonstrate the location of these heights relative to vehicle features.

4.5 Ceiling Heat Flux Analysis

This project focused on reconstruction of the fire dynamics of the vehicle and heat transfer to targets occurring in the test. However, the intent was for this model to be applicable to various scenarios, where ignition to near by objects and structural failure may be a concern. Hence, the importance placed upon determining the accuracy of the model as compared to the experimental

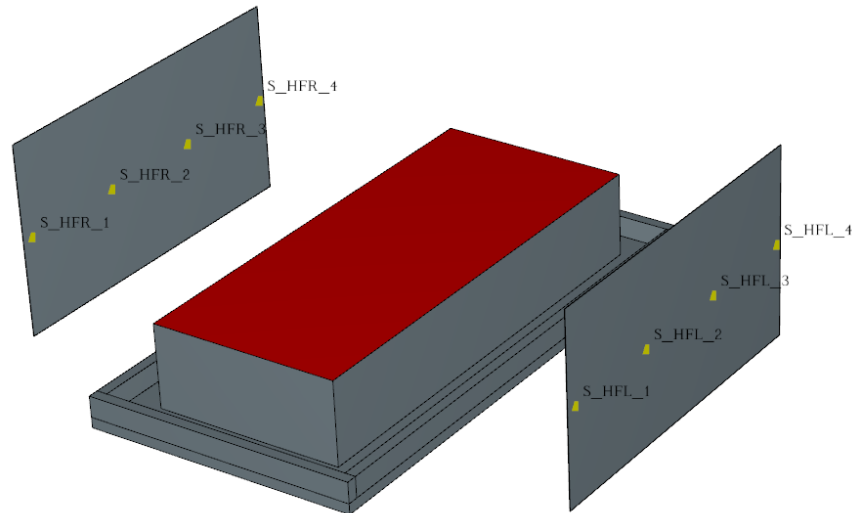


Figure 4.5: The simulation setup with a singular burner portraying the vehicle.

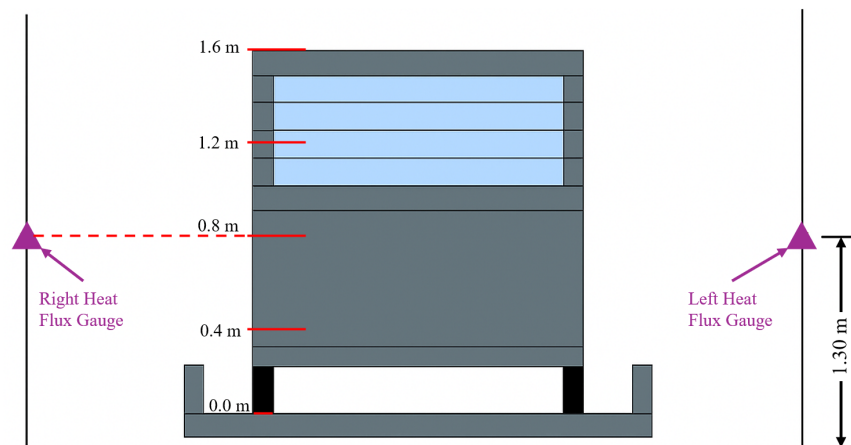


Figure 4.6: The single burner was tested at various heights to observe the impact on the heat flux measured and its agreement to the experimental data. Here the different heights applied to the single burner are shown relative to the heat flux gauges and the vehicle geometry.

data. The heat flux provides a comparison of how much energy could be reaching a near by object when exposed to a vehicle fire. However, heat would not solely be exposed to the lateral areas of the vehicle but also, in large quantities, directly above the fire itself. This is of great concern with vehicle fires particularly since large quantities of vehicles can be found in car park garages. Heat would not solely be exposed to the lateral areas of the vehicle as measured in

this test, but also directly above the fire. This is of great concern with vehicle fires particularly since large quantities of vehicles can be found in car park garages. The ceilings of the car park garages can be left vulnerable to direct flame impingement and thus weakening of its structural integrity. Additionally, traditional prescriptive fire codes have limited information on designing for an electric vehicle fire. In this situation, performance-based design is used to ensure the safety of the structure. To properly do this fire protection engineers must not only be able to model the vehicle but also ensure the proportion of heat energy reaching susceptible structures is accurate.

A ceiling could not be placed above the vehicle during the experimental burns, as doing so would disrupt the free-burning nature of the test and impede proper exhaust within the laboratory. The benefit of a simulation is a ceiling at standard height for parking garages could be included into both multi-burner and single burner runs to see any preliminary impacts from the vehicle fire. The ceiling was placed at a height of 3.1 m which is between the range in NFPA 88A given at 10 to 12 ft (3.05 m to 3.66 m) [43]. Since the addition of the ceiling is a trial concept, further investigation into heat flux as measured by gauges or at varying locations is not discussed. This merely serves as an example of a potential application to the free burn multi and single burner models constructed in this project.

Chapter 5: Analytical Procedure

5.1 Residual Area

Although comparison of the simulation outputs can be accomplished by graphing pairs of data, the comparison is only qualitative. A more quantitative measure to calculate the difference between data points, particularly between the heat flux data is needed. This development process began by considering the coefficient of determination [44]

$$R^2 = 1 - \frac{\sum_i (E_i - S_i)^2}{\sum_i (E_i - \bar{E})^2}, \quad (5.1)$$

where E_i and S_i are the experimental and simulated heat fluxes at time t_i with the sums carried out over a time period of interest. The experimental mean over the same time period is given by $\bar{E}$. The numerator of the second term is the residual sum of squares where the residual is the difference between the experimental and simulated heat flux. Figure 5.1 shows both a negative and positive residuals respectively. In the coefficient of determination, these residuals are squared and the numerator is positive definite with a range of $[0, \infty)$, where zero occurs when the simulation exactly matches the experiment. The denominator is the total sum of squares, and is

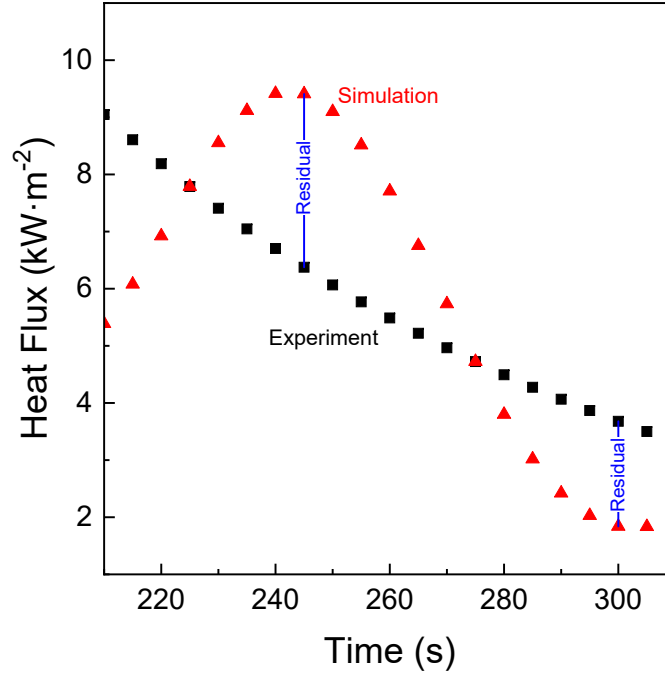


Figure 5.1: Depiction of residuals for a comparison between experimental (■) and simulated (▲) heat fluxes. The left labeled residual is negative, while the right labeled residual is positive. The experimental and simulated data is notional.

closely related to the experimental variance,

$$\sigma^2 = \frac{1}{n} \sum_{i=1}^n (E_i - \bar{E})^2, \quad (5.2)$$

where n is the number of samples. In regression analysis, R^2 measures the proportion of variance in the dependent variable which is explained by the independent variable. In practice, it is often considered to be a measure of the “goodness-of-fit” between the experimental data and a model equation, where unity indicates an equation that perfectly fits the experimental data. The R^2 value is further bound in linear regression analysis to be in the range of $[0, 1]$, but this range does not hold for non-linear functions [44].

In the research of this thesis, regression analysis was not chosen, rather a computational

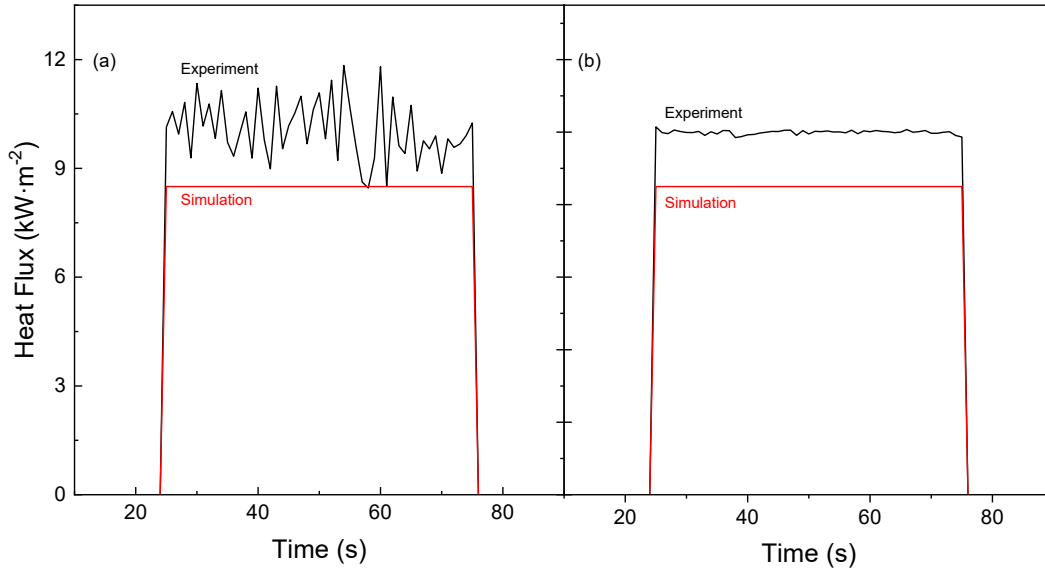


Figure 5.2: Notional experimental (—) and simulated (—) heat fluxes. In both graphs the mean of the experimental heat fluxes is $10 \text{ kW}\cdot\text{m}^{-2}$ between 25 and 50 s and the simulated heat flux means are $8.5 \text{ kW}\cdot\text{m}^{-2}$. In graph (a), the experimental heat flux has $1 \text{ kW}\cdot\text{m}^{-2}$ Gaussian noise, while in graph (b) this noise is decreased to $0.05 \text{ kW}\cdot\text{m}^{-2}$

fluid dynamics (CFD) model of the experiment was simulated. This allowed for a comparison between the experimental and simulated time series data. Due to the highly nonlinear simulation R^2 becomes unbound with a range of $(-\infty, 1]$ and its interpretation more nuanced. The unbound range is considered in figure 5.2 where the experimental and simulated heat flux is constant between 25 and 75 s. Figure 5.2a is a sample of the experimental heat flux including Gaussian noise with a standard deviation of 10% of the mean value. Similarly, figure 5.2b contains this noise but reduced to 0.5% of the mean value.

The simulated heat flux with no noise added is 15% below the experimental data. The R^2 calculated between 25 and 75 s goes from -1.8 for the 10% Gaussian noise to -766 for the 0.5% Gaussian noise, despite having no difference on how well the simulation agrees or disagrees with the experiment. Both data sets have a simulated heat flux on average of 15% below the experimental data.

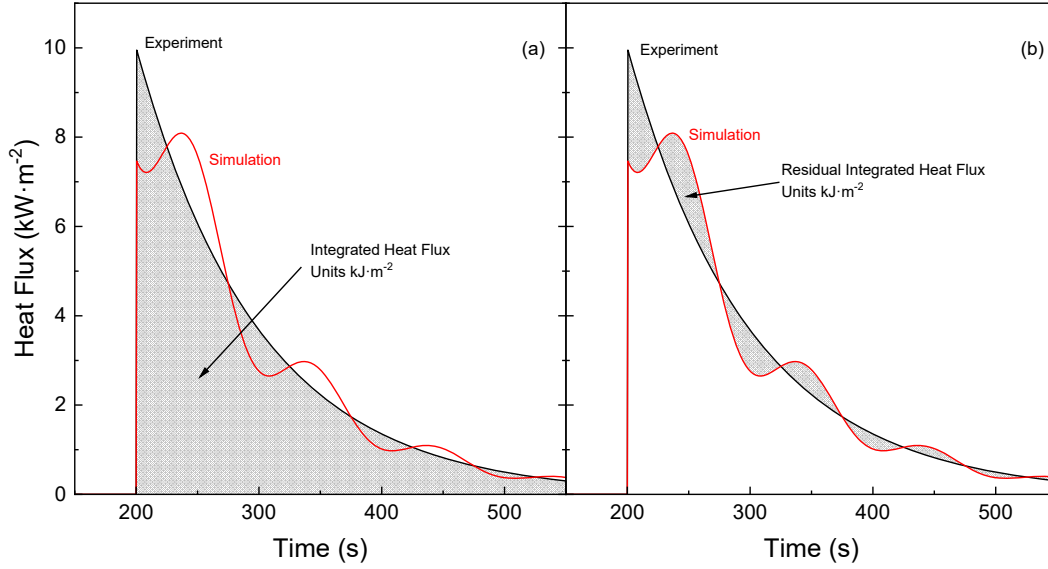


Figure 5.3: Notional experimental (—) and simulated (—) heat fluxes. In graph (a), the shaded region is the experimental energy per unit area. In graph (b), the shaded region is the absolute residual area between the experimental and simulated heat fluxes.

A less noise sensitive metric to compare experimental and simulated heat fluxes was desired. In the integral form, the metric of comparison used was

$$D = \frac{\int_{t_o}^{t_f} |E(t) - S(t)| dt}{\int_{t_o}^{t_f} E(t) dt}, \quad (5.3)$$

where $E(t)$ is the experimental heat flux and $S(t)$ is the simulated heat flux. The denominator is nothing more than the experimental thermal energy per unit area,

$$H = \int_{t_o}^{t_f} E(t) dt \quad (5.4)$$

which is the heat energy delivered to a surface per unit area (e.g. kJ·m⁻²) and is schematically depicted in figure 5.3a. The numerator is the absolute residual area between the experimental and simulated heat fluxes and is depicted in figure 5.3b. Hence, this difference, D , has a range

of $[0, \infty)$ with zero indicating perfect agreement between the experimental and simulated heat fluxes. Returning to the heat fluxes in figure 5.2, both differences are 15% independent of the noise added to the experimental heat fluxes.

The experimental and simulated heat fluxes are discrete data sets and the integrals in equations (5.3) must be discretized. Using a simple Riemann sum, the difference equation becomes,

$$D = \frac{\sum_{i=i_o}^{i_f} |E_i - S_i| \Delta t_i}{\sum_{i=i_o}^{i_f} E_i \Delta t_i}, \quad (5.5)$$

where Δt_i is the difference in time between the i and $i + 1$ data points, and the sum is carried out over the region of interest.

These difference formulas in equation (5.3) through (5.5) are for a single heat flux gauge but can be easily extended across all gauges to provide a single measure of the total difference between the experiment and a simulation. This total difference is given by

$$D_{tot} = \frac{\sum_{h=h_o}^{h_f} R_h}{\sum_{h=h_o}^{h_f} H_h}, \quad (5.6)$$

where the sum is carried over the the heat flux gauges in the system. The absolute residual area for a specific heat flux gauge, R_h , is the numerator in equation (5.5),

$$R_h = \sum_{i=i_o}^{i_f} |E_{h,i} - S_{h,i}| \Delta t_i, \quad (5.7)$$

and H_h is the experimental thermal energy per unit area for the gauge,

$$H_h = \sum_{i=i_o}^{i_f} E_{h,i} \Delta t_i. \quad (5.8)$$

with the experimental, $E_{h,i}$, and simulated $S_{h,i}$ now indexed to the heat flux gauges, in addition to time. When a single difference between experimental and simulated heatflux gauges is reported, it was calculated using equation (5.6).

5.2 Root Mean Square Error

The root mean square error (RMSE) is another common form to report closeness of fit of simulated data to experimental data. The premise of RMSE informs on average how far the predicted values are from the actual values. This is describes as the square root of the average of the squared differences between the actual and predicted values.

$$RMSE = \sqrt{\frac{1}{N} \sum_{i=1}^N (y_i - \hat{y}_i)^2} \quad (5.9)$$

where N is the number of data points, y_i is the experimental data, and $\hat{y}_i$ is the simulation data. RMSE provides the benefit that the units remain the same as the original data. I.e. the simulation data can be interpreted to be x units away from the experimental data. Thus, the lower RMSE value corresponds to a better predictive accuracy. With the inverse holding true that the higher RMSE indicates larger errors between predicted and actual values.

RMSE however, should not be the only error metric used because the difference between experimental and simulated data is not normalized. Resulting in a potentially inaccurate view of

error magnitude.

Chapter 6: Results

The outcomes of this project are a direct result of the model setup discussed in chapter 4. With the primary goal of this thesis is to create guidelines and estimate vehicle heat transfer based on experimental data using FDS. With the aim to assist fire protection engineers who model vehicle fires and produce information regarding the benefits and drawbacks of applying greater detail to vehicle modeling.

The EV Kona 2020 vehicle simulation was modeled in two different ways; one as a single burner fire, and the second with vehicle geometry containing 4 different burners to emulate fire spread. A key component of the analysis within this thesis is to compare the single burner and multi-burner simulations to each other and the experimental data provided by FSRI's free burn of the same vehicle modeled. The comparison includes the heat flux received by the heat flux gauges (see figure 3.1), and the fire spread based on video evidence.

The parametric analysis of the soot yield, radiative fraction, number of radiation angles, and the flame extinction is also discussed. These parameters were chosen to determine their effect on the simulations heat flux and the comparison to experimental data. The parameters were investigated under the assumption that they would yield the greatest impact on heat flux.

6.1 Soot Yield

The heat fluxes of the tested soot yields (0.001, 0.015, 0.05, and 0.1) were compared to the experimental data using equation 5.5. Where the soot yield with the lowest percentage would be sought after. The percentages found for each soot yield are shown in figure 6.1a.

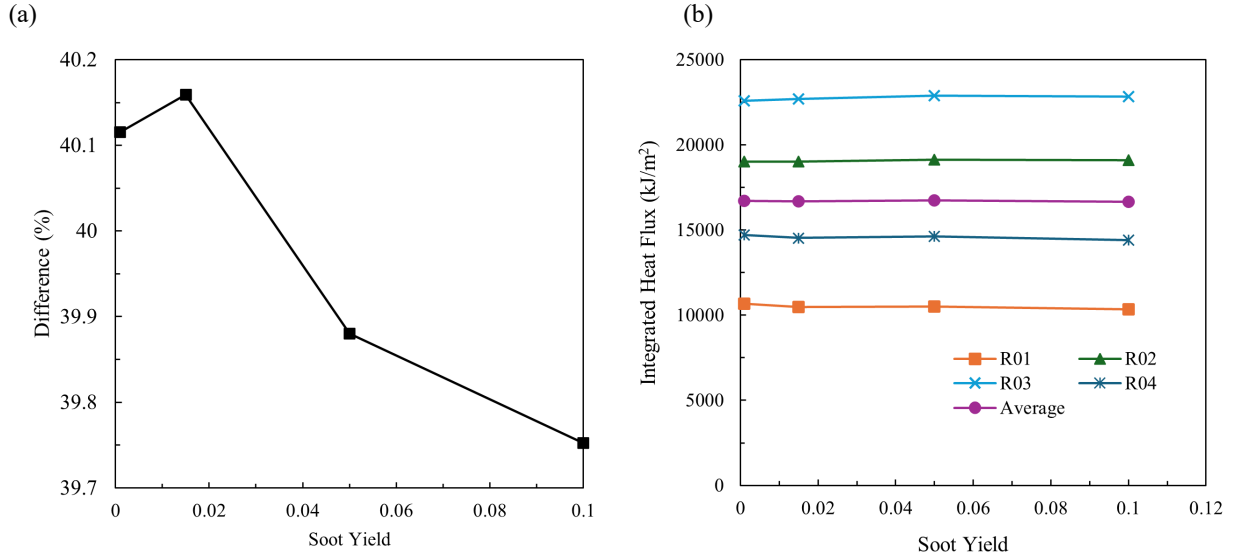


Figure 6.1: Figure (a) is the varying soot yields compared to the experimental data to determine their impact on the heat flux. Figure (b) is the integrated heat flux as a function of the varying soot yields ran over the full simulation time of 4000 s.

It is important to note the y-axis of figure 6.1a, as it only spans from 39.7% to 40.2%, thus indicating a mere 0.5% difference between the simulations and experimental results. A 0.5% difference in how the simulation and experiment compare is very minimal. Figure 6.1b highlights the negligible change in heat flux since the plot depicts the integrated heat flux over the varying soot yields. Although the heat flux was not significantly affected by the soot yield, other fire characteristics that affect tenability may be impacted. A very relevant example of this is parking garages, where the movement, respiratory, and visibility of the smoke is affected by the

soot yield. The higher the soot yield the more rapidly the smoke layer will descend, impacting visibility, and smoke exposure of those egressing the parking garage. Due to the minimal impact the soot yield was set at 0.015.

6.2 Radiative Fraction

The simulation with radiative fractions of 0.2, 0.35, 0.5, and 0.7, were compared with the experimental data using equation 5.5, the comparison is seen in figure 6.2a and b. The difference in figure 6.2a has a significant decrease of over 10% going from a radiative fraction of 0.2 to 0.35 or 0.7 to 0.5. Interestingly, there is a minimal difference in accuracy when the radiative fraction was changed from 0.35 to 0.5 even though the difference in these two values is close to 0.2. However, the radiative fraction of 0.5 matches the experimental data better than 0.35 by only 0.6%. Showing that there is minimal benefit in increasing the radiative fraction to 0.5. This is a physically reasonable finding since an assigned radiative fraction of 0.5 would not be based on the fire size or dynamics but rather on the attempt to match the experimental data. Figure 6.2b indicates a linear increase in heat flux from altering the radiative fraction, speaking to the impact this term can have simulation results. Fires where the soot stays within the flame region with a hydrocarbon fuel tend to have a radiative fraction of 0.35 which is what other vehicle modelers have used [7, 17].

A large change in the integrated heat flux seen in figure 6.2b does not translate to a correspondingly large difference in figure 6.2a. This arises because the difference to the experiment, figure 6.2a, does not account for the simulation being higher or lower than the experiment. For example, the radiative fraction of 0.2 is lower than the experiment while 0.7 is above but the

residual area between these simulations and experiment are nearly equal.

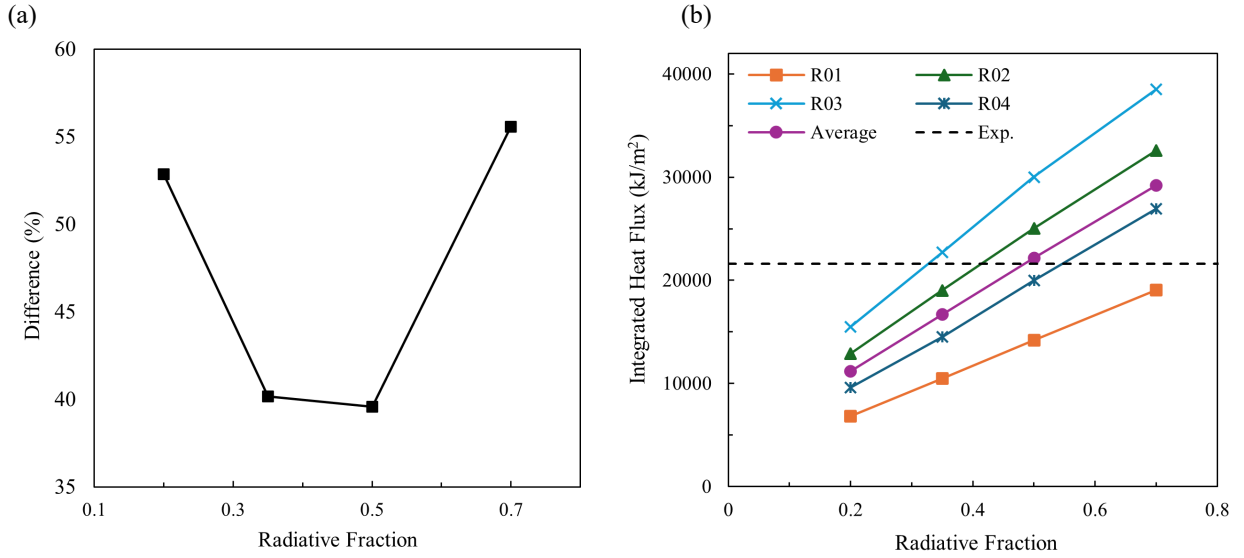


Figure 6.2: Figure (a) shows simulations with varying radiative fractions as compared to experimental data to determine their impact on the heat flux. Figure (b) is the integrated heat flux as a function of the varying radiative fractions ran over the full simulation time of 4000 s. The dashed line indicates the experimental integrated heat flux averaged over all eight gauges.

6.3 Number of Radiation Angles

The six sets of radiation angles produced similar results. 100 radiation angles did not produce the “star-like” pattern worthy of concern, however, the heat flux gauges did measure more radiation when compared to the models that ran with 50, 200, 300, 400 and 500 radiation angles. Furthermore, the difference in heat flux measurements between these six radiation angles was compared with each other and the experimental data. Figure 6.3a illustrates the total difference between heat flux gauges and experimental data for each radiation angle. Figure 6.3a demonstrates the radiation angle convergence and how after 400 radiation angles there is minimal return for increasing radiation angles. What might be perceived as unexpected is the increasing difference between the experimental and simulated heat flux as radiation angles are increased with

a 5 percentage point change from 100 to 500 radiation angles. However, this should not be a concern as increasing radiation angles increased the accuracy of radiation calculation from FDS. Hence, the relevance is placed on convergence, where increasing the radiation angles begins to produce the same results, thus suggesting the proper range of radiation angles to apply. Similarly, Figure 6.3b shows decreasing heat flux as the number of radiation angles increases, with the exception of 50 radiation angles. Hence, demonstrating stability of the radiation angles from 100 to 500 radiation angles. Both figures 6.3a and b showed the least change in heat flux between 400 and 500 radiation angles, resulting in 400 radiation angles as the optimal value.

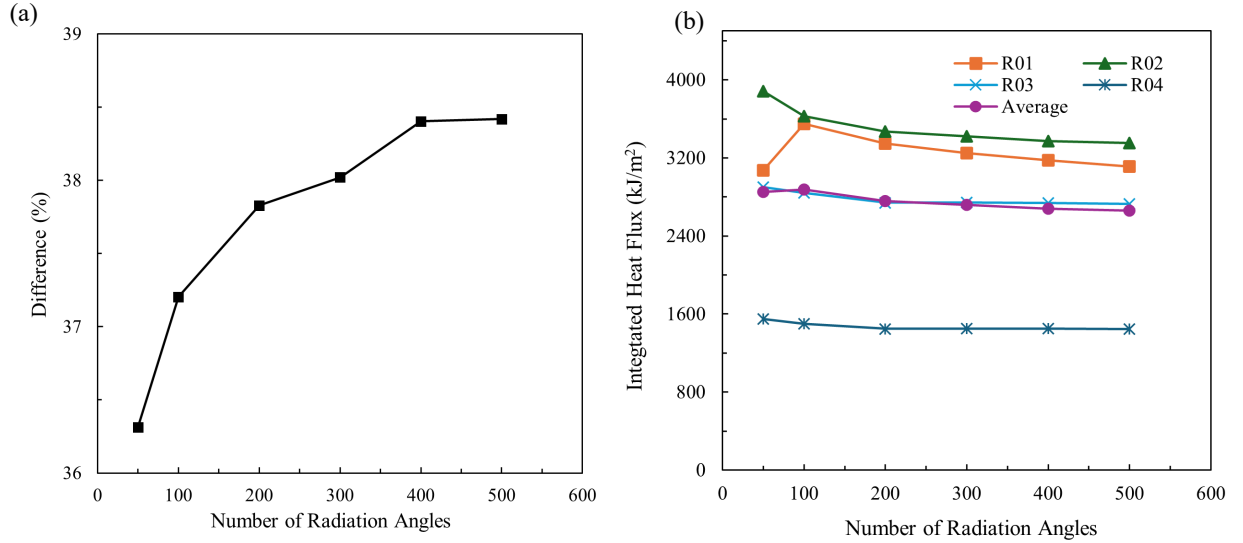


Figure 6.3: Radiation angles of 50, 100, 200, 300, 400 and 500 tested. Figure (a) is the difference between the heat flux gauges and the experimental data. Figure (b) is the integrated heat flux for each heat flux gauge and the average of all gauges ran over the first 800 s period.

6.4 Flame Extinction

Due to the binary nature and minimal impact this parameter had the individual heat flux gauge plots are not shown. The primary difference in shape is in the latter 2500 s where the

'EXTINCTION 2' model is lower than the SUPPRESSION='FALSE' version. Prior to those 2500 s these two simulations are practically identical to each other including heat flux gauge 3 where even during the 2500 s they remained aligned. When considering the difference for the entire run &COMB EXTINCTION_MODEL='EXTINCTION 2' was only 6% larger than &COMB SUPPRESSION='.FALSE.'. This increase was only due to gauge 4. When focusing only prior to 2500 s that difference decreased to 5% indicating that a 2% portion of the variability occurred only after a large portion of combustion had occurred. There was a greater variation when looking at the heat flux gauges moving front to back regardless of the time period observed. For the full 4000 s gauge number 2 matched the best with a difference of 4.5% while gauge number 4 had more than double that difference at 12.3%. Since the exterior provided minimal fuel and EXTINCTION 2 did not show significant increase in heat flux &COMB SUPPRESSION='.FALSE.' was chosen.

6.5 Vehicle Geometry

6.5.1 Multi-Burner Comparison to Experimental Data

In section 3.5 the experimental data was shown including some discussion regarding the irregularities within the data. With the applications of chapter 4 the simulation is compared to the experimental data. The simulation seen is considered the “baseline” simulation, where parameters chosen were based on best fit and engineering judgment. It should be noted that the simulations right and left heat flux is symmetric, hence all simulation heat flux is presented using the right heat flux gauge.

Observing figure 6.4(a)-(d) it can be seen that the first 800 s the simulation matched the ex-

perimental data well, with the simulation differing from the experimental by approximately 38% between 300 and 800 s (see equation 5.6). When difference analysis is extended to 4000 s that difference from experimental to simulation increases to approximately 41% (see equation 5.6). The increase is present because beyond 800 s burning begins to alternate favor between the right and left heat flux gauges. The data is not expected to perfectly match either the right or left experimental heat flux. This is due to the inherent tilt present in the experimental data. Although there was no noticeable wind flow or favor when viewing the experimental burn video it is obvious from section 3.5, and figure 6.4(a)-(d), that the heat flux measured was not symmetric. However, due to the nature of FDS its heat flux measurements remain symmetric unless air flow is introduced or the prescribed burners are placed asymmetrically. Figures 6.4(a)-(d) also demonstrates varying levels of agreement depending on what heat flux gauge is being observed. Figure 6.4(a) and (d) show gauge pairs for positions 1 and 4 with alternating agreement and disagreement between the left and the right gauges, with a difference of 53%. Figure 6.4(b) and (c) show gauge pairs for positions 2 and 3 with better overall agreement between left and right experimental gauges, with a difference of only 31%.

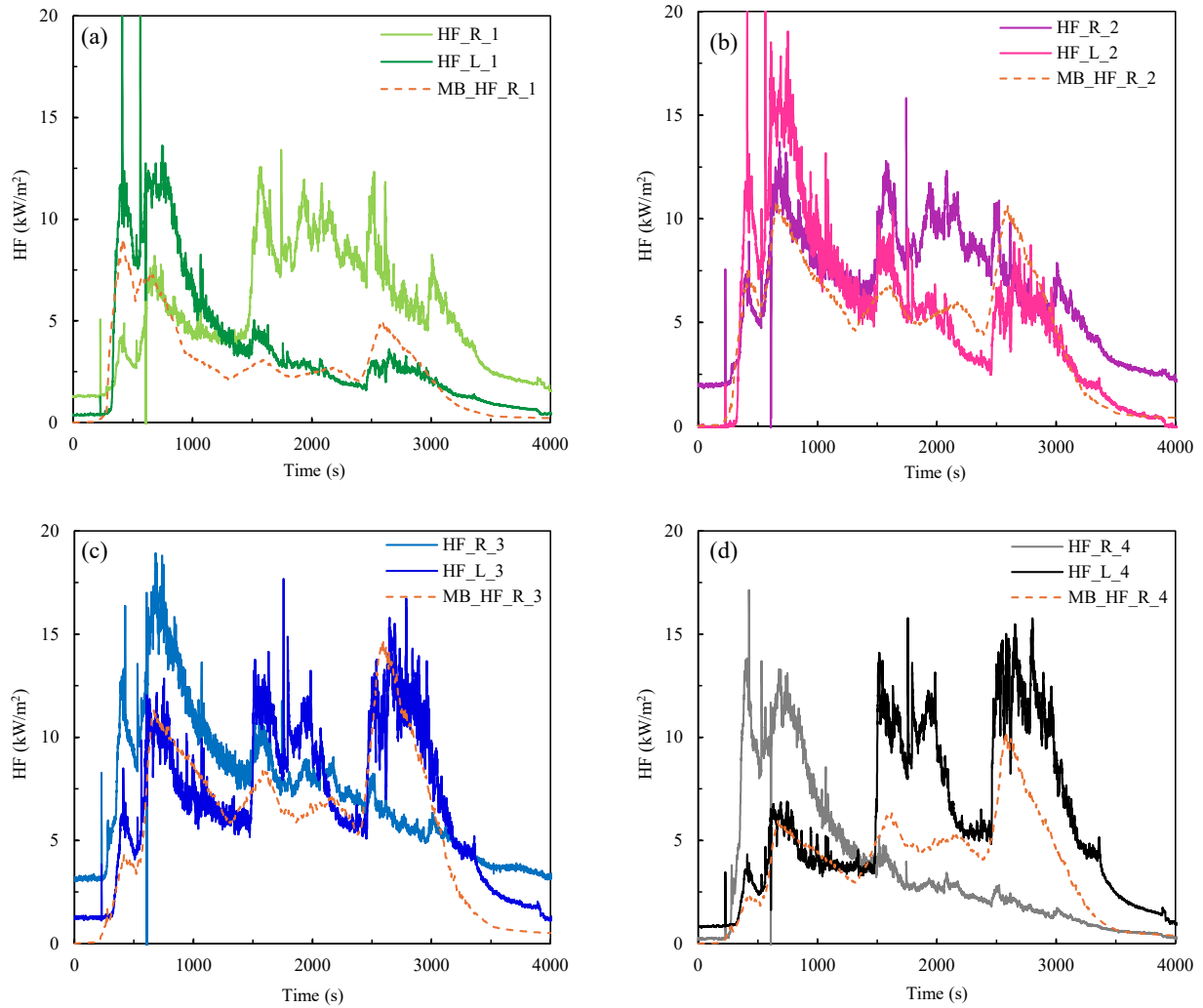


Figure 6.4: Figures (a)-(d) show the experimental heat flux plotted as like colors per gauge (i.e. green and light green is gauge 1, pink and purple is gauge 2, etc.). Meanwhile, the simulation with multiple burners is shown in as dashed orange curve to contrast the experimental data.

On a gauge by gauge comparison, gauge number 1, figure 6.4a, begins slightly below the left experimental gauge and above the right. At 1500 s the right experimental gauge drastically increases while the left begins to decrease over the next 1000 s. During this period the simulation adheres to the left experimental heat flux. Most likely due to the onset of thermal runaway at approximately 1500 s, causing drastic experimental increases in gauges 1, 3, and 4, this however did not increase visible burning. Heat flux gauge number 2, figure 6.4b, begins by adhering to

the right more than the left. At about the 1500 s mark the right heat flux gauge begins to increase more than the left and thus the simulation switches to match the left gauge's measurements. Near the end at about 2500 s the simulation surpasses both sets of experimental data. Experimental video evidence shows an increase in flaming at this time accompanied by a deep crackling sound, the cause is unclear as the rear portion of the vehicle including the tire seem to already be ignited. The left third heat flux gauge, figure 6.4c, begins with an incredible match to the simulation. After that 1500 s mark the left gauge begins to increase and the simulation prediction is lower than both measurements. Nevertheless, the shape of the measurement is accurate. At 2500 s the simulation's third gauge returns to match the left experimental heat flux gauge. The final heat flux gauge number 4, figure 6.4d is where the greatest differences lie. At the start the left and right gauges have the greatest difference with the right being approximately $6 \text{ kW}\cdot\text{m}^{-2}$ larger than the left. Interestingly, gauge 4 is where the simulation begins at its lowest but adhering to the left gauge until 1500 s where the left and right gauges quickly diverge and stay diverged. Meanwhile, the simulated measurement stays between both the left and right experimental measurement.

It is worth noting that the simulation's first heat flux gauge recorded the lowest overall heat fluxes followed by gauge number 4. Thus meaning that the simulation's gauges 2 and 3 had the highest measurement. The finding is not surprising considering gauges 2 and 3 are located in the middle of the panel and thus are in a prime location to view the most combustion from the vehicle. Hence, gauges 2 and 3 had the most similarities to the experimental data. However, the experimental data did not maintain that gauges 2 and 3 had to be the highest in value. Barring the first left and 4th right where the heat flux had steady decay after approximately 800 s.

Another option for comparing the experimental heat flux measurements is averaging all the gauge measurements then determining the difference. This provides a comparison without

the analysis of left and right or front to back. However, this value should not be taken without the consideration of the individual heat flux gauge as it can misconstrue data, appearing to have a lower heat flux than what was actually present. For comparison to the simulated heat flux this averaging provides insight to the simulations accuracy without regarding tilt. As a result the difference between the simulation and the experimental decreases by about 10% to 28% in closeness between the two. The heat flux comparison can also be distinguished using the RMSE method and the flux time product [45, 46]. The root mean square error provides the difference in units of the measurement, (i.e. $\text{kW}\cdot\text{m}^{-2}$) thus contextualizing the measurement as a heat flux as opposed to considering the area between the curves which is how equation 5.5. The flux time product is similar to RMSE in the units that are used. However, flux time product is used as a measure of ignitability. Hence, comparing the experimental heat flux to the simulation heat flux may be useful in determining the likelihood of either causing ignition. This could be particularly beneficial when comparing the simulations ability to model the ignition of a nearby object.

6.5.2 Single Burner Heights

The different single burner heights (0.4 m, 0.8 m, 1.2 m, 1.6 m) did indeed impact the level of heat flux measured by the gauges. Changing the height resulted in a percent difference ranging from 72.4% to 40.5% when compared to the experimental data. Hence indicating how relevant the single burner height would be when setting the burner to replicate a vehicle fire. Figure 6.5a demonstrates all the burner heights tested and their individual agreement to the experimental data. The heights of 1.2 m and 1.6 m caused the single burner fire to be above the line of sight of the heat flux gauge as seen in figure 4.6 and hence, is unsurprising they produced the largest difference as

their reported heat flux was significantly lower than the experimental data. The single burner fire with a height of 0.4 m had more combustion occurring in the direct line of sight to the heat flux gauges. Although this did increase the heat flux measurement it resulted in an overestimation of the heat flux, which is seen as a larger difference in figure 6.5a. Figure 6.5b shows the direct effect of the burner height on the heat flux, particularly due to the placement of the heat flux gauges. As a result, the lower the single burner is the higher the heat flux becomes. Based on figure 6.5a the best agreement to both the experimental data and the multi-burner simulation was found with half of the vehicle height (0.8 m), which is what was used for the primary comparison between multi-burner and single burner in section 6.5.3.

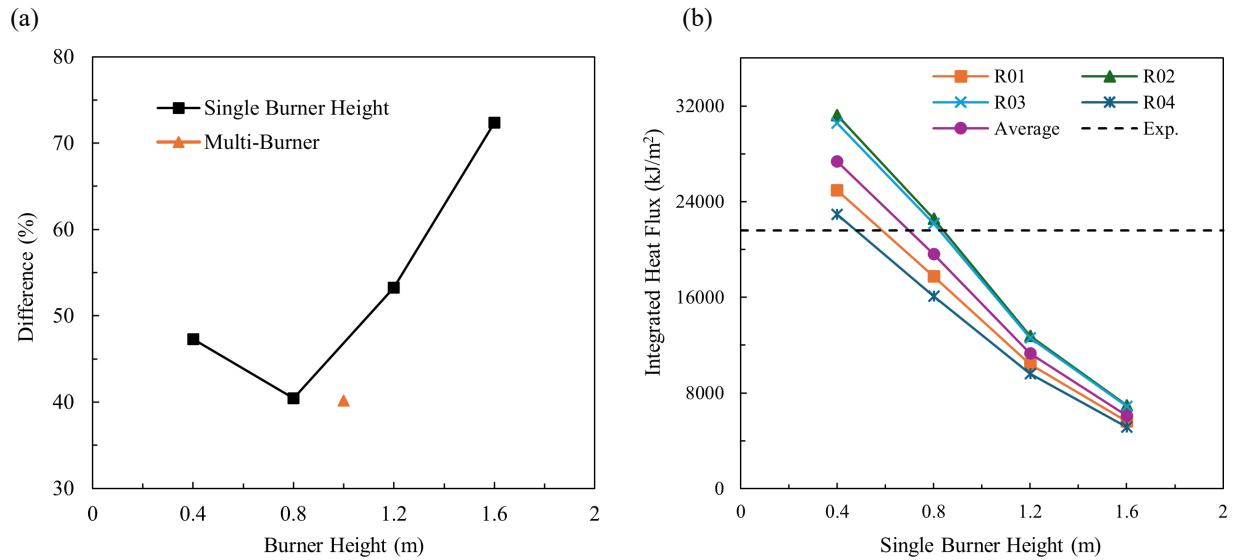


Figure 6.5: Figure (a) demonstrates varying the single burner height and its agreement to the experimental data. The multi-burner agreement is also shown for reference. Figure (b) is the integrated heat flux as a function of the varying single burner heights ran over the full simulation time of 4,000 s. The dashed line indicates the experimental integrated heat flux averaged over all eight gauges.

6.5.3 Single and Multi-Burner vs Experiment

The heat flux comparison between the experiment, multi-burner, and single burner model are shown in figure 6.6. The single burner tends to have a higher heat flux than the multi-burner simulation, most likely due to the geometry of the multi-burner vehicle obstructing the view of the fire to the heat flux gauges. This is especially true for gauges 1 and 4, which are located at the front and rear of the vehicle respectively. However, both are less than the experimental data. The single burner had a 25% times greater integrated heat flux on average compared to the multi-burner simulation. Nevertheless, the single burner does not have better agreement to the experimental heat flux than the multi-burner. This is demonstrated in figure 6.7, which plots the integrated heat flux difference for the 8 gauges versus gauge position. The single burner simulation has better agreement for the right heat flux gauges, figure 6.7a, while the multi-burner simulation has better agreement for the left heat flux gauges, figure 6.7b. Both simulations (multi and single burner) compared to the experiment show an expected finding that gauges 2 and 3 produce less of a difference when measuring heat flux than gauges 1 and 4. The first and last gauges are simply not as centered to the body of the vehicle as the middle two gauges, thus they have a worse “view” of the fire.

Observing figure 6.7, and 6.6 further it is clear the single burner heat flux measurements begin to approach the multi-burner simulation values; i.e. the green line begins to approach the orange line from front to back. Figure 6.6 individually plots the heat flux gauges and 6.6a shows the single burner further from the simulation the subsequent gauges, 6.6b, 6.6c, 6.6d. The difference between the single burner and multi-burner heat flux as the measurement shifts from heat flux gauge to heat flux gauge is detailed in figure 6.7. The downward trend in difference

is easily seen in this figure, thus proving how the single burner simulation begins to approach the multi-burner model towards the back of the vehicle. The smallest difference is in gauge 3 followed shortly by gauge 4. Again most likely due to the inherent obstructions created by the multi-burner vehicle geometry.

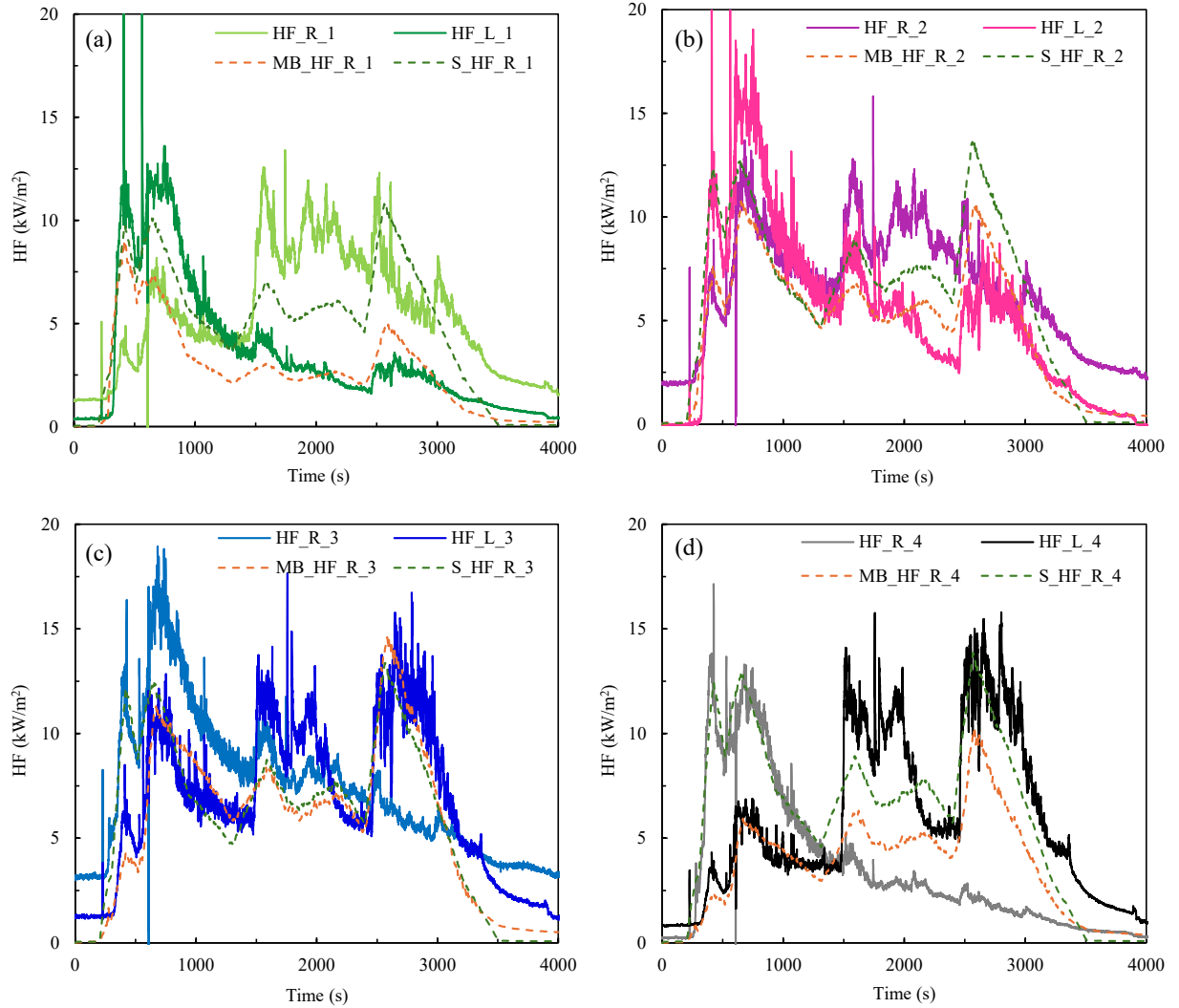


Figure 6.6: Figures (a)-(d) show the experimental heat flux plotted as like colors per gauge (i.e. green and light green is gauge 1, pink and purple is gauge 2, etc.). Meanwhile, the simulation with multiple burners is shown in as dashed orange curve to contrast the experimental data. While the single burner simulation is shown as a dashed green curve.

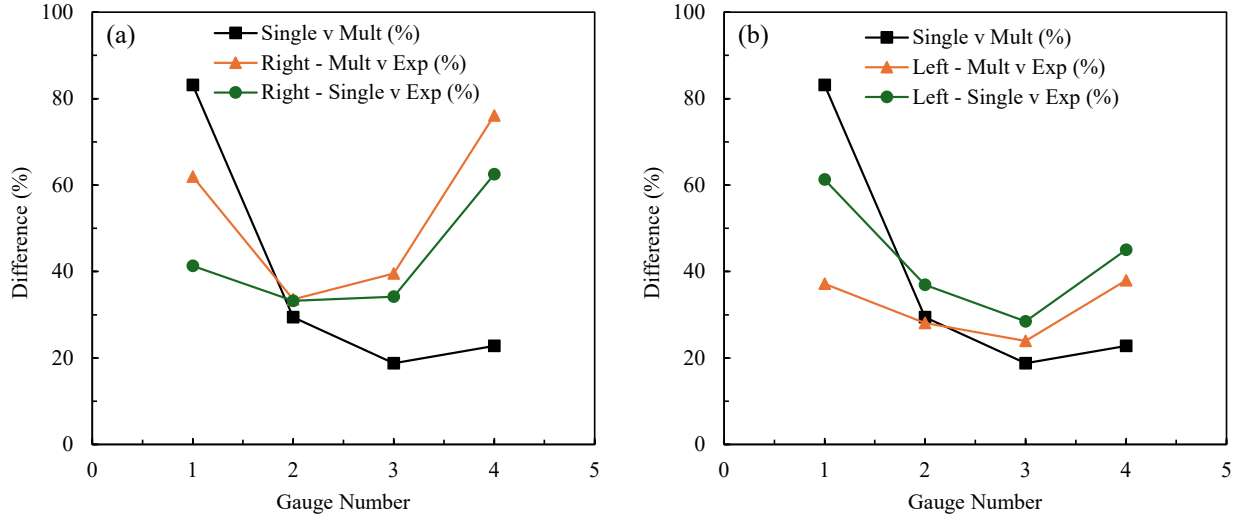


Figure 6.7: The gauges along the panel, i.e. gauge 1, 2, 3, and 4, are individually compared to gauge location to see how accuracy changes. Figure (a) demonstrates the right experimental gauges and figure (b) shows the left experimental gauges. The single versus multi-burner is shown in both figures (a) and (b) as a reference.

6.6 Heat Flux Impingement on Ceiling

Figure 6.8 and 6.9 demonstrate the effect of net ceiling heat flux with the multi-burner configuration as opposed to the single burner. Net heat flux was chosen for the still images (figures 6.8 and 6.9) since it provides a comprehensive view of the heat transfer to the entire ceiling including the hot gases. Although in the previous sections it was discussed that the heat flux from the vehicle to laterally distant surfaces remained similar in both cases it is obvious the geometry has a greater impact when looking at the heat flux above the vehicle. Figure 6.8a shows a small area with a heat flux of about $12 \text{ kW} \cdot \text{m}^{-2}$ with limited heat flux on the remaining area surrounding the vehicle. In contrast, figure 6.8b, the single burner model, is showing some flame impingement on the ceiling with a net maximum heat flux of $36 \text{ kW} \cdot \text{m}^{-2}$ with the surrounding area having a heat flux of approximately $30 \text{ kW} \cdot \text{m}^{-2}$. It is clear the geometry of the vehicle

is obstructing the flames from reaching the ceiling or even demonstrating significant burning outside of the vehicle. The opposite effect happened in figure 6.9 where figure 6.9a, which has vehicle geometry, has flame impingement upon the ceiling with a net heat flux of $60 \text{ kW} \cdot \text{m}^{-2}$. However, the single burner, figure 6.9b, had a slightly more dispersed heat flux, which did not impinge on the ceiling and had a heat flux of about $30 \text{ kW} \cdot \text{m}^{-2}$ or lower. Both figures 6.8 and 6.9 were chosen at representative times to highlight the differences in flame behavior and heat flux to the ceiling due to the vehicle geometry.

In addition to the still images of figures 6.8 and 6.9 a plot of the gauge heat flux at the ceiling is shown in figure 6.10 comparing the single and multi-burner models. The heat flux gauges were placed on the ceiling on the vehicle's center-line, with 3 gauges placed in the closest center corresponding location of the hood, cabin, and rear sections of the vehicle. The temporal behavior of ceiling heat flux is comparable to the lateral heat flux curves shown in figure 6.6. However, the scale is significantly larger with peak heat fluxes reaching between 80 and $90 \text{ kW} \cdot \text{m}^{-2}$ whereas the lateral heat fluxes were around $20 \text{ kW} \cdot \text{m}^{-2}$. What greatly differs between the single and multi-burner ceiling geometry is the comparison between the gauges in the same location. The heat flux gauge above the hood in the multi-burner model, figure 6.10a, had a significant rise and fall peaking at about $90 \text{ kW} \cdot \text{m}^{-2}$ in a span of 75 s with the remaining heat flux between 5 and $10 \text{ kW} \cdot \text{m}^{-2}$. Conversely, the same gauge location showed only peaks of $30 - 35 \text{ kW} \cdot \text{m}^{-2}$ peaks during the single burner simulation. This difference is due to the flame impingement on the ceiling from the hood burner as seen in figure 6.9. Figure 6.10b shows the single burner model consistently higher than the multi-burner, likely because the single burner develops a hotter plume in the center as compared to the dispersed plume created by the multiple burner areas. The rear ceiling heat flux device, figure 6.10c, had smaller peaks than both the hood

and ceiling devices measured as seen in figures 6.10a and b. However, it can be observed that when the rear of the vehicle ignites, around 1,250 s, the multi-burner model, as with the hood area, estimates increased heat flux as compared with the single burner approach. Unfortunately, no experimental data is available to compare so it cannot be concluded whether the single or multi-burner geometry is a better representation. However, it is clear with the difference in ceiling heat flux between the two that these geometries may not be interchangeable for the ceiling heat flux applications.

Figures 6.8, 6.9, and 6.10 demonstrate how the geometry and/or location of the burner can affect the heat flux experienced by the ceiling above a burning vehicle. As seen and discussed in section 6.5.3, there was little difference between the single burner and the multi-burner when comparing the lateral heat flux gauges. Which would allow for a vehicle to be simplified as a single burner fire. However, simplifying the vehicle to a single burner may ignore important temporal heat flux exposure to the ceiling based on fire spread. Given that this was a preliminary test considering the implementation of a ceiling, more rigorous heat flux measurements from the ceiling are needed to draw a definitive conclusion.

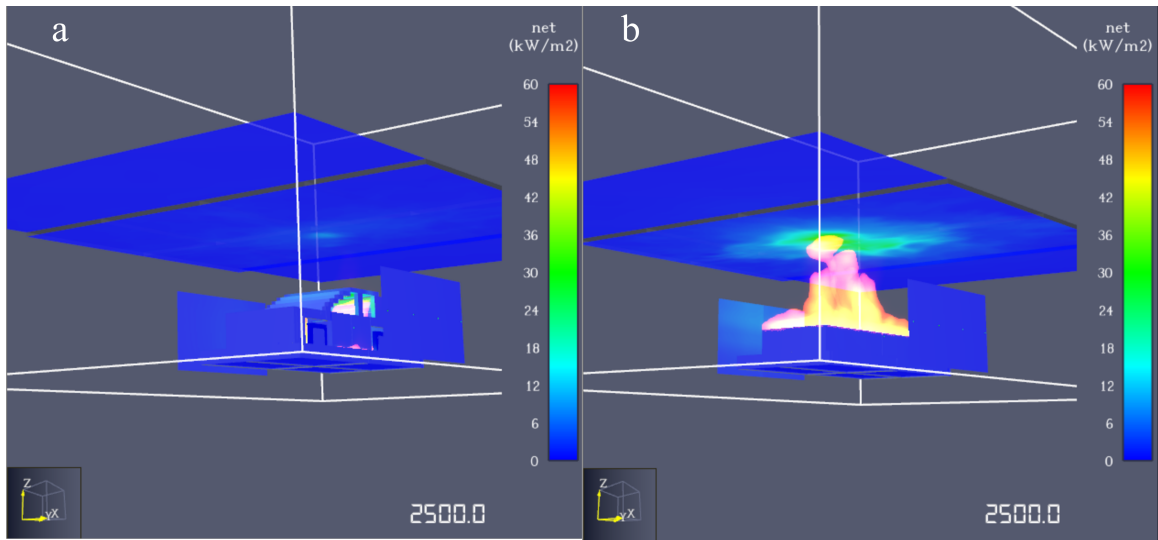


Figure 6.8: Figure (a) shows the net heat flux at the ceiling for the multi-burner vehicle. Figure (b) shows the ceiling heat flux of the single burner vehicle. The comparison is taken at 2500 s to demonstrate the varying ceiling impingement based on geometry.

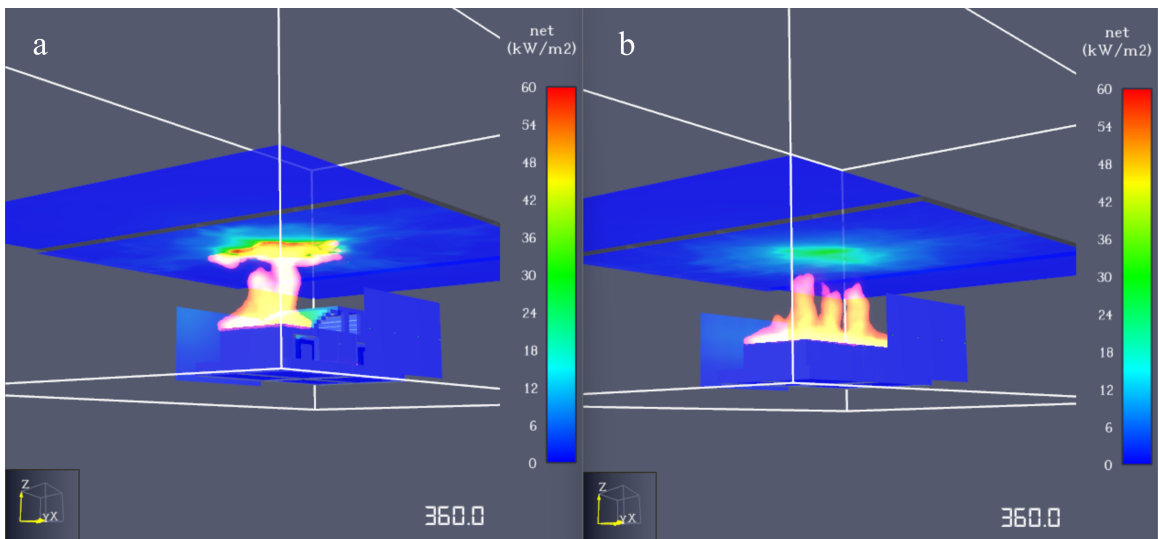


Figure 6.9: Figure (a) shows the flame impingement and net heat flux at the ceiling for the multi-burner vehicle. Figure (b) shows the ceiling heat flux of the single burner vehicle. The comparison is taken at 360 s to demonstrate the varying ceiling impingement based on geometry.

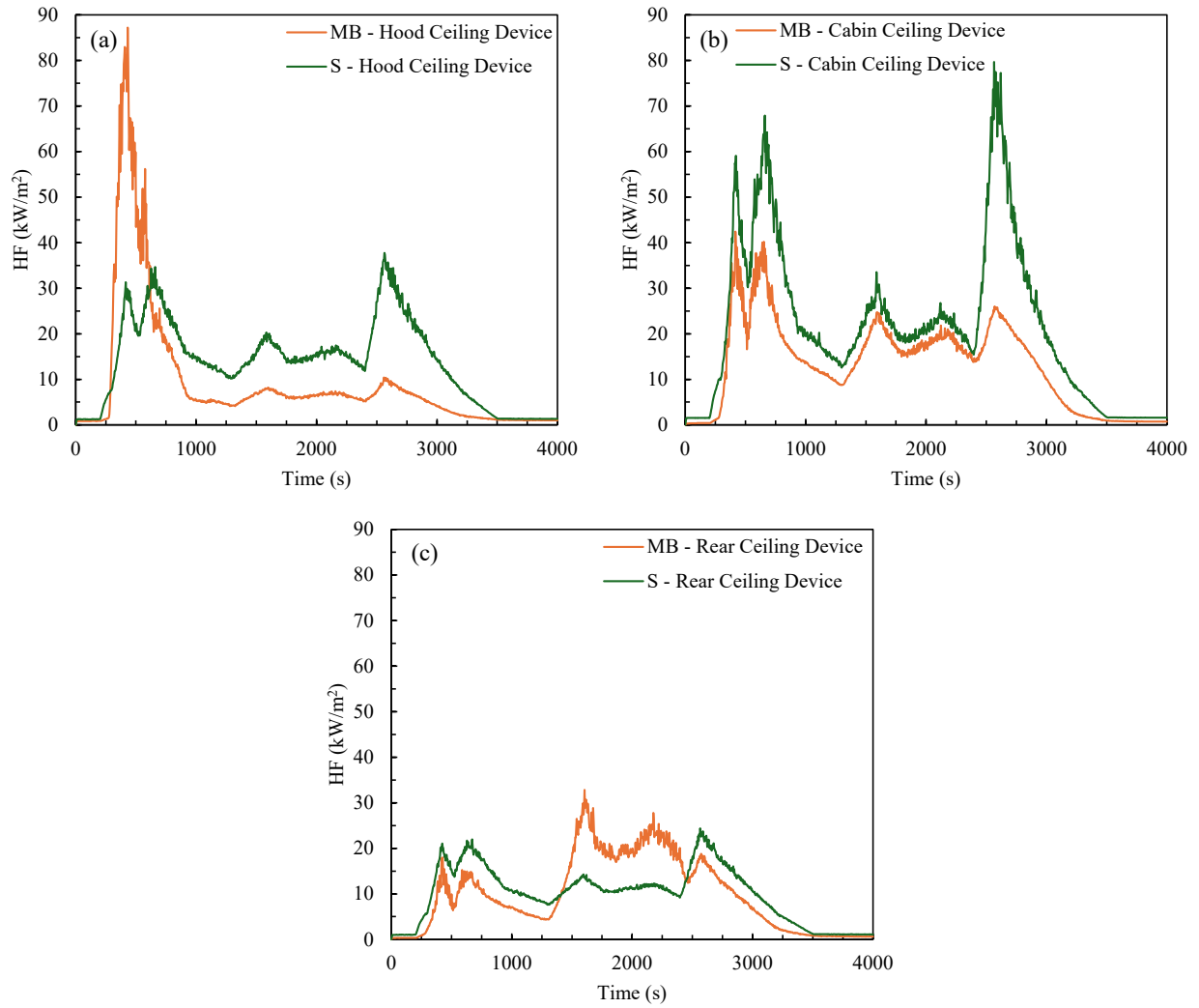


Figure 6.10: Figures (a) - (c) compare the gauge heat flux at the ceiling for the multi-burner vehicle and the simplified single burner representation. Heat flux gauges were included on the ceiling at the corresponding vehicle's hood, ceiling, and rear locations.

Chapter 7: Conclusions

The scope of this project was to document a methodology and recommendations for fire protection engineers to use when simulating the heat flux exposure from a vehicle fire to nearby targets with Fire Dynamics Simulator (FDS), and to demonstrate how closely the simulated heat fluxes matched the measured heat fluxes from UL FSRI's free burn data of an EV Hyundai Kona. This approach was chosen due to the limited amount of information and guidance regarding the specification of FDS input parameters for vehicle fires and the benefits that modeling can provide when researching EV fires as discussed in the literature review chapter 2. This work was motivated by the growing prevalence of EVs and the perceived unique hazards they present, including thermal runaway, prolonged burning, and complex heat transfer patterns. Meanwhile the issue is heightened with the scarcity of standardized modeling practices and transparent parameter selection in existing literature. By bridging these gaps, the study aimed to provide practitioners with a framework that balances a modeling methodology with recommendations and validation of experimental data.

The modeling process showed that selection of input parameters significantly impacts simulated heat fluxes and detailed geometric representation of the vehicle does not significantly improve the accuracy of the model output as compared with the experimental measurements. Constructing the Hyundai Kona EV model with multiple burners provided means for the sim-

ulation to capture the fire progression and its influence on heat flux measurements. This level of detail may be beneficial when assessing vertical heat transfer, such as ceiling impingement, where geometry altered the thermal environment at the ceiling. Conversely, when the objective was limited to lateral heat flux prediction, a simplified single burner vehicle representation performed comparably to the multi-burner model. Thus, providing an alternative for fire protection engineers constrained by time or computational resources. Hence, model complexity should be driven by the performance based design objectives and time constraints rather than perceived optimal performance.

Estimation of the heat flux through a FDS model showed differences in measured versus simulated heat fluxes even though the energy input to the model via the MLR was based on the measured MLR, with no errors in simulating the MLR. The comparability of the total delivered flux at the gauges between experimental and computational runs resulted in an integrated heat flux difference of about 40% (see equation 5.6) for the single and multi-burner simulations, even after optimizing input parameters. Greater differences existed when comparing individual locations for simulated and measured heat flux. The more comparable gauges were numbers 2 and 3 which viewed the middle of the vehicle geometry where intense burn is to be expected. The left and right gauges indicated how prevalent tilt was within the experiments and how the simulation did not capture these flame characteristics, thus producing a larger overall difference between experiment and simulation. Hence, if model validation to experimentation is sought the flame shape and tilt is of utmost importance.

Some parameter adjustments significantly impacted the difference accuracy, but the model is not as sensitive to others. Parameters with the most change to heat flux were the radiative fraction, number of radiation angle, and the single or multi-burner geometry. Less impactful pa-

rameters were the soot yield, and SUPPRESSION= 'FALSE' or EXTINCTION_MODEL= 'EXTINCTION 2' . For example, the radiative fraction demonstrated significant impact on the heat flux with a radiative fraction of 0.7 being 2.5 times larger than a radiative fraction of 0.2. Conversely, the soot yield variation exerted minimal change on the integrated heat flux. The differences in the effects of these two parameters on the heat flux highlights the importance of prioritizing parameters with meaningful affect while avoiding unnecessary tuning which may add complexity without a clear benefit.

The findings from this research also point to areas of simplification that are acceptable within certain specifications. If a lateral heat flux assessment is the priority, a single burner fire calibrated to experimental mass loss data can deliver results within the same margin of error as a multi-burner geometry, allowing the simplification of a vehicle fire to a single burner. However, if the evaluation is reliant on the ceiling effects or structural integrity, the single burner may ignore important temporal heat flux exposure to the ceiling based on fire spread and the multi-burner configuration may portray more realistic outcomes. Undoubtedly, this specific scenario requires further research since experimental heat flux data was not available at the ceiling. Rather, what is emphasized is the difference between the curves of either model.

This research investigated simple and moderately complex approaches for EV modeling and provides practical information that used to assist in future EV modeling efforts, but continued research should still be explored.

Some topics include expanding the material assignments of the vehicle, refining the fuel surrogates, and integrating lithium-ion battery modeling to capture phenomena such as thermal runaway and jetting. Additional studies on ceiling configuration and ignition scenarios will further the applicability of EV fire models into diverse build environments. Ultimately, the intent

is not simply to mirror experimental results but to provide modeling outputs that contribute to informed performance based design decision making.

By understanding EV fire behavior and providing a structured approach for simulation, this work contributes to the development of standardized practices in EV modeling. Such that modelers may apply these principles and recommendations to better their work and understanding of vehicle fire modeling. Thereby expanding the applicability and capability of EV modeling.

Chapter 8: Recommendations

Chapters 4, and 6 have specified how and why certain parameters were chosen and what their impact on the model. This structure facilitates the discussion of recommendations beyond this project or to those seeking insight to replicate a similar model. This chapter will provide recommendations for modeling vehicles broken into sections based on the type of recommendation.

8.1 Choosing a Grid Size

The grid size is the ultimate building block to the rest of the model. Equation A.1 can provide a first order estimate for a grid size of course, medium, or fine resolution. However, it should not be the only consideration for grid size. Due to the detail in vehicle geometry, maintaining the integrity of a “car shape” is key to burner dimension delineation, and what feature of vehicle burning is being observed. For example, wheel burning, or the inclusion of jetting may require a finer mesh than a vehicle hood fire. To approximate grid size, equation A.1 and the geometry or application of the model should be taken into account to determine an adequate size. Ultimately, various grid sizes should be tested for convergence as discussed in section A.1 to ensure the optimum grid size is chosen.

8.2 Vehicle Construction

To create the vehicle it is recommended to use PyroSim rather than a text editor due to its visual interface and tool box. This facilitates construction as there is immediate feedback of the shapes dimension, location, and proximity to other obstructions. A sketch of the vehicle being modeled is highly recommended to ensure the vehicle elements are of the right dimension, i.e. the doors, windows, hood, and body are all proportionally dimensioned with respect to the other. A sketch of the Hyundai Kona vehicle used for this project with scaled dimensions can be found in section 3.3, figure 3.5. FDS has a modular form of construction which does not allow the curved features found in vehicles. Hence, any round or curved shape should be approximated as a rectangle to the nearest grid cell. If an obstruction is not approximated to a grid cell FDS will do so automatically, thus slightly altering the element. This will cause errors with any calculations that were completed assuming different dimensions. Simply put, when measuring and constructing the vehicle elements ensure they align with the grid cells visible in PyroSim.

The order of construction is not particularly relevant, but it is best to construct the vehicle chassis (i.e. walls of the vehicle), rather than attempting to recreate them using a combination of solid obstructions and holes. The vehicle chassis construction provides the benefit of having an interior area of the vehicle, which is more accurate to reality, allows for more customization of burner placement and size, as well as allowing for heat transfer through the vehicle walls. Additionally, as discussed in section 4.2.1, if the obstruction is too thick in comparison to the defined surface thickness heat will not conduct through that element. Which reduces the amount of heat flux exposure any near by object would receive.

If a single burner model is sought, as opposed to a multi-burner geometry, the vehicle construction significantly simplifies. Since this option requires no geometry only the length and width of the vehicle is needed. As for the height of the burner, section 6.5.2 examines the optimal burner height for the case studied. It was found that using half the vehicle height produced the best agreement with the experimental data. Most importantly, as stated in the paragraph above, the dimensions of the burner should align with the grid cells to prevent calculation approximations and errors.

8.3 FDS Input Parameter

Chapter 4 has described how the FDS model was constructed and the parameter selection process. Many of these were described using mathematical equations and engineering judgment to best match the experimental data set. These parameters are not exhaustive but provide a basis for a basic vehicle fire simulation. For this reason, the parameters used have been listed in table 8.1, such that the reasoning behind their choice is clear and for further research to determine their impact or the need for further parametric study. The parameter column describes the FDS prompt, the value is the numeric or categorical setting, justification encapsulates why and how the value was chosen, and reference provides the supporting literature/section to the justification.

8.3.1 Final Input Parameter Table

Table 8.1: Model Inputs and Parameters

Parameter	Value	Justification	Reference
Grid Size	10 cm	Obstruction dimension, characteristic fire diameter	§ A.1, 4.1.4
Pyrolysis Method	Prescribed Burning	Greater control for estimation of experimental results	§ 4.2.3
Fuel Species	$C_3H_8O_{1.55}$	Adjusted for correct EPUMO ₂	§ 4.2.3.1
Heat of Combustion	25 MJ·kg ⁻¹	Literature review based	[2, 9, 10]
Number of Radiation Angles	400	Diminishing returns, see figure 6.3	§ 6.3, [42]
Humidity	45 %	Local weather conditions	[40]
Flame Extinction	SUPPRESION= ' . F . '	No heat flux benefit in a different extinction model	§ 6.4, [28]
Path Length	2 m	Distance from outer vehicle to heat flux gauges	§ 4.2.4.2
Burner Surface Temperature	300 °C	Average ignition temperature polymer's in vehicle	[39]
CO Yield	0.04	Average CO yield of ignitable polymer's in vehicle	[26]
Soot Yield	0.015	Average soot yield of ignitable polymer's in vehicle	[17]
Radiative Fraction	0.35	Common for hydrocarbons, literature review based	[7, 17]

8.4 Choosing Multi-Burner or Single Burner

Section 6.5.3, demonstrated how the single burner model compared to the multi-burner setup which included geometry. The difference between the two models was found to be 33% with the single burner, on average, producing higher heat fluxes than the multi-burner. When compared to the measured experimental heat fluxes, the multi-burner and single burner had the same difference of 40%.

The differences in the burner specification approaches are more pronounced once the heat flux at the ceiling. Section 6.6, explains the differences in heat flux between the single and multi-burner model, and are depicted in figures 6.8 and 6.9. The critical conclusion being that the ceiling heat fluxes are less comparable between the single and multi-burner model as they were with the lateral heat fluxes. The multi-burner approach suggests that there are potentially higher heat fluxes on the ceiling due to localized plume impingement during flame spread, which the single burner may not capture since the plume is dispersed over a larger burner area. Further analysis between the single and multi-burner ceiling heat flux is required if the single burner is to be used in place of the multi-burner.

Chapter 9: Future Work

In the process of creating the methodology for simulating an EV fire with FDS, it became clear the construction of the vehicle (including burner placement), the parameters required, and the difference between a single and multi-burner model were of utmost importance. Other applications, and avenues of research emerged the more the project evolved, for example parametric parameters, detailed vehicle components, lithium-ion battery inclusion, and alternating forms of fire spread. These topics however fell beyond the scope of the project, so they are discussed here.

9.1 Parameters

FDS input parameters were altered in order to achieve the value which caused the heat flux to best fit to the experimental data. The parameters which were investigated were the soot yield - §4.4.1, radiative fraction - §4.4.2, the defined combustion model (Suppression = False versus Extinction 2) - §4.4.4, and variations on vehicle geometry - §6.5.3, §6.5.2. However, as shown in table 8.1, many other parameters were also used. Hence, to further the methodology of FDS vehicle fire construction additional parameters should be altered with an emphasis on completing a cohesive sensitivity analysis on the parameters and their effect on the heat produced from combustion. Some parameters which could be adjusted include wind flow and the method of combustion. These two parameters in particular may have significant impact on the mass loss rate

and heat flux curves of the vehicle. Certain parameters in this study remained fixed, these could be continued into a parametric analysis such that their direct impact on the model is evaluated. These parameters include the CO yield and background CO₂ concentration, the `PATH_LENGTH`, and the initial temperature of the burners `TMP_FRONT`. Another way to further the vehicle model is to improve the means of flame discretizations from video evidence and measurements, such that enhanced detail can be included in the FDS prescribed burner model.

9.2 Materials

The materials included in the FDS simulation inputs are the base components found in a vehicle, such as steel, rubber, and the “fuel” used to emulate a polyvinyl chloride. Vehicles are constructed of significantly more materials which are often layered or in composites. The material selections impact the heat transfer throughout the vehicle, and the decision about how to represent all material burning as single combustion reaction. More research on vehicle materials, their quantities, and location within the vehicle should be researched to determine the impact on heat transfer through vehicle components. This avenue of detail may not be practical for practicing engineers but could be of interest to researches expanding on the accuracy of the multi-burner heat flux in comparison to experimental heat fluxes.

9.3 Fuels

A base “fuel” was chosen to represent numerous combustible materials, including the lithium-ion battery. The rationale for the chosen fuel is discussed in section [4.2.3.1](#). However, other permutations could be explored further to determine if better representations are possible.

Additionally, the lithium-ion battery in particular should be further approximated to a fuel. Doing so would result in a model setup more akin to a real electric vehicle. The most complete EV representation would include fully modeling the lithium-ion battery to investigate the disparity in measured and simulated heat flux when the battery was in thermal runaway. This would significantly increase the complexity of the model construction but is expected to produce higher heat fluxes and better approximate battery-induced effect. Any unique flaming effects or lower heat flux from the models output could be due to battery jetting or unexpected flaming occurring beyond the vehicle, as opposed to the current simulation which primarily maintains flaming within the vehicle geometry. Some have modeled the lithium-ion battery such as [8], however, this does not include the batteries implementation into the vehicle geometry which would be an important research avenue.

9.4 Ceiling Configuration

The primary focus of this project is to compare simulated heat fluxes to the heat flux measured in experiments. This evaluation is important for determining how well the simulated heat fluxes represent real thermal exposures when conducting a thermal hazard assessment. However, the structure in which the vehicle is found is also of concern. Often these are parking garages or home garages. Parking garages are considered to pose a greater threat due to their size and vehicle load capacity, and fire protection engineers can meaningfully impact their design for fire safety. Hence, estimating the heat flux on the ceiling is important for studying the structural integrity of the parking garage. Since the experimental data provided by FSRI did not include heat flux measurements above the vehicle, or include a ceiling further free burn experiments should be

conducted to capture ceiling effects. Section 6.6 discussed the effect of including a ceiling into the model, the potential heat fluxes the ceiling might experience, and the differences in that heat flux based on using a single or multi-burner model. However, the ceiling simulation completed here was preliminary and included only a horizontal ceiling configuration with no obstructions. Further simulations should be completed on various ceiling configurations including horizontal, slanted, and a double-tee concrete structure. Further measurement analysis such as temperature and additional heat flux gauges should be included such that comparison to experimental data is possible. Due to the lack of experimental data for the ceiling heat flux analysis only the temporal differences in heat flux with a single or multi-burner configuration are demonstrated.

9.5 Ignition Scenarios

The project aimed to mimic the progression of flame spread through the vehicle as seen in the FSRI vehicle experiment studied in this thesis. However, data is often not available to study specific vehicle fire scenarios. Fire protection engineers are often asked to model performance based design models for a nonexistent scenario. In these situations, the ignition location and direction(s) of fire spread is unknown. The model presented here can be used to further research ignition scenarios and provide adjustability to a performance-based design. Additional full-scale fire experiments based on ignition scenarios would be of key importance to fire modeling such that simulations could alter the order of burner ignition and intensity. Ultimately, the mass loss rate curve and subsequent heat release rate curve would change providing more vehicle burning characteristics for fire protection engineers.

The project aimed to mimic the progression of flame spread through the vehicle as seen in

the FSRI vehicle experiment studied in this thesis. However, data is often not available to study specific vehicle fire scenarios. Fire protection engineers are often asked to model performance based design models for a nonexistent scenario. In these situations, the ignition location and direction(s) of fire spread is unknown. The model presented here can be used in these situations. Further research into ignition scenarios could provide adjustability to a performance-based design. This includes altering in which order the burners ignite and what intensity. Ultimately, the mass loss rate curve and subsequent heat release rate curve would change providing more vehicle burning characteristics for fire protection engineers.

Appendix A: Appendix

A.1 Computational Mesh

The domain size is, typically, about 5 times larger than the size of the object being model. Upon running a simple burner with this domain it was clear the y-directional depth could be shortened to reduce computational cost as the extremes were sufficiently far from the combustion. After this adjustment the final domain was specified as $10.4 \text{ m} \times 16 \text{ m} \times 8 \text{ m}$. The grid size has two distinct areas, the “near field” where combustion occurs, and the “far field” away from the combustion area. While it is desirable to have a fine resolution everywhere in the domain, it is often impractical based on the computing resources available. Typical design practice is to use a fine resolution in the “near field” and a coarser resolution in the “far field”. This approach balances the need for more accurate predictions in the regions where complex calculations are taking place with available computational resources. The “near field” was determined using a combination of a mathematical and practicality approach. Equation A.1, [28] is used to determine the characteristic fire diameter, D^* , to then obtain the grid size,

$$D^* = \left(\frac{\dot{Q}}{\rho_\infty C_p T_\infty \sqrt{g}} \right)^{2/5}, \quad (\text{A.1})$$

where $\dot{Q}$ is the total heat release rate of the fire (kW), ρ_∞ is the density of air ($\text{kg}\cdot\text{m}^{-3}$), C_p is the specific heat of air ($\text{kJ}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$), g is the gravitational constant ($\text{m}\cdot\text{s}^{-2}$), T_∞ is the ambient temperature (K). $\dot{Q}$ may differ based on the simulation scope, for this project the “smallest unit of fire” is more relevant as modeling the fire characteristics is important. To capture the “smallest unit of fire”, which the model would begin simulating, the HRR when the fire began to grow more rapidly was selected, resulting in a $\dot{Q}$ of 300 kW. To determine the cell size D^* is divided by dx (the cell size), and set to the arbitrary values of 4, 10, and 16, where the mesh is course, medium, and fine respectively. D^*/dx can be thought of as the number of computational cells spanning the characteristic fire. With D^*/dx set to 4, 10, and 16, D^* can be solved for and substituted into equation A.1, allowing the cell size, dx to be resolved. This yielded cell sizes of 14.8 cm, 5.9 cm, and 3.7 cm respectively. Due to restrictions on vehicle dimensions and accurate burner representation, these cell sizes were adjusted to integer values that are divisible to the vehicle and burner dimensions. These potential cell sizes now become 10 cm, 5 cm, and 2.5 cm.

The three potential grid sizes were simulated to test for stability (minimal change in integrated heat flux) and running time. Due to the length of time for a 2.5 cm grid size to fully run, only the first peak was used as a comparison, which allowed for about 500 s of simulation running time. To analytically check for convergence, the integral of the heat flux as a function of grid size is plotted in A.1. The four right gauges are shown along with their average value. It should be noted that the simulations right and left heat flux is symmetric. Comparing grid sizes of 2.5 cm and 5 cm there is a minimal decrease in the integrated heat flux that on average corresponds to $29.3 \text{ kJ}\cdot\text{m}^{-2}$, or 4.4%. Looking at the difference between a 5 cm grid and a 10 cm grid there is an increase of $59.6 \text{ kJ}\cdot\text{m}^{-2}$ or 9.4%. This increase is primarily due to gauges 1 and 2 as opposed to 3 and 4 which show a negligible change. Based purely on stability the simulations performed

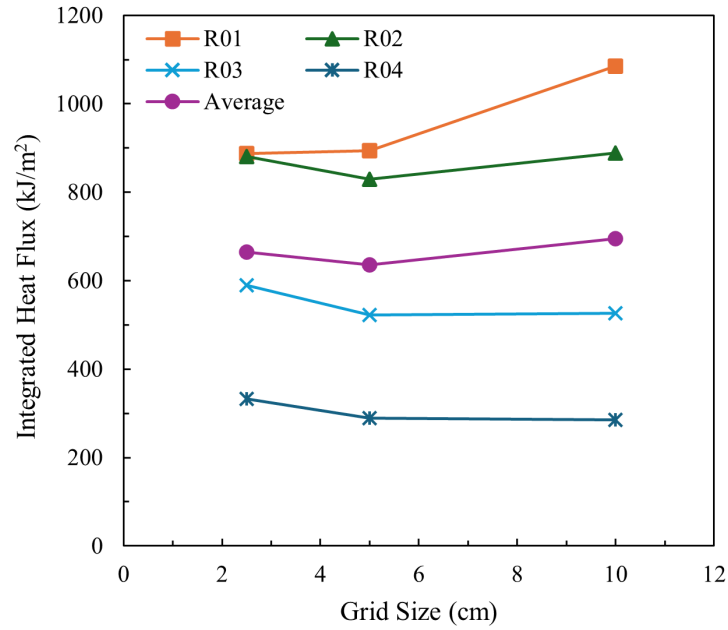


Figure A.1: Grid study of the three mesh sizes tested.

well, with the exception of R01 which increased 18 % over the three tested grid sizes. Note R01 observed the fire at the most oblique angle.

The other major consideration is running time. For a full run of 4000 s when the simulations were run on 18 cores, the 2.5 cm grid size would take over 1 year to complete. A 5 cm grid size takes 6 weeks to complete, and a 10 cm grid size takes 2.5 days. Given the significant amount of time that both grid sizes 2.5 cm and 5 cm take to run and the overall minimal changes to integrated heat flux, a grid size of 10 cm was deemed the most practical option for this thesis. However, the 5 cm resolution was better converged and would be more appropriate in a design application where the magnitude of the quantitative predictions is important.

Appendix B: FDS Code

Kona_121.fds

Generated by PyroSim 2024.2.1120

Jul 24, 2025, 4:16:26 PM

```
&HEAD CHID='Kona_121'!/&HEAD signifies the start of the file
&TIME T_BEGIN=0, T_END=4000.0!/&Time how long file is running for
&DUMP DT_RESTART=300.0!/the .restart file is at 300s increments
&COMB SUPPRESSION=.FALSE./
```

```
!Mesh Definition, FYI size of a grid cell,
!IJK is the number of grid cells in each direction,
!XB is the coordinate definition
&MESH ID='Mesh1', IJK=24,40,40, XB=-0.4,2.0,7.0,11.0,0.0,4.0/
&MESH ID='Mesh2', IJK=28,40,40, XB=2.0,4.8,7.0,11.0,0.0,4.0/
&MESH ID='Mesh3', IJK=24,40,40, XB=-0.4,2.0,7.0,11.0,4.0,8.0/
&MESH ID='Mesh4', IJK=28,40,40, XB=2.0,4.8,7.0,11.0,4.0,8.0/
&MESH ID='Mesh5', IJK=24,40,40, XB=-0.4,2.0,11.0,15.0,0.0,4.0/
&MESH ID='Mesh6', IJK=28,40,40, XB=2.0,4.8,11.0,15.0,0.0,4.0/
&MESH ID='Mesh7', IJK=24,40,40, XB=-0.4,2.0,11.0,15.0,4.0,8.0/
&MESH ID='Mesh8', IJK=28,40,40, XB=2.0,4.8,11.0,15.0,4.0,8.0/
&MESH ID='Mesh9', IJK=26,40,40, XB=4.8,7.4,7.0,11.0,0.0,4.0/
&MESH ID='Mesh10', IJK=26,40,40, XB=7.4,10.0,7.0,11.0,0.0,4.0/
&MESH ID='Mesh11', IJK=26,40,40, XB=4.8,7.4,7.0,11.0,4.0,8.0/
&MESH ID='Mesh12', IJK=26,40,40, XB=7.4,10.0,7.0,11.0,4.0,8.0/
&MESH ID='Mesh13', IJK=26,40,40, XB=4.8,7.4,11.0,15.0,0.0,4.0/
&MESH ID='Mesh14', IJK=26,40,40, XB=7.4,10.0,11.0,15.0,0.0,4.0/
&MESH ID='Mesh15', IJK=26,40,40, XB=4.8,7.4,11.0,15.0,4.0,8.0/
&MESH ID='Mesh16', IJK=26,40,40, XB=7.4,10.0,11.0,15.0,4.0,8.0/
&MESH ID='Mesh17', IJK=52,20,40, XB=-0.4,10.0,15.0,19.0,0.0,8.0/
&MESH ID='Mesh18', IJK=52,20,40, XB=-0.4,10.0,3.0,7.0,0.0,8.0/
```

```
&RADI NUMBER_RADIATION_ANGLES=400, PATH_LENGTH=2/
&MISC HUMIDITY=45.0/
```

```
&SPEC ID='FUEL', H=8, C=3, O=1.55!/Defining a fuel, &SPEC refers to species
```

```
!Defining the reactions for specific fuels and what yields they may have
&REAC ID='Reaction1',
    FUEL='FUEL',
    CO_YIELD=0.04,
```

SOOT_YIELD=0.015,
HEAT_OF_COMBUSTION=2.5E+4/

!Mass flux devices placed on the burner, Quantity is what it is measuring
!SPEC_ID is which species it is looking for/measuring the mass flow of
!Spatial_Statistics is over what bounds the measurement is taking place,
! the corrdinate that stays the same is the surfaces
!while the other two are the bounds of integration see ex, on line below
&DEVC ID='HOOD', QUANTITY='MASS FLUX', SPEC_ID='FUEL',
SPATIAL_STATISTIC='SURFACE INTEGRAL', XB=4.0,5.8,8.8,9.8,1.5,1.5/
&DEVC ID='UNDER CAR', QUANTITY='MASS FLUX', SPEC_ID='FUEL',
SPATIAL_STATISTIC='SURFACE INTEGRAL', XB=4.0,5.8,9.8,12.2,0.7,0.7/
&DEVC ID='UNDER CAR TRUNK', QUANTITY='MASS FLUX', SPEC_ID='FUEL',
SPATIAL_STATISTIC='SURFACE INTEGRAL', XB=4.0,5.8,12.2,12.8,0.8,0.8/
&DEVC ID='CAR FLOOR', QUANTITY='MASS FLUX', SPEC_ID='FUEL',
SPATIAL_STATISTIC='SURFACE INTEGRAL', XB=4.0,5.8,9.8,12.2,0.8,0.8/
&DEVC ID='TRUNK VERTICAL', QUANTITY='MASS FLUX', SPEC_ID='FUEL',
SPATIAL_STATISTIC='SURFACE INTEGRAL', XB=4.0,5.8,13.0,13.0,0.8,1.5/
&DEVC ID='TRUNK HORIZONTAL', QUANTITY='MASS FLUX', SPEC_ID='FUEL',
SPATIAL_STATISTIC='SURFACE INTEGRAL', XB=4.0,5.8,12.8,13.0,1.5,1.5/

!HFG's placed on the panels
!Quantity is what it is measuring,
!XYZ defines the location of the device
!IOR is the orientation of the device relative to the surface
!it is attached to (1 = +X, -1 = -X, 2 = +Y, -2 = -Y, 3 = +Z, -3 = -Z)
&DEVC ID='S HF_RAD_WALL.R01', QUANTITY='GAUGE HEAT FLUX',
XYZ=2.1568,9.44,1.3, IOR=1/
&DEVC ID='S HF_RAD_WALL.R02', QUANTITY='GAUGE HEAT FLUX',
XYZ=2.1568,10.48,1.3, IOR=1/
&DEVC ID='S HF_RAD_WALL.R03', QUANTITY='GAUGE HEAT FLUX',
XYZ=2.1568,11.52,1.3, IOR=1/
&DEVC ID='S HF_RAD_WALL.R04', QUANTITY='GAUGE HEAT FLUX',
XYZ=2.1568,12.56,1.3, IOR=1/
&DEVC ID='S HF_RAD_WALL.L01', QUANTITY='GAUGE HEAT FLUX',
XYZ=7.6432,9.44,1.3, IOR=-1/
&DEVC ID='S HF_RAD_WALL.L02', QUANTITY='GAUGE HEAT FLUX',
XYZ=7.6432,10.48,1.3, IOR=-1/
&DEVC ID='S HF_RAD_WALL.L03', QUANTITY='GAUGE HEAT FLUX',
XYZ=7.6432,11.52,1.3, IOR=-1/
&DEVC ID='S HF_RAD_WALL.L04', QUANTITY='GAUGE HEAT FLUX',
XYZ=7.6432,12.56,1.3, IOR=-1/

!Wall temperature devices, look at wall, IOR to face wall
&DEVC ID='Right_Wall_Temp-Out', QUANTITY='WALL TEMPERATURE',
XYZ=5.85,11,1.2, IOR=1/
&DEVC ID='Left_Wall_Temp-Out', QUANTITY='WALL TEMPERATURE',
XYZ=3.95,11,1.2, IOR=-1/
&DEVC ID='Right_Wall_Temp-In', QUANTITY='WALL TEMPERATURE',
XYZ=5.65,11,1.2, IOR=-1/
&DEVC ID='Left_Wall_Temp-In', QUANTITY='WALL TEMPERATURE',
XYZ=4.15,11,1.2, IOR=1/

!&MATL is the definition of materials

```

&MATL ID='CONCRETE',
    FYI='NBSIR 88-3752 - ATF NIST Multi-Floor Validation',
    SPECIFIC_HEAT=1.04, !kJ/kgK
    CONDUCTIVITY=1.8, !W/mK
    DENSITY=2280.0/ !kg/m3
&MATL ID='Rubber',
    SPECIFIC_HEAT=2.15, !kJ/kgK
    CONDUCTIVITY=0.09, !W/mK
    DENSITY=910.0/ !kg/m3
&MATL ID='STEEL',
    FYI='Drysedale, Intro to Fire Dynamics -
        ATF NIST Multi-Floor Validation',
    SPECIFIC_HEAT=0.46, !kJ/kgK
    CONDUCTIVITY=45.8, !W/mK
    DENSITY=7850.0, !kg/m3
    EMISSIVITY=0.95/
&MATL ID='Aluminum',
    SPECIFIC_HEAT=0.9, !kJ/kgK
    CONDUCTIVITY=237.0, !W/mK
    DENSITY=2.7E+3, !kg/m3
    EMISSIVITY=0.96/
&MATL ID='Glass',
    SPECIFIC_HEAT=0.84, !kJ/kgK
    CONDUCTIVITY=1.05, !W/mK
    DENSITY=2.5E+3, !kg/m3
    EMISSIVITY=0.92
    ABSORPTION_COEFFICIENT = 0.5/

&SURF ID='CONCRETE',
    RGB=99,102,110,
    TRANSPARENCY=0.501961,
    MATL_ID(1,1)='CONCRETE', !the (1,1) refers
                                !to the (layer number, material index)
    MATL_MASS_FRACTION(1,1)=1.0,
    THICKNESS(1)=0.15/
&SURF ID='Tire',
    COLOR='BLACK',
    BACKING='INSULATED',
    MATL_ID(1,1)='Rubber',
    MATL_MASS_FRACTION(1,1)=1.0,
    THICKNESS(1)=4E-3/
&SURF ID='Steel',
    RGB=117,131,138,
    MATL_ID(1,1)='STEEL',
    MATL_MASS_FRACTION(1,1)=1.0,
    THICKNESS(1)=0.0012/
&SURF ID='Wheel Steel',
    RGB=117,131,138,
    MATL_ID(1,1)='STEEL',
    MATL_MASS_FRACTION(1,1)=1.0,
    THICKNESS(1)=3.0E-2/
&SURF ID='Pan Steel',
    RGB=117,131,138,
    MATL_ID(1,1)='STEEL',

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MATL_MASS_FRACTION(1,1)=1.0,
THICKNESS(1)=0.025/
&SURF ID='Aluminum looks solid',
RGB=200,202,231,
MATL_ID(1,1)= 'Aluminum',
MATL_MASS_FRACTION(1,1)=1.0,
THICKNESS(1)=7.0E-4/
&SURF ID='Glass',
RGB=153,204,255,
!TRANSPARENCY=0.298039,
MATL_ID(1,1)= 'Glass',
MATL_MASS_FRACTION(1,1)=1.0,
THICKNESS(1)=2.5E-3/

!Defining the Burner surface
!RAMP_Q is used to define how th HRRPUA or MLRPUA changes over time and links
!the surface to the &RAMP fxn that describes the temporal profile of the
! fire (scales the MLRPUA over time using the RAMP function)
!TMP_FRONT is the initial temperature of the front face of the
! surface releasing heat or mass
&SURF ID='Under car Burner',
RGB=102,0,0,
MLRPUA=0.046136,
RAMP_Q='Under car Burner_RAMP_Q',
TMP_FRONT=300.0/

!&RAMP defines how the growth and decay of the MLR over time,
!T = time (s) F = fraction of the maximum value at that time
!F contains the time dependent scaling factor, the area and the MLRPUA,
! that is why MLRPUA is 1
!F = f(t)*Area*MLRPUA, f(t) = (alpha*MLR_total(t))/(MLR_max)
&RAMP ID='Under car Burner_RAMP_Q', T=0.0, F=0.0/
&RAMP ID='Under car Burner_RAMP_Q', T=200.0, F=0.0/
&RAMP ID='Under car Burner_RAMP_Q', T=232.0, F=0.115868/
&RAMP ID='Under car Burner_RAMP_Q', T=272.0, F=0.202522/
&RAMP ID='Under car Burner_RAMP_Q', T=298.0, F=0.045581/
&RAMP ID='Under car Burner_RAMP_Q', T=416.0, F=0.089242/
&RAMP ID='Under car Burner_RAMP_Q', T=514.0, F=0.179671/
&RAMP ID='Under car Burner_RAMP_Q', T=553.0, F=0.178145/
&RAMP ID='Under car Burner_RAMP_Q', T=570.0, F=0.117648/
&RAMP ID='Under car Burner_RAMP_Q', T=648.0, F=0.18201/
&RAMP ID='Under car Burner_RAMP_Q', T=934.0, F=0.048279/
&RAMP ID='Under car Burner_RAMP_Q', T=1156.0, F=0.057805127/
&RAMP ID='Under car Burner_RAMP_Q', T=1304.0, F=0.046297/
&RAMP ID='Under car Burner_RAMP_Q', T=1588.0, F=0.0/
&RAMP ID='Under car Burner_RAMP_Q', T=1760.0, F=0.0/
&RAMP ID='Under car Burner_RAMP_Q', T=1838.0, F=0.0/
&RAMP ID='Under car Burner_RAMP_Q', T=2174.0, F=0.0/
&RAMP ID='Under car Burner_RAMP_Q', T=2400.0, F=0.0/
&RAMP ID='Under car Burner_RAMP_Q', T=2562.0, F=0.5/
&RAMP ID='Under car Burner_RAMP_Q', T=2748.0, F=0.355331703/
&RAMP ID='Under car Burner_RAMP_Q', T=2776.0, F=0.350405658/
&RAMP ID='Under car Burner_RAMP_Q', T=3214.0, F=0.194774/

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&RAMP ID='Under car Burner_RAMP_Q', T=3500.0, F=0.0/
&SURF ID='Hood Burner ',
      RGB=204,0,0,
      MLRPUA=0.109701,
      RAMP_Q='Hood Burner_RAMP_Q',
      TMP_FRONT=300.0/
&RAMP ID='Hood Burner_RAMP_Q', T=0.0, F=0.0/
&RAMP ID='Hood Burner_RAMP_Q', T=200.0, F=0.0/
&RAMP ID='Hood Burner_RAMP_Q', T=232.0, F=0.0/
&RAMP ID='Hood Burner_RAMP_Q', T=272.0, F=0.0/
&RAMP ID='Hood Burner_RAMP_Q', T=298.0, F=0.182324/
&RAMP ID='Hood Burner_RAMP_Q', T=416.0, F=0.80318/
&RAMP ID='Hood Burner_RAMP_Q', T=514.0, F=0.419233/
&RAMP ID='Hood Burner_RAMP_Q', T=553.0, F=0.498806/
&RAMP ID='Hood Burner_RAMP_Q', T=570.0, F=0.549025/
&RAMP ID='Hood Burner_RAMP_Q', T=648.0, F=0.36402/
&RAMP ID='Hood Burner_RAMP_Q', T=934.0, F=0.048279/
&RAMP ID='Hood Burner_RAMP_Q', T=1156.0, F=0.0/
&RAMP ID='Hood Burner_RAMP_Q', T=1304.0, F=0.0/
&RAMP ID='Hood Burner_RAMP_Q', T=1588.0, F=0.0/
&RAMP ID='Hood Burner_RAMP_Q', T=1760.0, F=0.0/
&RAMP ID='Hood Burner_RAMP_Q', T=1838.0, F=0.0/
&RAMP ID='Hood Burner_RAMP_Q', T=2174.0, F=0.0/
&RAMP ID='Hood Burner_RAMP_Q', T=2400.0, F=0.0/
&RAMP ID='Hood Burner_RAMP_Q', T=2562.0, F=0.0/
&RAMP ID='Hood Burner_RAMP_Q', T=2748.0, F=0.0/
&RAMP ID='Hood Burner_RAMP_Q', T=2776.0, F=0.0/
&RAMP ID='Hood Burner_RAMP_Q', T=3214.0, F=0.0/
&RAMP ID='Hood Burner_RAMP_Q', T=3500.0, F=0.0/
&SURF ID='Cabin Burner ',
      COLOR='RED',
      MLRPUA=0.064111088,
      RAMP_Q='Cabin Burner_RAMP_Q',
      TMP_FRONT=300.0/
&RAMP ID='Cabin Burner_RAMP_Q', T=0.0, F=0.0/
&RAMP ID='Cabin Burner_RAMP_Q', T=200.0, F=0.0/
&RAMP ID='Cabin Burner_RAMP_Q', T=232.0, F=0.0/
&RAMP ID='Cabin Burner_RAMP_Q', T=272.0, F=0.0/
&RAMP ID='Cabin Burner_RAMP_Q', T=298.0, F=0.0/
&RAMP ID='Cabin Burner_RAMP_Q', T=416.0, F=0.0/
&RAMP ID='Cabin Burner_RAMP_Q', T=514.0, F=0.0/
&RAMP ID='Cabin Burner_RAMP_Q', T=553.0, F=0.035629/
&RAMP ID='Cabin Burner_RAMP_Q', T=570.0, F=0.117648/
&RAMP ID='Cabin Burner_RAMP_Q', T=648.0, F=0.36402/
&RAMP ID='Cabin Burner_RAMP_Q', T=934.0, F=0.38623/
&RAMP ID='Cabin Burner_RAMP_Q', T=1156.0, F=0.308294/
&RAMP ID='Cabin Burner_RAMP_Q', T=1304.0, F=0.231484723/
&RAMP ID='Cabin Burner_RAMP_Q', T=1588.0, F=0.300443835/
&RAMP ID='Cabin Burner_RAMP_Q', T=1760.0, F=0.223405757/
&RAMP ID='Cabin Burner_RAMP_Q', T=1838.0, F=0.219023681/
&RAMP ID='Cabin Burner_RAMP_Q', T=2174.0, F=0.259726387/
&RAMP ID='Cabin Burner_RAMP_Q', T=2400.0, F=0.192660054/
&RAMP ID='Cabin Burner_RAMP_Q', T=2562.0, F=0.4/
&RAMP ID='Cabin Burner_RAMP_Q', T=2748.0, F=0.284265362/

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&RAMP ID='Cabin Burner_RAMP_Q', T=2776.0, F=0.280324526/
&RAMP ID='Cabin Burner_RAMP_Q', T=3214.0, F=0.0/
&RAMP ID='Cabin Burner_RAMP_Q', T=3500.0, F=0.0/
&SURF ID='Trunk Burner ',
    RGB=255,51,51,
    MLRPUA=0.12189,
    RAMP_Q='Trunk Burner_RAMP_Q',
    TMP_FRONT=300.0/
&RAMP ID='Trunk Burner_RAMP_Q', T=0.0, F=0.0/
&RAMP ID='Trunk Burner_RAMP_Q', T=200.0, F=0.0/
&RAMP ID='Trunk Burner_RAMP_Q', T=232.0, F=0.0/
&RAMP ID='Trunk Burner_RAMP_Q', T=272.0, F=0.0/
&RAMP ID='Trunk Burner_RAMP_Q', T=298.0, F=0.0/
&RAMP ID='Trunk Burner_RAMP_Q', T=416.0, F=0.0/
&RAMP ID='Trunk Burner_RAMP_Q', T=514.0, F=0.0/
&RAMP ID='Trunk Burner_RAMP_Q', T=553.0, F=0.0/
&RAMP ID='Trunk Burner_RAMP_Q', T=570.0, F=0.0/
&RAMP ID='Trunk Burner_RAMP_Q', T=648.0, F=0.0/
&RAMP ID='Trunk Burner_RAMP_Q', T=934.0, F=0.0/
&RAMP ID='Trunk Burner_RAMP_Q', T=1156.0, F=0.019268/
&RAMP ID='Trunk Burner_RAMP_Q', T=1304.0, F=0.030865/
&RAMP ID='Trunk Burner_RAMP_Q', T=1588.0, F=0.300443835/
&RAMP ID='Trunk Burner_RAMP_Q', T=1760.0, F=0.223405757/
&RAMP ID='Trunk Burner_RAMP_Q', T=1838.0, F=0.219023681/
&RAMP ID='Trunk Burner_RAMP_Q', T=2174.0, F=0.259726387/
&RAMP ID='Trunk Burner_RAMP_Q', T=2400.0, F=0.192660054/
&RAMP ID='Trunk Burner_RAMP_Q', T=2562.0, F=0.1/
&RAMP ID='Trunk Burner_RAMP_Q', T=2748.0, F=0.071066/
&RAMP ID='Trunk Burner_RAMP_Q', T=2776.0, F=0.070081/
&RAMP ID='Trunk Burner_RAMP_Q', T=3214.0, F=0.0/
&RAMP ID='Trunk Burner_RAMP_Q', T=3500.0, F=0.0/

```

!Wheel obstructions

```

&OBST ID='RfW Rim', XB=5.7,5.8,9.5,9.9,0.6,1.1,
    PERMIT_HOLE=.FALSE., SURF_ID='Wheel Steel '/
&OBST ID='RfW Top', XB=5.7,5.8,9.36,10.02,1.1,1.16,
    PERMIT_HOLE=.FALSE., SURF_ID='Tire '/
&OBST ID='RfW Bottom', XB=5.7,5.8,9.36,10.02,0.5,0.6,
    PERMIT_HOLE=.FALSE., SURF_ID='Tire '/
&OBST ID='RfW Side', XB=5.7,5.8,9.36,9.5,0.6,1.1,
    PERMIT_HOLE=.FALSE., SURF_ID='Tire '/
&OBST ID='RfW Side', XB=5.7,5.8,9.9,10.02,0.6,1.1,
    PERMIT_HOLE=.FALSE., SURF_ID='Tire '/

&OBST ID='LFW Rim', XB=4.0,4.1,9.5,9.9,0.6,1.1,
    PERMIT_HOLE=.FALSE., SURF_ID='Wheel Steel '/
&OBST ID='LFW Top', XB=4.0,4.1,9.36,10.02,1.1,1.16,
    PERMIT_HOLE=.FALSE., SURF_ID='Tire '/
&OBST ID='LFW Bottom', XB=4.0,4.1,9.36,10.02,0.5,0.6,
    PERMIT_HOLE=.FALSE., SURF_ID='Tire '/
&OBST ID='LFW Side', XB=4.0,4.1,9.36,9.5,0.6,1.1,
    PERMIT_HOLE=.FALSE., SURF_ID='Tire '/
&OBST ID='LFW Side', XB=4.0,4.1,9.9,10.02,0.6,1.1,
    PERMIT_HOLE=.FALSE., SURF_ID='Tire '/

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```

&OBST ID='RBackW Rim', XB=5.7,5.8,12.1,12.5,0.6,1.1,
    PERMIT_HOLE=.FALSE., SURF_ID='Wheel Steel' /
&OBST ID='RBackW Top', XB=5.7,5.8,11.98,12.64,1.1,1.16,
    PERMIT_HOLE=.FALSE., SURF_ID='Tire' /
&OBST ID='RBackW Bottom', XB=5.7,5.8,11.98,12.64,0.5,0.6,
    PERMIT_HOLE=.FALSE., SURF_ID='Tire' /
&OBST ID='RBackW Side', XB=5.7,5.8,11.98,12.1,0.5,1.1,
    PERMIT_HOLE=.FALSE., SURF_ID='Tire' /
&OBST ID='RBackW Side', XB=5.7,5.8,12.5,12.64,0.5,1.1,
    PERMIT_HOLE=.FALSE., SURF_ID='Tire' /

&OBST ID='LBackW Rim', XB=4.0,4.1,12.1,12.5,0.6,1.1,
    PERMIT_HOLE=.FALSE., SURF_ID='Wheel Steel' /
&OBST ID='LBackW Top', XB=4.0,4.1,11.98,12.64,1.1,1.16,
    PERMIT_HOLE=.FALSE., SURF_ID='Tire' /
&OBST ID='LBackW Bottom', XB=4.0,4.1,11.98,12.64,0.5,0.6,
    PERMIT_HOLE=.FALSE., SURF_ID='Tire' /
&OBST ID='LBackW Side', XB=4.0,4.1,11.98,12.1,0.5,1.1,
    PERMIT_HOLE=.FALSE., SURF_ID='Tire' /
&OBST ID='LBackW Side', XB=4.0,4.1,12.5,12.64,0.5,1.1,
    PERMIT_HOLE=.FALSE., SURF_ID='Tire' /

!Panel Obstructions
&OBST ID='Right Panel', XB=2.1518,2.1568,9.35,12.65,0.38,2.21,
    SURF_ID='Steel' /
&OBST ID='Left Panel', XB=7.6432,7.6482,9.35,12.65,0.38,2.21,
    SURF_ID='Steel' /

```

!Components of vehicle obstructions

```

&OBST ID='Ceiling', XB=4.0,5.8,11.0,12.2,2.0,2.1, SURF_ID='Steel' /
&OBST ID='Hood Burner', XB=4.0,5.8,8.8,9.8,1.4,1.5,
    SURF_ID6='Steel','Steel','Steel','Steel','Steel','Steel','Hood Burner' /
&OBST ID='Hood Front', XB=4.0,5.8,8.8,8.9,0.8,1.4, SURF_ID='Steel' /
&OBST ID='Hood Side R', XB=5.7,5.8,8.9,10.0,0.8,1.4, SURF_ID='Steel' /
&OBST ID='Hood Side L', XB=4.0,4.1,8.9,10.0,0.8,1.4, SURF_ID='Steel' /
&OBST ID='Hood Wheel Side R', XB=5.4,5.7,9.1,9.2,0.8,1.4, SURF_ID='Steel' /
&OBST ID='Hood Wheel Side L', XB=4.1,4.4,9.1,9.2,0.8,1.4, SURF_ID='Steel' /
&OBST ID='Hood Floor', XB=4.0,5.8,8.8,9.8,0.7,0.8, SURF_ID='Steel' /
&OBST ID='Hood 2', XB=4.0,5.8,9.8,10.0,1.4,1.5, SURF_ID='Steel' /
&OBST ID='Front Windshield', XB=4.1,5.7,10.0,10.2,1.5,1.6, SURF_ID='Glass' /
&OBST ID='Front Windshield', XB=4.1,5.7,10.1,10.3,1.6,1.7, SURF_ID='Glass' /
&OBST ID='Front Windshield', XB=4.1,5.7,10.3,10.5,1.7,1.8, SURF_ID='Glass' /
&OBST ID='Front Windshield', XB=4.1,5.7,10.5,10.7,1.8,1.9, SURF_ID='Glass' /
&OBST ID='Front Windshield', XB=4.1,5.7,10.7,11.0,1.9,2.0, SURF_ID='Glass' /
&OBST ID='Front Windshield Support R',XB=5.7,5.8,10.0,10.2,1.5,1.6,
    SURF_ID='Steel' /
&OBST ID='Front Windshield Support R',XB=5.7,5.8,10.1,10.3,1.6,1.7,
    SURF_ID='Steel' /
&OBST ID='Front Windshield Support R',XB=5.7,5.8,10.3,10.5,1.7,1.8,
    SURF_ID='Steel' /
&OBST ID='Front Windshield Support R',XB=5.7,5.8,10.5,10.7,1.8,1.9,
    SURF_ID='Steel' /
&OBST ID='Front Windshield Support R',XB=5.7,5.8,10.7,11.0,1.9,2.0,

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SURF_ID='Steel '/
 &OBST ID='Front Windshield Support L',XB=4.0,4.1,10.0,10.2,1.5,1.6,
 SURF_ID='Steel '/
 &OBST ID='Front Windshield Support L',XB=4.0,4.1,10.1,10.3,1.6,1.7,
 SURF_ID='Steel '/
 &OBST ID='Front Windshield Support L',XB=4.0,4.1,10.3,10.5,1.7,1.8,
 SURF_ID='Steel '/
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 SURF_ID='Steel '/
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 SURF_ID6='Steel ', 'Steel ', 'Steel ', 'Trunk Burner ', 'Steel ', 'Trunk Burner '/
 &OBST ID='Back Top ', XB=4.0,5.8,12.8,12.9,1.4,1.5,
 SURF_ID6='Steel ', 'Steel ', 'Steel ', 'Steel ', 'Steel ', 'Trunk Burner '/
 &OBST ID='Back Side R ', XB=5.7,5.8,12.8,12.9,0.8,1.4, SURF_ID='Steel '/
 &OBST ID='Back Side L ', XB=4.0,4.1,12.8,12.9,0.8,1.4, SURF_ID='Steel '/
 &OBST ID='Back Bottom ', XB=4.1,5.7,12.8,12.9,0.8,0.9, SURF_ID='Steel '/
 &OBST ID='Back Windshield ', XB=4.1,5.7,12.2,12.4,1.9,2.0, SURF_ID='Glass '/
 &OBST ID='Back Windshield ', XB=4.1,5.7,12.7,12.8,1.5,1.6, SURF_ID='Glass '/
 &OBST ID='Back Windshield ', XB=4.1,5.7,12.4,12.5,1.8,1.9, SURF_ID='Glass '/
 &OBST ID='Back Windshield ', XB=4.1,5.7,12.6,12.7,1.6,1.7, SURF_ID='Glass '/
 &OBST ID='Back Windshield ', XB=4.1,5.7,12.5,12.6,1.7,1.8, SURF_ID='Glass '/
 &OBST ID='Trunk Floor ', XB=4.4,5.4,12.2,12.8,0.8,0.9,
 SURF_IDS='Steel ', 'Steel ', 'Under car Burner '/
 &OBST ID='Car Floor ', XB=4.0,5.8,9.8,12.2,0.7,0.8,
 SURF_ID6='Steel ', 'Steel ', 'Steel ', 'Steel ',
 'Under car Burner ', 'Cabin Burner '/
 &OBST ID='Right Side Door ', XB=5.7,5.8,10.0,12.8,0.8,1.5, SURF_ID='Steel '/
 &OBST ID='Right Side Window ', XB=5.7,5.8,12.0,12.8,1.5,2.0, SURF_ID='Steel '/
 &OBST ID='R Middle Support ', XB=5.7,5.8,11.3,11.5,1.5,2.0, SURF_ID='Steel '/
 &OBST ID='L Middle Support ', XB=4.0,4.1,11.3,11.5,1.5,2.0, SURF_ID='Steel '/
 &OBST ID='Left Side Door ', XB=4.0,4.1,10.0,12.8,0.8,1.5, SURF_ID='Steel '/
 &OBST ID='Left Side Window ', XB=4.0,4.1,12.0,12.8,1.5,2.0, SURF_ID='Steel '/
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 &OBST ID='RF Wheel Covers ', XB=5.3,5.4,10.0,10.3,0.8,1.3, SURF_ID='Steel '/
 &OBST ID='RF Wheel Covers ', XB=5.4,5.7,10.2,10.3,0.8,1.3, SURF_ID='Steel '/
 &OBST ID='LF Wheel Covers ', XB=4.1,4.5,10.0,10.3,1.3,1.5, SURF_ID='Steel '/
 &OBST ID='LF Wheel Covers ', XB=4.4,4.5,10.0,10.3,0.8,1.3, SURF_ID='Steel '/
 &OBST ID='LF Wheel Covers ', XB=4.1,4.4,10.2,10.3,0.8,1.3, SURF_ID='Steel '/
 &OBST ID='LB Wheel Covers ', XB=4.1,4.5,11.7,12.8,1.4,1.5, SURF_ID='Steel '/
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 &OBST ID='LB Wheel Covers ', XB=4.1,4.4,11.7,11.8,0.8,1.4, SURF_ID='Steel '/
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 &OBST ID='RB Wheel Covers ', XB=5.3,5.4,11.7,12.2,0.8,1.4, SURF_ID='Steel '/
 &OBST ID='RB Wheel Covers ', XB=5.4,5.7,11.7,11.8,0.8,1.4, SURF_ID='Steel '/

 !Pan Obstructions
 &OBST ID='Pan Base ', XB=3.6,6.2,8.4,13.4,0.4,0.5, SURF_ID='Pan Steel '/
 &OBST ID='Back Lip ', XB=3.6,6.2,13.3,13.4,0.5,0.7, SURF_ID='Pan Steel '/
 &OBST ID='Front Lip ', XB=3.6,6.2,8.4,8.5,0.5,0.7, SURF_ID='Pan Steel '/
 &OBST ID='Left Lip ', XB=3.6,3.7,8.5,13.3,0.5,0.7, SURF_ID='Pan Steel '/
 &OBST ID='Right Lip ', XB=6.1,6.2,8.5,13.3,0.5,0.7, SURF_ID='Pan Steel '/

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!Holes that are part of the car geometry
&HOLE ID='RFH', XB=5.4,5.8,9.2,10.2,0.5,1.3, COLOR='INVISIBLE' /
&HOLE ID='LFH', XB=4.0,4.4,9.2,10.2,0.5,1.3, COLOR='INVISIBLE' /
&HOLE ID='RBackH', XB=5.4,5.8,11.8,12.8,0.5,1.3, COLOR='INVISIBLE' /
&HOLE ID='LBackH', XB=4.0,4.4,11.8,12.8,0.5,1.3, COLOR='INVISIBLE' /
&HOLE ID='Hole', XB=4.5,5.3,9.8,10.4,0.9,1.4, COLOR='INVISIBLE' /
&HOLE ID='Back Hole', XB=4.0,5.8,12.6,12.8,1.7,2.1, COLOR='INVISIBLE' /
&HOLE ID='Back Hole', XB=4.0,5.8,12.5,12.6,1.8,2.1, COLOR='INVISIBLE' /
&HOLE ID='Back Hole', XB=4.0,5.8,12.7,12.8,1.6,1.7, COLOR='INVISIBLE' /
&HOLE ID='Back Hole', XB=4.0,5.8,12.4,12.5,1.9,2.1, COLOR='INVISIBLE' /

```

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!Describing the Open Vents and their locations
&VENT ID='Mesh Vent: Mesh1 [XMIN]', SURF_ID='OPEN',
XB=-0.4,-0.4,7.0,11.0,0.0,4.0/
&VENT ID='Mesh Vent: Mesh1 [ZMIN]', SURF_ID='CONCRETE',
XB=-0.4,2.0,7.0,11.0,0.0,0.0/
&VENT ID='Mesh Vent: Mesh2 [ZMIN]', SURF_ID='CONCRETE',
XB=2.0,4.8,7.0,11.0,0.0,0.0/
&VENT ID='Mesh Vent: Mesh3 [XMIN]', SURF_ID='OPEN',
XB=-0.4,-0.4,7.0,11.0,4.0,8.0/
&VENT ID='Mesh Vent: Mesh3 [ZMAX]', SURF_ID='OPEN',
XB=-0.4,2.0,7.0,11.0,8.0,8.0/
&VENT ID='Mesh Vent: Mesh4 [ZMAX]', SURF_ID='OPEN',
XB=2.0,4.8,7.0,11.0,8.0,8.0/
&VENT ID='Mesh Vent: Mesh5 [XMIN]', SURF_ID='OPEN',
XB=-0.4,-0.4,11.0,15.0,0.0,4.0/
&VENT ID='Mesh Vent: Mesh5 [ZMIN]', SURF_ID='CONCRETE',
XB=-0.4,2.0,11.0,15.0,0.0,0.0/
&VENT ID='Mesh Vent: Mesh6 [ZMIN]', SURF_ID='CONCRETE',
XB=2.0,4.8,11.0,15.0,0.0,0.0/
&VENT ID='Mesh Vent: Mesh7 [XMIN]', SURF_ID='OPEN',
XB=-0.4,-0.4,11.0,15.0,4.0,8.0/
&VENT ID='Mesh Vent: Mesh7 [ZMAX]', SURF_ID='OPEN',
XB=-0.4,2.0,11.0,15.0,8.0,8.0/
&VENT ID='Mesh Vent: Mesh8 [ZMAX]', SURF_ID='OPEN',
XB=2.0,4.8,11.0,15.0,8.0,8.0/
&VENT ID='Mesh Vent: Mesh9 [ZMIN]', SURF_ID='CONCRETE',
XB=4.8,7.4,7.0,11.0,0.0,0.0/
&VENT ID='Mesh Vent: Mesh10 [XMAX]', SURF_ID='OPEN',
XB=10.0,10.0,7.0,11.0,0.0,4.0/
&VENT ID='Mesh Vent: Mesh10 [ZMIN]', SURF_ID='CONCRETE',
XB=7.4,10.0,7.0,11.0,0.0,0.0/
&VENT ID='Mesh Vent: Mesh11 [ZMAX]', SURF_ID='OPEN',
XB=4.8,7.4,7.0,11.0,8.0,8.0/
&VENT ID='Mesh Vent: Mesh12 [XMAX]', SURF_ID='OPEN',
XB=10.0,10.0,7.0,11.0,4.0,8.0/
&VENT ID='Mesh Vent: Mesh12 [ZMAX]', SURF_ID='OPEN',
XB=7.4,10.0,7.0,11.0,8.0,8.0/
&VENT ID='Mesh Vent: Mesh13 [ZMIN]', SURF_ID='CONCRETE',
XB=4.8,7.4,11.0,15.0,0.0,0.0/
&VENT ID='Mesh Vent: Mesh14 [XMAX]', SURF_ID='OPEN',
XB=10.0,10.0,11.0,15.0,0.0,4.0/
&VENT ID='Mesh Vent: Mesh14 [ZMIN]', SURF_ID='CONCRETE',
XB=7.4,10.0,11.0,15.0,0.0,0.0/

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&VENT ID='Mesh Vent: Mesh15 [ZMAX]', SURF_ID='OPEN',
XB=4.8,7.4,11.0,15.0,8.0,8.0/
&VENT ID='Mesh Vent: Mesh16 [XMAX]', SURF_ID='OPEN',
XB=10.0,10.0,11.0,15.0,4.0,8.0/
&VENT ID='Mesh Vent: Mesh16 [ZMAX]', SURF_ID='OPEN',
XB=7.4,10.0,11.0,15.0,8.0,8.0/
&VENT ID='Mesh Vent: Mesh17 [XMAX]', SURF_ID='OPEN',
XB=10.0,10.0,15.0,19.0,0.0,8.0/
&VENT ID='Mesh Vent: Mesh17 [XMIN]', SURF_ID='OPEN',
XB=-0.4,-0.4,15.0,19.0,0.0,8.0/
&VENT ID='Mesh Vent: Mesh17 [YMAX]', SURF_ID='OPEN',
XB=-0.4,10.0,19.0,19.0,0.0,8.0/
&VENT ID='Mesh Vent: Mesh17 [ZMAX]', SURF_ID='OPEN',
XB=-0.4,10.0,15.0,19.0,8.0,8.0/
&VENT ID='Mesh Vent: Mesh17 [ZMIN]', SURF_ID='CONCRETE',
XB=-0.4,10.0,15.0,19.0,0.0,0.0/
&VENT ID='Mesh Vent: Mesh18 [XMAX]', SURF_ID='OPEN',
XB=10.0,10.0,3.0,7.0,0.0,8.0/
&VENT ID='Mesh Vent: Mesh18 [XMIN]', SURF_ID='OPEN',
XB=-0.4,-0.4,3.0,7.0,0.0,8.0/
&VENT ID='Mesh Vent: Mesh18 [YMIN]', SURF_ID='OPEN',
XB=-0.4,10.0,3.0,3.0,0.0,8.0/
&VENT ID='Mesh Vent: Mesh18 [ZMAX]', SURF_ID='OPEN',
XB=-0.4,10.0,3.0,7.0,8.0,8.0/
&VENT ID='Mesh Vent: Mesh18 [ZMIN]', SURF_ID='CONCRETE',
XB=-0.4,10.0,3.0,7.0,0.0,0.0/

```

```

!&BNDF is defined as the boundry file output, visualizes surface quantities
&BNDF QUANTITY = 'GAUGE HEAT FLUX'/
&BNDF QUANTITY = 'RADIATIVE HEAT FLUX'/
&BNDF QUANTITY = 'NET HEAT FLUX'/
&BNDF QUANTITY='WALL TEMPERATURE'/

```

```

!&SLCF defines a slice file (2D data visualization in Smokeview)
!Quantity is the physical quantity that will be shown on the slice
!CELL_CENTERED true records the data at the center of each
! grid cell as opposed to at the edges
!ID is the label of the slice file (primarily for organization)
!PB is the plane location of the slice (i.e. X, Y, or Z)
!The slice file 'Intigrated Intensity' refers to the total radiative
! intensity integrated over all direction at a given point on a surface
!^ measures how much radiative energy is emitted or recieved per unit area
&SLCF QUANTITY='TEMPERATURE', CELL_CENTERED=.TRUE.,
ID='Temperature', PBX=4.9/

```

```

&SLCF QUANTITY='VELOCITY', VECTOR=.TRUE., CELL_CENTERED=.TRUE.,
ID='Velocity', PBX=4.9/
&SLCF QUANTITY='HRRPUV', CELL_CENTERED=.TRUE.,
ID='HRRPUV', PBX=4.9/
&SLCF QUANTITY='INTEGRATED INTENSITY', CELL_CENTERED=.TRUE.,
ID='Source Visibility Non-Driver', PBX=4.0/
&SLCF QUANTITY='INTEGRATED INTENSITY', CELL_CENTERED=.TRUE.,
ID='Source Visibility Driver', PBX=5.8/

```

&TAIL /

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