

Assessment of the Fire Dynamics Simulator for Modeling Fire Suppression in Engine Rooms of Ships with Low-Pressure Water Mist

Abstract. Water mist-based fire-extinguishing systems are gaining acceptance for the protection of ship machinery spaces. The use of simulation tools presents a great potential for taking a performance-based design (PBD) approach to these fire scenarios. The Fire Dynamics Simulator (FDS) is the most frequently used and validated fire modeling software; however, studies of low-pressure water mist fire suppression modeling in ship engine rooms are rare. This paper contributes to the current literature by using the FDS to model a series of fire suppression scenarios defined by the International Maritime Organization (IMO) Circulars, including spray and pool fires with heptane and diesel oil, as well as exposed and obstructed fires. The simulation results are compared to data from full-scale tests conducted at recognized fire testing laboratories. Furthermore, an analysis of both the experimental and model uncertainties is carried out to assess the simulations performance. In general, a good agreement in compartment temperature evolution and fire extinguishing time is found for the modeled fire scenarios. The results support the application of FDS in a PBD approach for the design of water mist fire extinguishing systems for machinery spaces in ships. In this way, designers and engineers could model different machinery volumes and nozzles spacings that differ from those prescribed for a one story square engine room of the IMO, and, thus, predict the evolution of temperatures and extinguishing times for get the authorities approval.

Keywords: water mist, modeling, performance-based design, fire suppression, compartment fires, FDS

1. Introduction

The engine room is one of the most critical spaces in a ship in terms of fire protection, as it includes the main propulsion, auxiliary generators, fuel system and different electrical equipment in a very confined and complex space. Approximately 50% of all fires on merchant ships originate in the engine room, with 60% of those fires caused by fuel leaks from pressurized systems (1). Similarly, statistical data from the ship classification society Det Norske Veritas in 2007 show that 63% of fires on ships start in engine rooms, and 56% of all engine room fires, excluding those in yard repairs, were caused by oil leakages onto hot surfaces.

Fire suppression systems for ship engine rooms have typically been based on gaseous total flooding technology, either with carbon dioxide (CO₂) or with halons or halocarbons as fire extinguishing agents. However, water mist has gained growing acceptance as an alternative to those systems because it is not potentially harmful to humans (unlike CO₂), it is environmentally friendly (unlike most halons and halocarbons), and it generally presents a lower lifecycle cost (2). Additionally, water mist systems may offer greater fire protection in spaces with large openings because gaseous total flooding agents require the maintenance of gas concentration in the compartment to be effective.

When compared to classic water suppression systems, with significantly higher flows and larger droplet sizes, water mist uses less water, thus requiring smaller bore pipework, producing little water damage, and offering enhanced fuel cooling, oxygen depletion and dilution (3,4). Compared with high-pressure system (pressure ≥ 34.5 bar) (5), low-pressure systems (pressure ≤ 12.1 bar) (5), exhibit worse gas mixing and evaporation rates and higher probabilities of electric equipment damage due to larger drop sizes and water

flows. Additionally, low-pressure systems have higher water consumption, pipe bores and overall system weight. Despite the above, the modern low-pressure systems reach an equilibrium between drop atomization and momentum, which allows sufficient oxygen displacement and fuel vapor dilution combined with better cooling of fuel surface and the capability of reaching more distant fires. The use of conventional pipework, couplings, ancillaries and pumps also reduce total lifecycle costs. As a result, low-pressure water mist technology is currently widely used for local application and total flooding engine room fire suppression systems.

The water mist-based fire-extinguishing systems for machinery spaces in ships must be approved by Notified Bodies according to the detailed requirements for design, installation and fire test procedures established in the Circulars 1165 (6), 1387 (7) and the subsequent amendments of the International Maritime Organization Maritime Safety Committee (IMO-MSC). However, the performance of water mist systems is so related to the arrangement and ventilation of the compartment, the fire location and the exact position of the nozzles that the results obtained in the IMO tests may significantly differ from those corresponding to a “real-world” situation. Therefore, when designing water mist systems, the performance based design (PBD) approach can be used to complement or even to replace the design made according to the prescriptive regulations (2). PBD methods necessarily require reasonably accurate simulation tools that comprehensively consider the effect of the real conditions on fire development and extinguishment, which is particularly challenging for the case of water-mist-based fire suppression systems installed in ship engine rooms.

Computational fluid dynamics (CFD) modeling has been increasingly used by designers and researchers as an effective tool to predict fire behavior and extinguishment under different conditions. A number of CFD models are commercially available, including the Fire Dynamics Simulator (FDS), which was developed by the National Institute for Standards and Technology (NIST), is one of the most widely used models, and is validated for diverse fire scenarios including water mist applications (8). The VTT Technical Research Centre of Finland has developed some research projects (9–11) to improve and enhance the capabilities of the FDS software to describe water spray dynamics, gas phase cooling by water sprays, flame extinguishment, and suppression of large complex fire loads. These studies have yielded several new features that have been added to the FDS code related to the description of water sprays and flame extinguishment. These new features have been validated with experimental data found in the literature and by experiments conducted at the VTT.

The inherent difficulty of modeling fire dynamics in complex fire scenarios is emphasized by various studies in the literature, such as (12–14), where the results from a realistic fire scenario involving multiple fuel packages and nontrivial fire growth were compared against the numerical results from different CFD models with rather poor accuracy. Furthermore, the full-scale tests and benchmark exercises performed within the frame of the PRISME project highlighted the limitations of the current CFD models in predicting smoke and heat propagation and the performance of various extinguishing systems in certain complex scenarios (15). In turn, the field results from a series of fire tests conducted in a large-scale facility using pool fires of 1.3 to 5.3 MW were in reasonably good agreement with those predicted by FDS simulations in terms of smoke layer turbulence behavior, especially in the far field of the plume (16–18).

Computational simulation of water mist fire extinguishing is a challenging task; because it includes modeling water droplet evaporation and flame interaction, heat absorption, reduction of temperature and combustion rate. A number of studies have addressed the CFD modeling of water mist fire suppression systems in recent years. The extinguishment of liquid pool fires in both open and closed compartments has been assessed through experimental tests and CFD modeling in (19–24), with generally good agreement between numerical and measured results. A real-scale study of pool fire suppression by water mist applied in both the early and late stages of fire development was performed in (25–27), demonstrating that the suppression model did not exactly follow the experimental observations.

However, modeling fire evolution, smoke flowing and water mist suppression in ship machinery rooms is challenging due to the diversity of fire loads in a confined space with complex distribution and variable ventilation conditions (28). Fire suppression and extinguishing mechanisms are also strongly dependent on scenario-specific conditions [9].

Despite significant efforts being devoted to studying and computational modeling of fire and water mist suppression, little research has been performed on the computational modeling of fire suppression in ship engine rooms. The FDS Validation Guide (8) includes a set of water mist suppression models based on the US Coast Guard (USCG) full scale tests for small machinery spaces described in the report CG-21-99 (29). However, only part of the test performed were modeled (heptane spray fires with three ventilation conditions). The validations used were simple, only considering fire control time, which is determined when the heat release rate (HRR) drops to half of its specified value. Prasad et al. (30) developed algorithms to predict the suppression of opposed flow diffusion flames and produced valuable insight into the dynamical interaction between water mist and fires. Later, studies conducted at the Naval Postgraduate School (31,32) showed that CFD software can successfully model smoke propagation in shipboard spaces with geometric interference. Zou et al. (33) also demonstrated that the smoke moving process in a large cabin space of a ship can be reasonably simulated by FDS. Wang and Yin (34) developed a numerical model to simulate the fire spreading within an enclosed cabin and proposed a fire safety design for a passenger ship. Su and Wang (28) developed an FDS model with a multilayer structure and multiple subregions to represent a real ship engine room structure, which allowed the assessment of the temperature gradient, the variation of the temperature versus oxygen concentration and the pressure gradient between engine subregions. Last, a framework for using computational fire simulations during the early phases of ship design was explored by Kang et al. (35).

The following issues have been identified as key knowledge gaps in current low-pressure water mist modeling for ship machinery spaces: i) limitations in the actual CFD tools, which include inaccurate algorithms to model fire suppression of different types of fuels by cooling, wetting and oxygen displacement, inaccuracies in the spray model, and difficulties in predicting drop transport and flow around obstructions (2); ii) scarce information about real-scale tests, which may hinder the improvement of aforementioned algorithms; and iii) few detailed validations that encourage the use of CFD tools to accurately model these kinds of scenarios

In light of the above, the principal objective of this paper is to assess the capabilities of the FDS computational simulation tool for predicting low-pressure water mist suppression under different scenarios that can occur in ship engine rooms. This study includes spray and pool fires with heptane and diesel oil as fuels, which may be

exposed or obstructed from water mist. Additionally, local application and total flooding water mist fire suppression are considered. The results from FDS simulations undertaken with version 6.7.0 are compared against the data from tests conducted in full-scale engine rooms at the U.S. Coast Guard (USCG) R&D Center and the Danish Fire Laboratories (DFL) in accordance with IMO MSC Circulars. Variables, such as the evolution of temperatures, HRR and fire extinguishment time, are used to evaluate the simulation performance. An analysis of both experimental and model uncertainties is also conducted. It is hoped that the results of this investigation may help and encourage designers to apply the PBD approach in the design of effective low-pressure water mist suppression systems for ship machinery spaces.

2. Modeling methodology and experimental tests

2.1 *Fire dynamics and suppression modeling*

In this section, the essential characteristics of the FDS model of fire for this work are presented. The governing equations and solutions for the gas and liquid phase as well as the energy conservation and the radiation heat transfer are described in the FDS Technical Reference Guide (36) and, therefore, will not be repeated herein.

2.1.1 *Fire modeling*

FDS is a specific CFD model for low Mach number flows driven by combustion heat release and buoyancy. FDS numerically solves the Navier–Stokes equations by using a second-order finite volumes numerical scheme. The fire modeling is based on conservation equations of multiple variables involved in the combustion phenomena. Governing equations are applied to the model geometry through uniformly spaced three-dimensional grids.

Flow velocity and pressure are calculated by solving the continuity and momentum transport equations. The temperature field is solved through the energy conservation and the concentrations of each chemical species are calculated through an additional conservation equation for each specie. Several turbulence models are used to simplify the complex turbulent flow phenomena. The LES framework for subgrid-scale turbulence modeling coupled with a modified version of the Deardorff eddy-viscosity model is used in this work.

The redistribution of energy via thermal radiation is very important and needs to be included in the energy transport equation. For the radiative heat transfer model, the radiative transport equation (RTE) for non-scattering gray gas is used, and in some limited cases a wide band model. The RTE is solved using a finite volume method, in which radiation intensity is computed over a discrete number of solid angles and through a source term introduced in the energy equation that accounts for the energy absorbed and emitted by the cells. The absorption coefficients of the gas-soot mixtures are computed using the RadCal narrow-band model. Liquid droplets are also considered to affect the absorption and scattering of thermal radiation. This effect is important in cases involving mist sprinklers, and plays a role in all sprinkler cases. The absorption and scattering coefficients are based on Mie theory (37).

Combustion is a very complex combination of transport and multiple reaction phenomena. To be feasible in computational terms, the FDS simplifies the process to a few reactions and chemical species. The combustion model determines the mean chemical mass production rate of species per unit volume, in the

species transport equation. The large number of chemical species is modeled through a few representative species known as lumped species, which are usually a generic hydrocarbon, $C_nH_mO_a$, representing the fuel, air (N_2 , O_2 , H_2O , and CO_2) and products (N_2 , CO , CO_2 , H_2O , and soot). The heat released by the reaction of lumped species is added to the energy transport equation as the source term. The reactions kinetics are controlled by temperature and mixing, which is modelled with the Eddy Dissipation Concept model.

2.1.2 *Water Mist fire suppression modeling*

In CFD, Lagrangian particles are used to represent a wide variety of objects that cannot be resolved on the numerical grid as liquid droplets (water, or fuel sprays). CFD numerical modeling has a limited ability to predict water mist nozzle atomization, i.e., the process of breaking up a volume of liquid into small drops (38). To bypass this complexity, water Lagrangian particles are introduced into the computational domain at a prescribed distance, r_0 , from the nozzle using an Eulerian-Lagrangian approach. Each computational Lagrangian particle represents a large group of real droplets. Thus, the spray consists of a spherical surface defined by r_0 and two angles: the elevation angle and the azimuthal angle. Because it is impractical to simulate the motion of every particle in the sprinkler spray, a computed particle injection rate, N_p , must be prescribed. This value of droplets per second is independent of water flow and other physical phenomena, even if the value change from 5×10^3 (default) to 5×10^6 , there is no effect in the water flow.

When the droplets encounter burning surfaces, two effects occur: cooling of the surface and the surrounding gas and reduction of the fuel pyrolysis rate. The reduction of the pyrolysis rate due to water can be modeled through an exponential behavior of fire suppression by water, that considers both the cooling of non-burning surfaces and the decrease in the HRR of burning surfaces (39).

FDS uses simple empirical rules to predict local extinction within a grid cell (36). There are three models ('EXTINCTION' 1, 2 and 3) based on the concept of critical flame temperature (CFT) related with the oxygen index, introduced by Beyler (40). The EXTINCTION 2 model is the default for LES turbulence model and was used in this study. For a comprehensive explanation of the mathematical and thermodynamic models employed in FDS, refer to the FDS Technical Reference Guide (36).

The details of the compartment, as relative positions of the fire and vents/openings are critical to local oxygen and fuel concentrations and to determining whether a fire will be suppressed.

2.2 *Fire extinguishing tests*

The most usual fire scenarios in a ship engine room are considered according to the IMO MSC/Circular 1165 (6) and subsequent amendments. Eight tests performed at DFL and USCG R&D Center were chosen to be simulated in this work as the most representative fire situations in ship machinery spaces. Table 1 shows the tests simulated in this work and the main test characteristics, such as fire type, obstruction locations and performance required. After ignition of fuel sources, a preburn time is required before the extinguishing agent is discharged. The first five tests refer to spray fires. No. 1 and No. 2 are heptane and diesel-exposed flames on top of the mock-up, while No. 3 and No. 4 are obstructed-concealed diesel spray fires on the side of the mock-up, and No. 5 involves local application instead of total flooding. The three last fires are obstructed diesel and heptane pool fires. No. 6 is an obstructed heptane tray fire, and No. 7 is an obstructed diesel tray fire. Lastly, No. 8 is a thermal management test, where the size of the fire must be scaled in accordance with

the test hall size and the acceptance criterion is managing the temperature below 100°C. All the tests are total flooding water mist (nozzles are distributed over the entire roof) except No. 5, which is local application with nozzles located only over the mock-up. As fires are quite different, the relative importance of the various extinguishment mechanisms varies.

Table 1
Fire extinguishing tests and performance required

Test No.	Fire Type	Fuel	Testing facility	Fire Scenario	Acceptance extinguish time.
1	Spray on top exposed	Heptane	DFL	IMO 8, low-pressure 6.6 MW horizontal spray on top of mock-up at 0.5 m from the front edge of a steel plate (30 x 60 x 5 cm) offset 20° to the spray axis and heated to 350°C.	< 15 min
2	Spray on top exposed	Diesel	DFL	IMO 1, low-pressure 6 MW horizontal spray on top of simulated engine (mock-up).	< 15 min
3	Spray side concealed	Diesel	DFL	IMO 4, low-pressure 6 MW horizontal spray fire on the side of mock-up under exhaust obstruction plate. Tray 0.1 m ² positioned on top of the bilge plate.	< 15 min
4	Spray side concealed	Diesel	USCG	IMO 4, low-pressure 1.3 MW horizontal spray fire on the side of mock-up under exhaust obstruction plate. Tray 0.1 m ² positioned on top of the bilge plate.	< 15 min
5	Spray side concealed	Diesel	USCG	Local application water mist, low-pressure 1.3 MW horizontal spray fire on the side on of mock-up.	< 15 min
6	Pool fire side concealed	Heptane	DFL	IMO 5, fire tray 0.7 x 3.0 m 7 MW on top of bilge plate under exhaust obstruction plate.	< 15 min
7	Pool fire side concealed	Diesel	USCG	IMO 5, fire tray 0.74 x 0.74 m, 1 MW on top of bilge plate under exhaust obstruction plate.	< 15 min
8	Pool fire under bilge	Heptane	DFL	Thermal management, fire tray 1 MW fire size correlated with compartment volume. A square horizontal obstruction steel plate shielded the fire tray from water spray.	<100°C no later than 300s

2.3 Modeling of fire scenarios

In this section, the modeling of fire scenarios, fire conditions and water mist characterization are shown. FDS requires a large amount of information to run a simulation and achieve reliable results, which can be grouped as follows:

- Compartment geometry and obstructions: room dimensions and layout, vent conditions, materials
- Fire type: gas or liquid spray nozzle characteristics, pool dimensions, solid dimensions
- Fuel characteristics: composition, heat of combustion, CO, CO₂, soot yields, specific heat, conductivity, emissivity, pyrolysis
- Initial boundary conditions: temperature, pressure, humidity
- Water mist: drop size, flow, speed, orientation and layout, time delay
- Measurement devices: gas, species, temperature, layout

2.3.1 Fire test compartment geometry and boundaries

The tests simulated in this work (Table 1) were performed in a simulated machinery space constructed to meet the dimensional requirements of the IMO test protocols: area greater than 100 m², height of at least 5 m and ventilation through a door opening of 2×2 m. As shown in Figure 1, a simulated engine mock-up with dimensions 1×3×3 m (w×l×h) is equipped with a 5-mm nominal thickness steel sheet and fitted with two steel

tubes of 0.3 m diameter and 3 m length that simulate exhaust manifolds and a grating. A floor plate system of 4×6×0.5 m surrounds the mock-up. At the top of the mock-up, a 0.7x3.5 m steel plate is arranged, which obstructs spray and pool fires No. 3, 4, 5, 6 and 7.

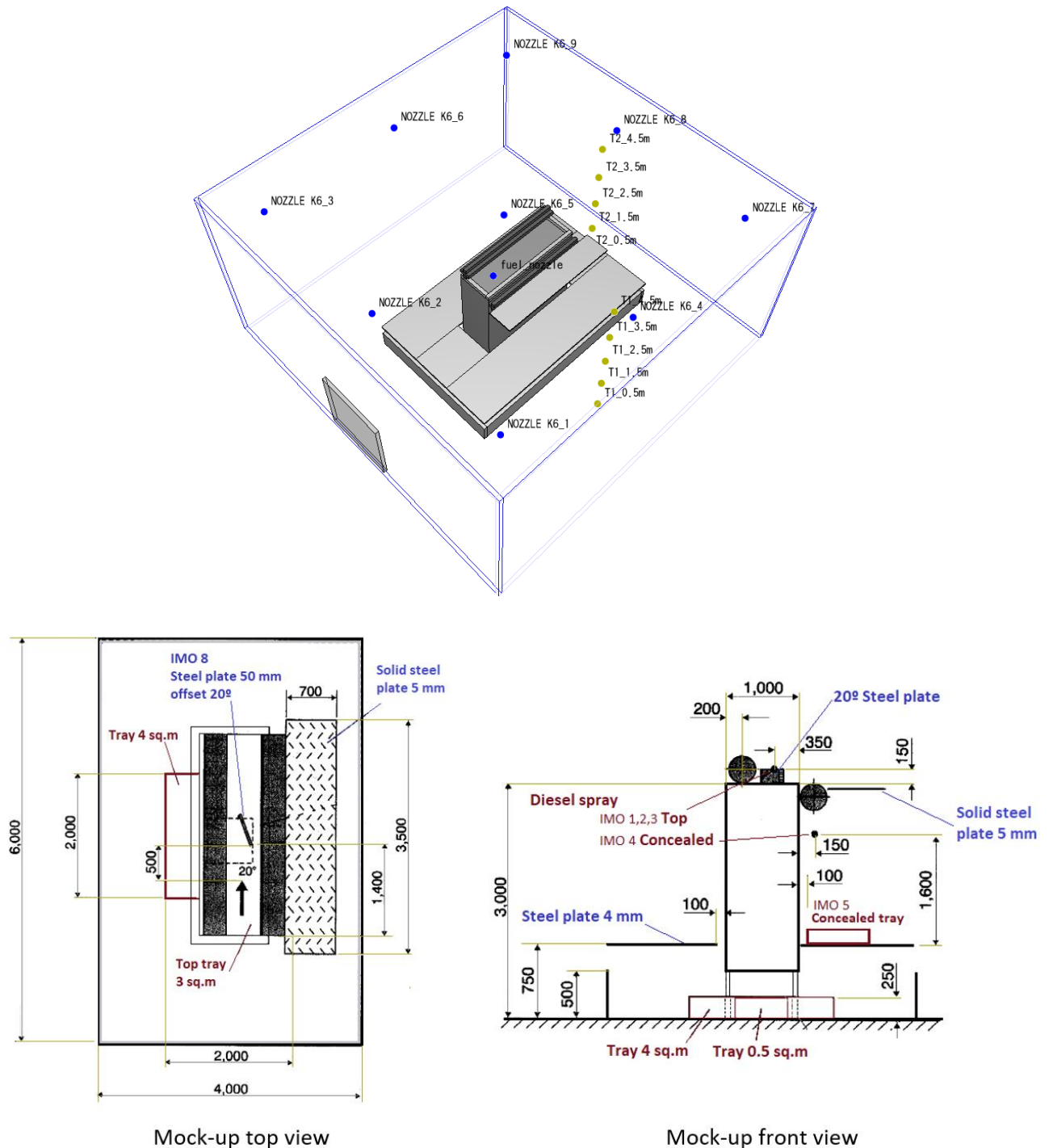


Figure 1. IMO test hall with layout of nozzles and thermocouples (up) and engine mock-up (down)

The simulated tests were performed at USCG R&D Center and DFL facilities, as shown in Table 1. Tests conducted at USCG were performed for evaluating local application and total flooding systems in a simulated machinery space measuring, $w \times l \times h = 10 \text{ m} \times 10 \text{ m} \times 5 \text{ m}$ (500 m^3 volume). Four generic water mist systems produced using off-the-shelf industrial spray nozzles were tested. The grid had 36 nozzles spaced $1.5 \times 1.5 \text{ m}$, as shown in Figure 2 (left). The compartment contained three large vent openings: two 2 m^2 vent openings

located on front bulkhead and a 6 m² vertical stack located in the overhead of the compartment. Tests were performed with one or two of the 2 m² vents opened. The vertical one was closed during the tests and was opened only during local application experiments and on all other for smoke evacuation once the test was finished. Tests conducted at DFL were performed in the same standard IMO machinery space volume of 500 m³. The aim was to test the extinguishing performance of the VID Fire-kill low-pressure K6 nozzle according to MSC/Circ. 1165 and subsequent amendments fire test guidelines. The performance of the K6 nozzle allows a grid with only 9 nozzles spaced 4 x 4 m, as shown in Figure 2 (right).

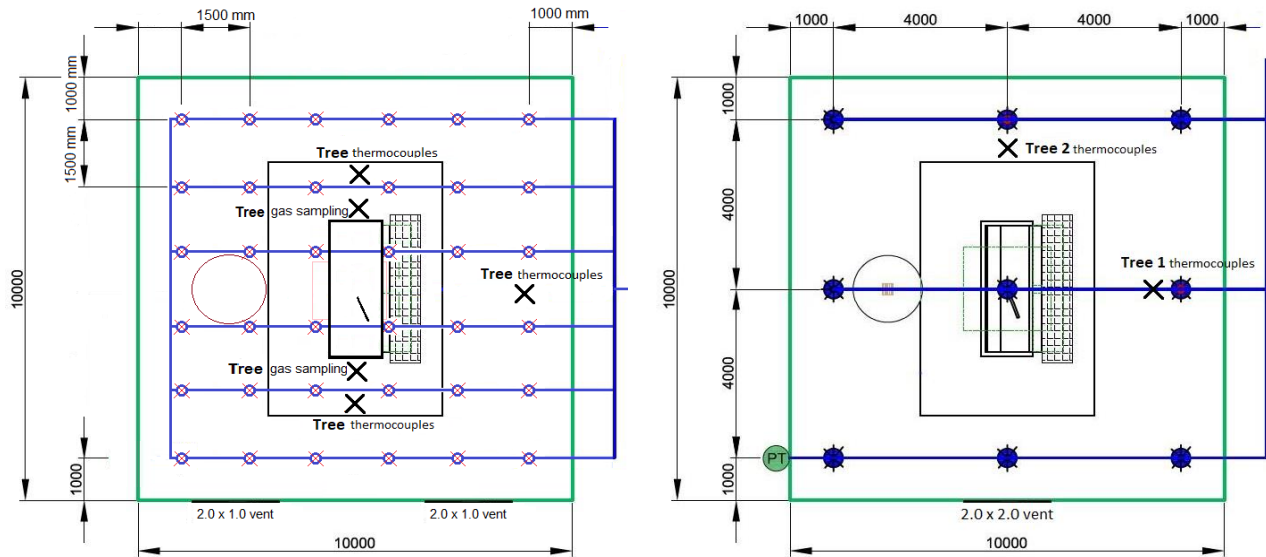


Figure 2. Layout of nozzles at the USCG facility (left) and the DFL facility (right)

Thermocouples are grouped in two trees. One tree is located on the right side, 4 m from the center of the mock-up and opposite to the tray for the thermal management test. The other tree is located 4 m from the center of the mock-up, on the opposite side of the door opening as shown in Figure 1 and Figure 2 (right). Each tree consists of five thermocouples (THCP) covered with an upper screen to avoid direct water spray impingement. In the USCG test a third THCP three was located on the front side, in the middle of the vents, in addition, two trees of gas (CO, CO₂, O₂) sensors at 1.0, 2.5 and 4.5 m heights were located in front and behind the mock-up as shown in Figure 2 (left).

The thermocouples used in DFL are type K, with a diameter of 0.5 mm and a measuring range of 0-400°C $\pm 1.5^\circ\text{C}$, connected to a multiplexer and a data-logger. The water and fuel flows are measured with flow-meters and the pressure with transducers. All the measuring devices are calibrated to an acceptance criterion of 5%. The thermocouple trees installed in the test compartment of USCG consisted of eight thermocouples type K with a diameter of 3.25 mm. The available reports claim an accuracy of 0.1 % in flow meters, 0.01 % in pressure transducers and a fuel flow rate accuracy of $0.03 \pm 0.005\%$ kg/s, which is equivalent to a HRR of 1.1 ± 0.1 MW.

The solid objects in FDS are defined by group obstructions and aligned with the cells on the Cartesian grid. Walls, floor, roof, mock-up, plates, trays and pipes are modeled as obstructions and include the object dimensions shown in Figure 1. Openings, such as doors or hatches, are specified in FDS as open surfaces type vent, and the simulations meet the standard requirement of 4 m² ventilation opening to the test hall (Test No.5

local application has also a 6 m² ceiling exhaust vent). The ambient temperature must be 20°C at the start of each test; thus, the initial temperature was defined as an initial condition.

The material properties are shown in Table 2. Most of these properties are from FDS validation work (8) and manufacturer data sheets; however specific heat and conductivity values of steel, insulation and fuels depending on temperature were introduced as a table. To reduce computational cost, the floor was set as porous, so droplets disappear upon reaching the floor.

Table 2
Materials of the test hall and mock-up

OBST	SURF	MTL	DENSITY (kg/m ³)	SPECIFIC HEAT (kJ/kg·K)	CONDUCTIVITY (W/m·K)
Mock-up	Basic	Steel 5 mm	7,850	0.45 (20°C) to 0.85 (677°C)	48 (<20°C) to 30 (>677°C)
Walls and Roof	Layered	Steel 5 mm	7,850	0.45 (20°C) to 0.85 (677°C)	48 (<20°C) to 30 (>677°C)
		Insulation 30mm	208	0.8 (20°C) to 2.0 (677°C)	0.05 (<20°C) to 0.2 (677°C)
Floor	Basic	Concrete	2,200	1.04	1.8

2.3.2 Spray and pool fire modeling

Two types of fire are simulated in this paper: spray and pool fire. The spray fire represents an engine injection hose fuel leak and is modeled by a constant flow of fuel particles entering the computational domain. The pool fire represents a pool of fuel in the floor that burns, and it is modeled by a surface of fuel that burns by liquid pyrolysis inside the fire scenario. Seven tests were modeled by predicting the HRR and only Test No. 7 was modeled with prescribed HRR curve because predicted results with diesel-dodecane pool fire were not satisfactory.

The diesel oil and heptane sprays are modeled following the values measured in the tests and required by the standard IMO: heptane or diesel range of 9 to 10.2 kg/min ± 0.6 kg/s. The nozzle used for the spray fire was a Lechler 460.728.30. CC with a spray pattern angle of 120°. Liquid spray fire in FDS is specified as Lagrangian particles of fuel produced by a nozzle, the properties of the particles, the nozzle and also the reaction (heptane or diesel oil combustion) of the fuel must be defined. The nominal HRR was estimated in the experiments assuming that all of the fuel is consumed with a 100 percent combustion efficiency, so for simulations, the same assumption was considered.

The pool fires are modeled with information of reaction, which accounts for heptane or diesel oil combustion. In addition, the pool surface must can be defined in two ways: a fuel that burns with liquid pyrolysis prediction governed by the fuel characteristics or a prescribed HRR curve. In Test No. 6 (see Table 1), the concealed pool fire consists of a steel tray of 0.7 x 3.0 m with 50 mm Heptane on a water base of 50 mm. Test No.7 has a concealed diesel tray of 0.74 x 0.74 m and the same fuel/water layer heights than the previous one. As indicated above, Test No.7 is the only one modeled with prescribed HRR curve. Both trays are obstructed from direct impingement of water with the exhaust steel plate at the top of mock-up, as shown in Figure 1. The thermal management (Test No. 8) consists of a steel tray pool fire placed at floor with a squared steel plate concealment at the left of the mock-up. The nominal fire size and squared steel plate size of the thermal management test must be correlated with the test compartment volume, according to the fire

size-scaling Table 2 of IMO MSC/Circ.1165. Interpolation of data results for the DFL 500 m³ test hall leads to a 1 MW fire with a square steel obstruction plate of 2 m sides. For Tests No. 6 and 8, a combustion efficiency of 85% was assumed according with similar pan fire experiments conducted by other works in the past (10,16,27).

The fuel regression rate, the fuel surface area and the heat of combustion of the fuel were used to estimate the HRR of the pan fires in the corresponding experiments. The HRR and the burning duration of pool fires were checked with the help of Nuclear Regulatory Commission (NUREG) Fire Dynamic Tools. The properties of the fuels are listed in Table 3.

Table 3
Combustion and fuel characteristics

Properties	n-heptane	Diesel dodecane
Density (kg/m ³)	684	918
Specific heat (kJ/kg·K)	2.24	1.89
Thermal conductivity (W/m·K)	0.16 (0°C) – 0.14 (100°C)	0.16 (25°C) - 0.15 (100°C)
Emissivity	1	0.9
Heat of combustion (kJ)	44,600	44,400
Combustion efficiency	0.8 - 0.9	0.8 - 0.9
Critical flame temperature (°C)	1,560	1,507
Boiling temperature (°C)	98.4	214
Heat of vaporization (kJ/kg)	318	365
CO yield	0.006 - 0.08 (0.01)	0.006 - 0.012 (0.012)
SOOT yield	0.01 - 0.037 (0.037)	0.01 - 0.2 (0.042)

Sources: (8,10,40–47), most common yields in parentheses

The conduction through tray walls dominates the heat transfer for smaller pool fires, but for medium trays (up to 0.2 m diameter) the convective transport is the most important mode, while the radiative transport dominates for larger pool fires (48,49). Thus, the burning mode in pool fires of Tests 6, 7 and 8 is radiative dominated. FDS fast chemistry combustion assumes ignition over the entire pool surface instantly; however, in a real pool, there is a slight delay for the flame be propagated across the pool. Moreover, the convection within the pool removes heat from the surface layer. For simplicity, FDS do not consider the convection of the liquid within the pool and treats the liquid fuel like a thermally-thick solid for the purpose of computing the heat conduction. The fuel thermal conductivity was introduced in the models with temperature correction from 20°C to 100 °C. In FDS, every surface cell of the pool is treated as an independent 1D solid wall. Initial conditions in pool fire modeled by FDS are quiescent by default, but in a real pool, there will be some form of slight ambient air movement (36). Furthermore, in a grid of 5 or 10 cm cells, those above the pool have uniform fuel vapor and temperature over that entire 5 or 10 cm. These approximations have an impact on the early heating of the pool, so temperatures will rise higher than in real experiments.

2.3.3 Ventilation and air consumption considerations

The dimensions of the test room and openings are designed to allow a free burning of larger design fires in well-ventilated conditions to reach the flashover phase if the water mist is not released.

The possible under-ventilated (ventilation-controlled) condition can be given for larger fires of 6-7 MW (Table 1). When the water mist is released, the compartment is cooled and only partially filled with hot gases, leading to consider a stratified case with partially developed fires and temperatures lower than those of flashover. According to (50), the mass flow of combustion gases through an opening in a stratified pre-

flashover fire can be calculated if some conditions are present: temperatures above 200°C and $0.6 < H_D < 1.4$ (for $H_0 \approx 2$ m) where H_D = interface between hot and cold layer and H_0 = height of the opening. The combustion gas mass flow rate through an opening of area A , can be calculated with the following equation:

$$\dot{m}_g = 0.684 \cdot A \cdot \sqrt{H_0} \cdot (1 - H_D/H_0) \quad (1)$$

The result of this equation varies depending on the smoke layer height, H_D . The air mass flow calculated for the tests with a higher HRR require an H_D lower than 0.76 m. However, in the simulations conducted with water mist, the hot layer is higher than 1.5 m, and thus, the opening is not sufficient for the flow of combustion gases and air consumption entrance in tests with larger fires (Nos. 1, 2, 3, and 6). As stated above, the water mist controls the fire and the enclosure is only partially filled with hot gases; thus, it is more likely a pre-flashover stratified case, and the results of Eq.1 lead to the assumption that tests with higher HRR are under-ventilated

2.3.4 Water mist modeling and nozzle location

The nozzles used in this work are low-pressure water mist. The ones used in the USCG experiments (29,51) can be classified as Class 2 ($D_{v90} < 400 \mu\text{m}$) water mist and they are similar to those studied by Tanner et al. (52,53) and Heskestad (54): G1 swirl full cone and G3 spiral full cone nozzles, as shown in Figure 3. The swirl full cone nozzle has an internal vane that imparts radial velocity and counterswirl to form a full cone pattern with a vertical angle of 60°. It has a nominal flow of 11.0 liters/min at 7 bar pressure. The spiral full cone is a deflector-type nozzle that creates a crude full cone spray pattern in a tightly controlled spray angle and usually features the largest possible flow rate for a given pipe connection size (52). Its pattern has a vertical angle of 66° and a nominal flow of 10.1 liters/min at 10 bar pressure. The type of nozzle used in DFL test rooms (K6 cluster swirl in Figure 3) is a high-performance nozzle with a more complex design that can be classified as Class 1 ($D_{v90} < 200 \mu\text{m}$). It is a cluster with 7 swirl micro nozzles, one faced down and 6 distributed at 60° (longitude) and oriented at 45° vertically (latitude). The 7 orifices have a spray pattern of 80° full cone as Figure 3 (right) shows, thus the Z axis angle is 2x85°. It has a nominal flow of approximately 15.5 l/min at 8 bar pressure (55).

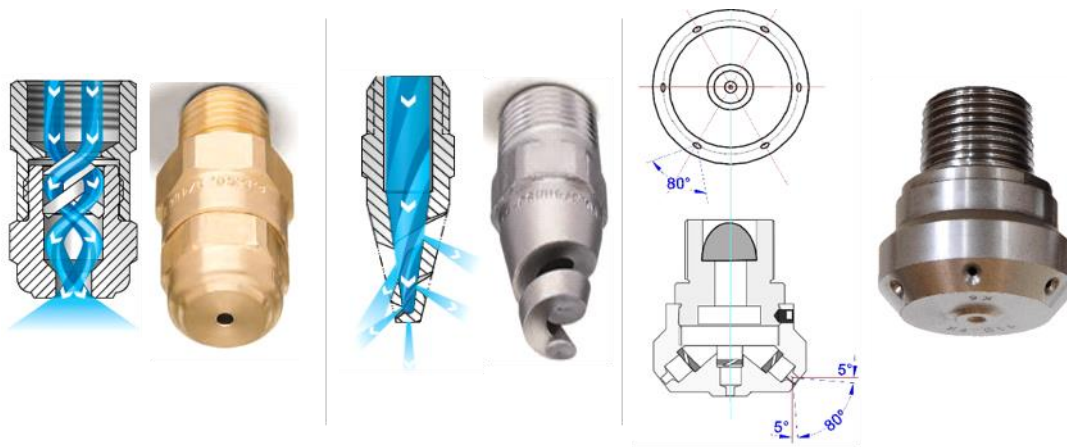


Figure 3. Nozzles: G1 swirl full cone (left), G3 spiral full cone (center), K6 cluster swirl (right)

Table 4 shows the main characteristics of the nozzles. For G1 and G3, the drop size and the velocity are given for three distances (0, 0.5, 1 m) from the center (LoC) and three heights (1, 2, 3 m). In the case of K6, the drop size and the velocity are only available at 0.5 m height and LoC=0.

As explained earlier, water droplets are introduced into the computational domain at a prescribed distance, r_0 from the nozzle using a Eulerian-Lagrangian approach. As a result, the spray boundary consists of a spherical surface defined by r_0 and two elevation angles. The distance r_0 must be within 0.05 m and 0.2 m from the nozzle. The velocities provided by test reports and manufacturers datasheets are related to longer downstream distances (0.5 m, 1 m); thus, the initial velocities, v_o were calculated with a variation of Bernoulli equation.

Table 4
Characteristics of the nozzles

Nozzle Type Pressure	K bar ⁿ /lpm	Angle Z axis	LoC (m)	Drop diameter μm vs height at 1 m DV ₅₀ DV ₉₀		LN-RR constant γ D _{min} (μm)	Velocity(m/s) (manufacturers datasheets) 1 m 2 m 3 m			Velocity v_o (m/s) (FDS input) $r_0 = 0.1$ m
G1 Full cone swirl P = 7 bar	4.3 $n \approx 0.47$	1 x 60°	0	155	355	2.1 44	4.9	1.7	2.1	12
			0.5	310	505		1.2	0.8	0.9	
			1	390	530		--/--	--/--	--/--	
G3 Spiral nozzle P = 10 bar	3.2 $n \approx 0.5$	1 x 66°	0	127	186	3.8 50	2.8	1.4	0.8	10
			0.5	209	273		1.4	0.7	0.2	
			1	349	381		--/--	--/--	--/--	
K6 Cluster swirl P = 8 bar	5.6 $n \approx 0.5$	2 x 85°	0	100	200 at 0.5 m height	0.75 18	5 at 0.5 m			20

These initial velocities v_o were tested in simulations using the phase Doppler particle analysis (PDPA) device feature of FDS as Fig.5 shows. The results at 0.5 m and 1 m were compared with the velocities shown in Table 4, which allowed the initial velocities of the G1 and G3 nozzles to be adjusted. The results for the K6 nozzle were accurate; thus, these velocities at 0.1 m were used in the nozzle spray pattern table in the simulations.

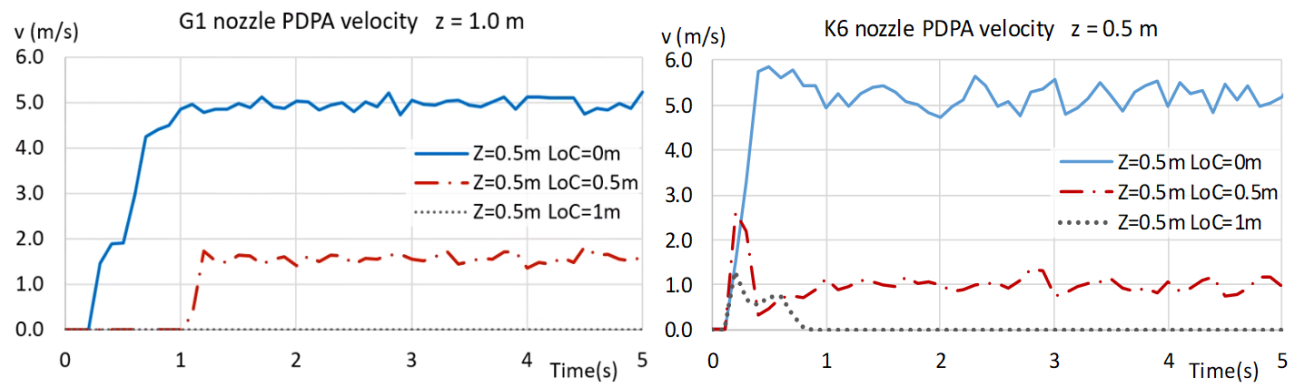


Figure 4. Simulation of PDPA velocities, G1(left) and K6(right) nozzles at z=1 and 0.5 m respectively

The nozzle characterization is similar to the fuel spray one: the water drop particles, properties of the nozzle and situation must be specified with FDS keywords. Among the drop characteristics, size distribution options are Constant, Rosin-Rammler, Lognormal and Rosin-Rammler-Lognormal (default). The water flow can be specified directly or by pressure and K factor. The particle injection rate (Nps), by default is 5,000 but higher

values were used 50,000 for shorter simulations and 20,000 for larger ones. The jet stream pattern can be defined as conical, elliptical or a complex spray pattern table. In the case of a multi orifice nozzle like the K6 one, the 7 orifices can be modeled using a table of sprays or using 7 individual single orifice nozzles placed at the same location with the latitude and longitude orientations of the nozzle cones. The spray table has 7 lines, one for each orifice, i.e. the first orifice/line defined as initial latitude and end angle (5.0°, 80.0°), initial longitude and end angle (-40.0°, 40.0°), velocity (30 m/s), and flow fraction (14.28%); for the rest orifices, latitude and longitude angles vary. Both approaches were tested, and the water droplet distribution was found to be similar.

As an example of water drop and velocity of a nozzle, Figure 5 (left) shows a comparison between experimental data (measured by RISE) and theoretical Lognormal-Rosin-Rammler distribution. Figure 5 (right) shows the diameters measured by the PDPA device of FDS. Both graphs are for the K6 nozzle. The distribution of the number fraction droplet (NFD) experimental data, were measured by Research Institutes of Sweden (RISE) through an interferometric technique with double pulsed laser and double-framed camera. Several FDS simulations were conducted by modifying $\pm 5\%$ (acceptance criterion for test equipment calibration) the parameters of the distribution and no significant changes were observed in the results of temperatures and extinguishment times. This is reasonable since the uncertainty tests did not show significant changes for 5% variations in the total mass flow rate, which is a more aggressive modification than a variation of drop size distribution.

Spray characterization simulations were performed in order to check if the velocities and drop sizes calculated at droplet injection distance produce the velocities and drop sizes at 0.5 m or 1m supplied by reports that are shown in Table 4. These simulations use the phase Doppler particle analysis (PDPA) device of FDS, that allows to analyze properties of the water mist as velocities and drop sizes.

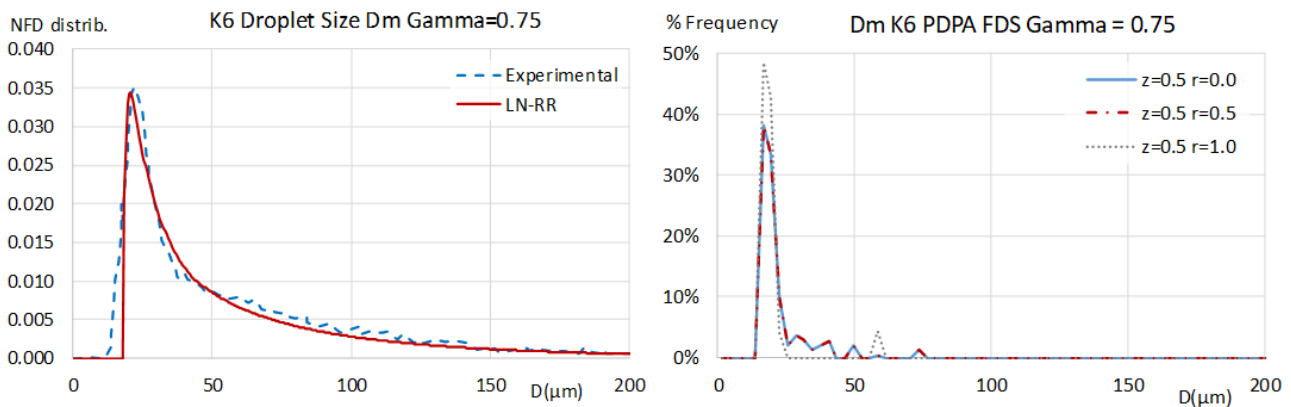


Figure 5 Experimental vs LRR $\sigma=0.8$ (left). PDPA FDS Dm at Z= 0.5 m, LoC= 0, 0.5, and 1m (right).

2.4 Mesh sizes and computational cost

FDS uses a Cartesian grid where the cells can be uniform in size (default mode) or stretched in the three coordinate directions. It is possible to use more than one mesh in a calculation; this allows the creation of an efficient computational domain for geometries that cannot be easily fitted into a single rectangular grid. The use of multiple meshes allows parallel processing, using one or more cores to each mesh.

A mesh sensitivity analysis with 5, 10 and 20 cm mesh sizes was conducted to validate the adequate cell size for the fires modeled in the present work, several tests were utilized for this grid study. The domain was

divided into 11 meshes with sizes of 5, 10 and 20 cm. After study the results, the mesh sizes of 5 cm and 10 cm provided similar temperature and HRR curves. The domain is defined in FDS by MESH keywords; a grid of 9 to 12 meshes was defined. With regard to grid refinement, center meshes that contain engine mock-up where spray or pan fire is modeled, have a fine grid of 5 cm or 10 cm depending on the size of the fire (1 – 7 MW), while the left and right side meshes have a grid of 10 cm. The total number of cells was 949,968. To reduce computational times, FDS allows the use of Message Passing Interface. The simulations with 12 meshes were executed in 2 Xeon nodes with 48 cores and 128 GB of memory, using a parallel process with 2 to 4 cores per mesh. The average calculation cost for a model with 11 meshes and 2 cores per mesh was approximately 22x18 h CPU core time for a 240 s simulation. The CPU nodes are located at a high performance-computing center: Galicia Supercomputing Centre (CESGA).

3. Results and discussion

FDS outputs are csv data sheets with HRR, radiation, thermal convection and conduction, fuel mass consumption, thermocouple temperatures and many measurements of gas and species involved in the fire.

The experiments conducted at USCG follow the IMO guidelines MSC/Circ. 668 (56), and the experiments conducted later at DFL follow the revised IMO guidelines MSC/Circ.1165 (6) and subsequent amendments. Both guidelines determine the extinguishment time according to the following definition: “fire extinguishment is a reduction of the heat release from the fire and a total elimination of flames and glowing parts by means of direct and sufficient application of extinguishing media”.

The reports of both experiments did not include HRR evolution, and only the USCG Tests (Nos. 4, 5 and 7) included O₂ evolution. Therefore, comparison graphs of these variables between experiments and simulations cannot be made. Since the data available from experimental tests are primarily average tree temperature evolution, this section compares the temperature evolution obtained with FDS simulations against that of the experimental tests for all the fire scenarios included in Table 1. For clarity, four of the five thermocouples (THCPs) shown in Figure 1 are compared. In the figures below, the temperature evolution graphs compare the THCPs in pairs: experiments (Ex) vs simulations (S). The graphs on the left show Tree 1, and the graphs on the right show Tree 2. The temperature graphs are complemented with the graphs of HRR, radiative heat and O₂ concentration evolution of the simulations.

3.1 *Spray fire simulations*

3.1.1 *Test No.1 low-pressure heptane spray fire on top of mock-up with 20° steel plate*

This test corresponds to IMO No. 8 in the DFL test hall. Figure 6 shows the temperature evolution of upper THCP of Tree 1 and Tree 2, Figure 7 shows the evolution of lower THCP. The experiment had an extinguishing time of 20 s after a preburn time (time allowed to free-burn before water mist is released) of 15 s. The temperatures reached at Tree 1 are lower than those of Tree 2 because the spray is directed towards Tree 2 and the steel plate with 20° does not deflect the flame to Tree 1. In both cases, the ranges and shapes of temperature prediction graphs are close to the experimental values.

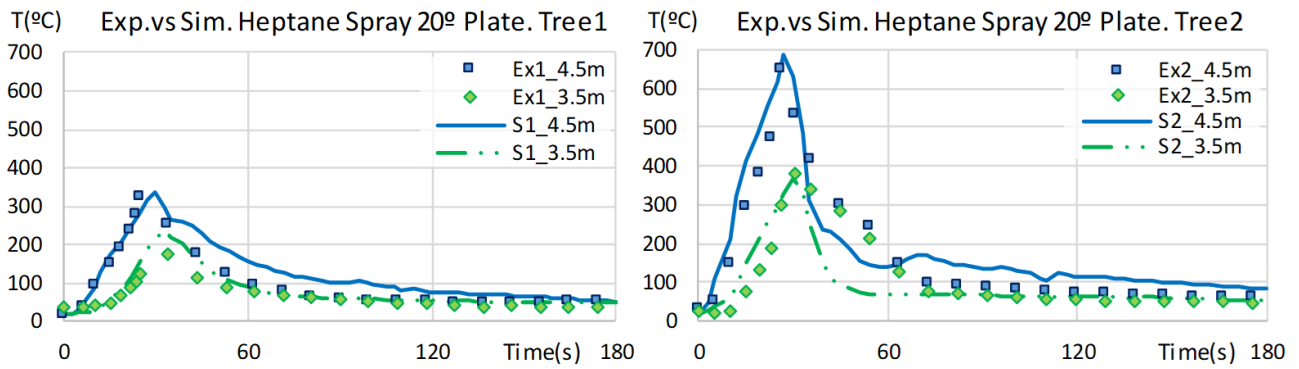


Figure 6. Experiment vs simulation Test No.1 lower temperatures

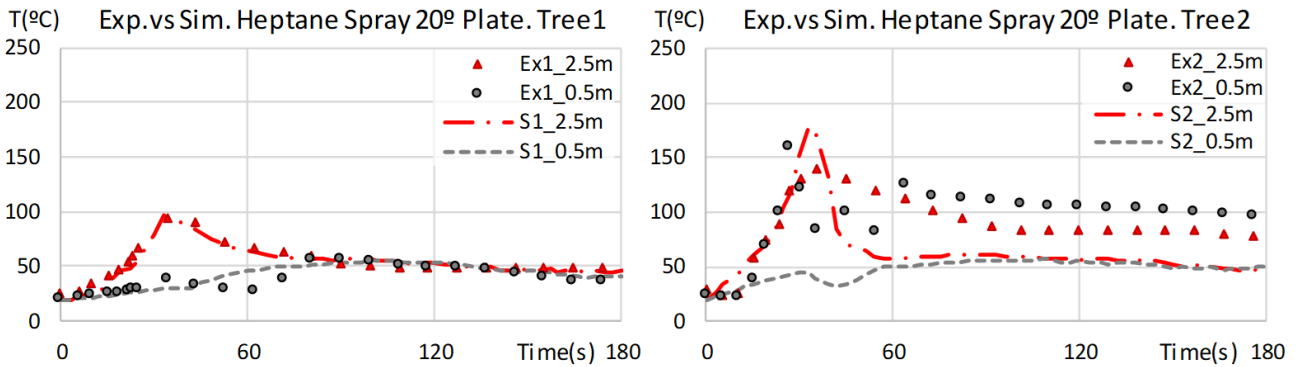


Figure 7. Experiment vs simulation Test No.1 lower temperatures

Nozzles are close to the fire spray; thus, water impingement without obstructions reduces the fire following the trends already identified at earlier naval tests conducted (29,51): unobstructed fires close to water mist nozzles experimented with a fast reduction of temperature and were extinguished in approximately 1 min by direct flame interaction (gas phase cooling, localized oxygen depletion and kinetic effects). On the other hand, the total flow supplied by 9 nozzles was 142 liters/min, the ratio per volume was 0.28 l/m³ and the flow ratio per area was 0.023 l/m²·s, which are higher than the values estimated by Anderson (57) and Wighus (58), thus ensuring the flame cooling and local oxygen depletion effects.

Figure 9 (left) shows the HRR, the radiation, and the O₂ evolution of the Test No. 1. The O₂ is measured by a sensor placed between the fuel nozzle and door, 0.3 m from the back of the nozzle. The HRR drops faster when the O₂ concentration is lower to 15% which is consistent with the results of other experiments (59) with water mist and obstructed fires in IMO tests machinery space of 500 m³, where the average oxygen concentrations at extinction was 14.5% for heptane spray fires and 13.5% for pool fires.

A simulation without water mist was conducted to compare the evolution of temperatures, HRR and O₂. Without the cooling and inerting effects of the water mist, the average temperatures of the compartment are higher, stabilizing at approximately 400°C, while the hot layer reaches the THCP located at height of 0.5m (see Figure 8). Gas temperatures then remain uniform over the entire volume, as a well-mixed case typical of post-flashover fires. Figure 9 (right) also shows that the HRR remains at an average of 5,000 kW throughout the simulation, despite the O₂ concentration falling to 10%, due to the higher temperatures reached without the water mist application.

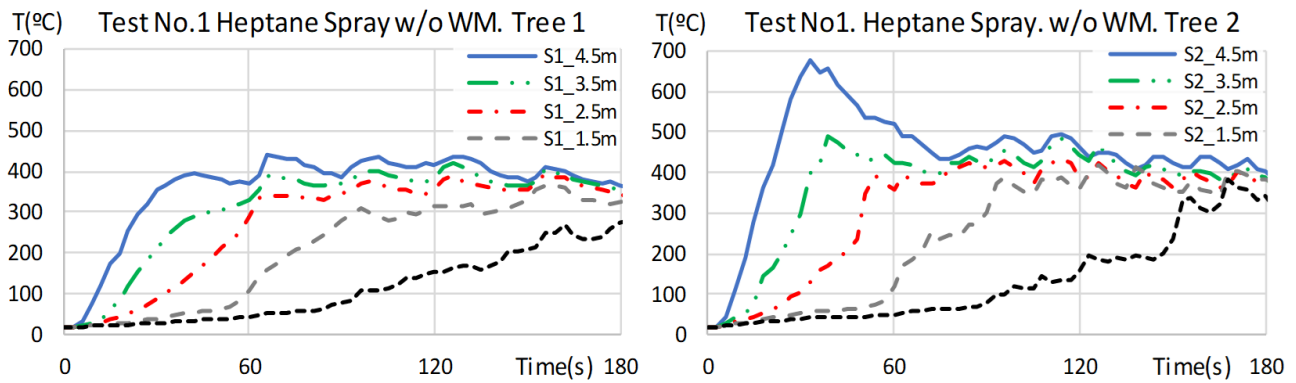


Figure 8. Temperature evolution Test No.1 simulation without water mist. Tree 1 (left) and Tree 2 (right)

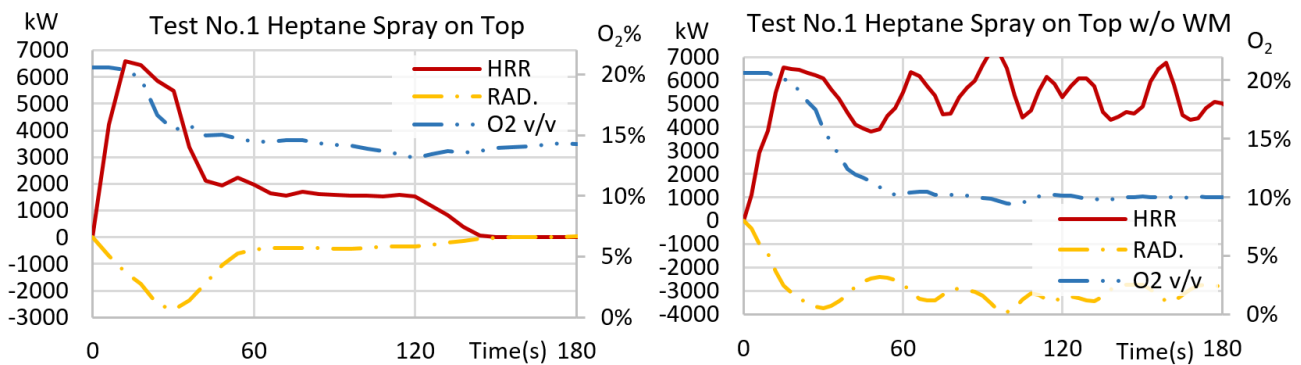


Figure 9. HRR, Radiation and O₂ concentration for Test No.1 simulation with water mist (left) and without water mist (right)

Figure 10 shows the 3D simulation of Test No.1. The image on the left is taken at 6 s after fire release, when the HRR of the heptane spray reaches the maximum value of 6.6 MW; water mist was released at 15 s, which cooled down the flames as illustrated by the image of the center, taken at 30 s, when HRR decreased until 5.5 MW. Finally, the image on the right is taken at 60 s, when the fire is extinguished, this time is longer than that of the experiment, in the simulation small flames remain until shortly before 60 s, however the temperatures drop faster than in the experiment.

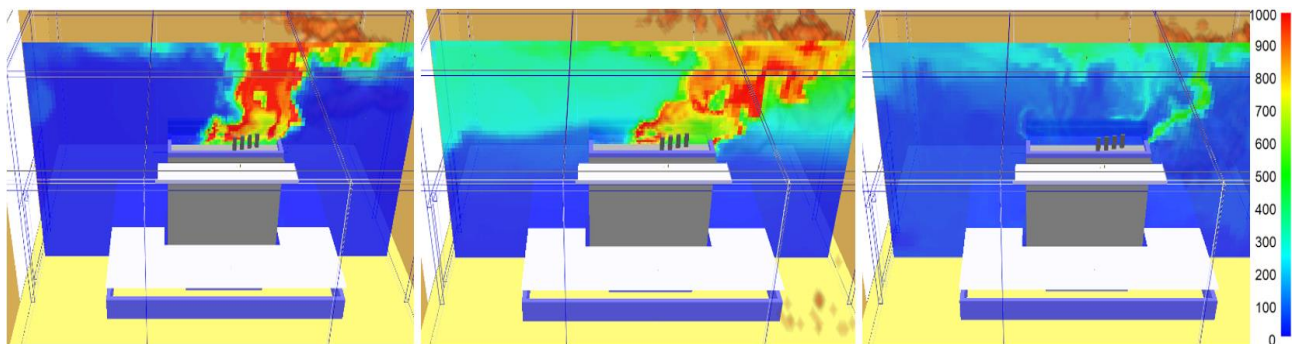


Figure 10. 3D simulation of Test No. 1 with a temperature slice at 6 s (left), 30 s (center), and 60 s (right)

3.1.2 Test No.2 low-pressure diesel spray fire on top of mock-up

This test corresponds to IMO No.1 performed in the DFL test hall, Figures 11 and 12 show the temperature evolution of Tree 1 and Tree 2. In the experiment, the water mist released reduces the temperature, but after approximately 20 s, the flame reignites for one minute to finally be extinguished after 120 s. According to the laboratory manager, the reignition of the fuel is common when suppressing horizontal sprays due to the

following sequence: when the water mist starts suppressing the fire, it loses the ability to burn the fuel, and the unburned diesel falls in the top tray of the mock-up (Figure 1). The fire still heats the rim of the top tray and the fuel boils and reignites later, causing a more significant fire to start. This helps in extinguishing the fire completely as the higher temperatures reached imply more combustion gases trapped underneath the ceiling and oxygen depletion to extinguish the fire. The FDS mixing-limited, infinite fast reaction combustion model could reproduce the reignition phenomenon with an appropriate combination of auto ignition temperature (AIT) and critical flame temperature (CFT). However, the authors are not aware of any similar work in which re-ignition is reproduced at a similar time and HRR magnitude.

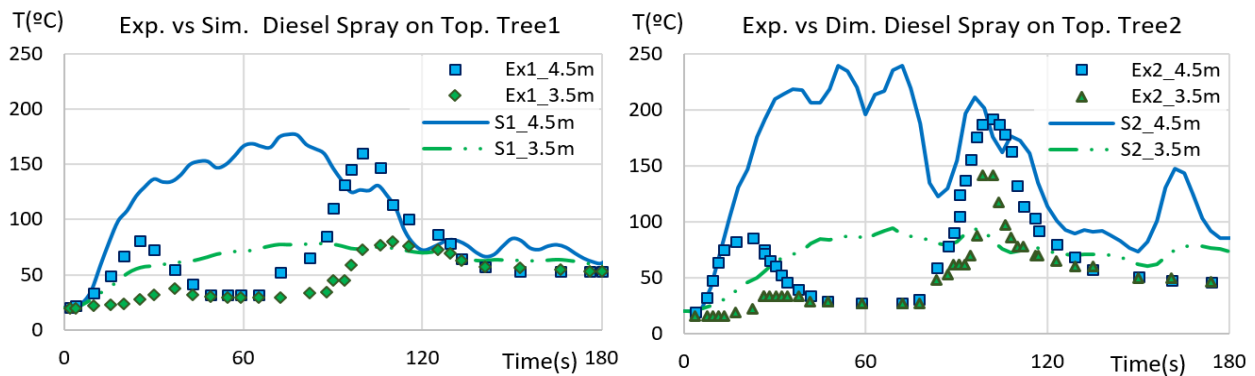


Figure 11. Experiment vs simulation Test No. 2 higher temperatures

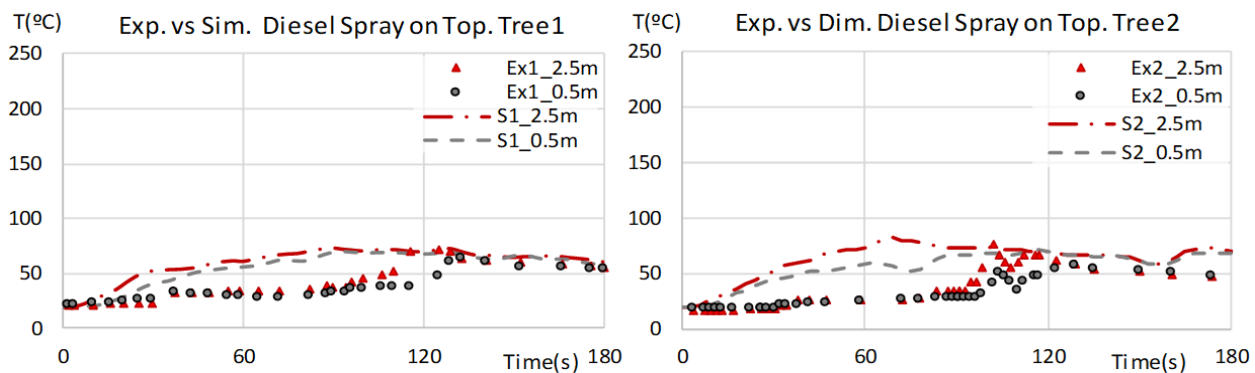


Figure 12. Experiment vs simulation Test No. 2 lower temperatures

Figure 13 (left) shows the HRR, the radiation, and the O_2 evolution of the Test No. 2. The O_2 is measured by a sensor placed between the fuel nozzle and the door, 0.3 m from the back of the nozzle. The HRR begins to decay at 60 s, and the combined mechanisms of water cooling, oxygen depletion and consumption begin to affect the flame at 60 s. The O_2 concentration is approximately 14.5% at this time, which agrees with the value in Test No.1.

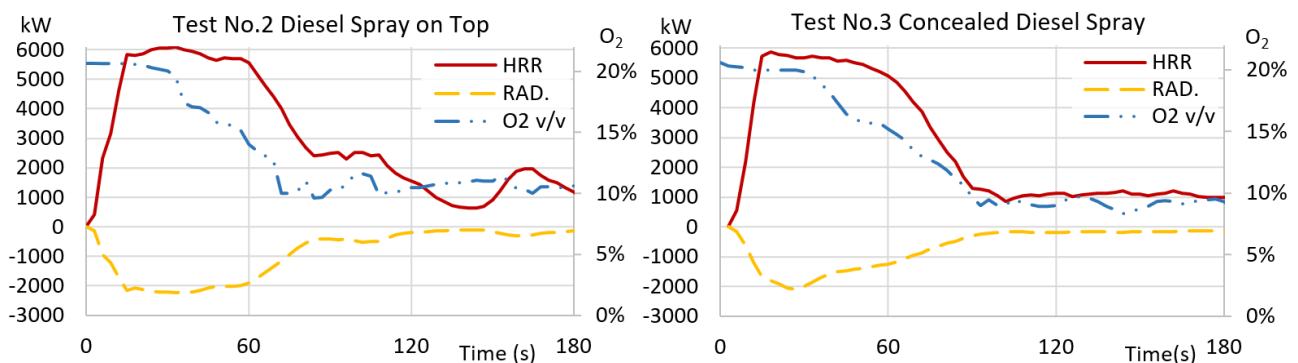


Figure 13. HRR, Radiation and O₂ for simulation of Test No.2 diesel spray on top (left) and Test No.3 concealed diesel spray (right)

Figure 14 shows the 3D simulation of Test No. 2. The image on the left is taken at 6 s when the HRR of the diesel spray reaches the maximum value of 6 MW, water mist is released at 10 s; cooling down the flames as shown in the image of the center, taken at 80 s when the HRR drops to 2.4 MW; finally, the image on the right is taken at 120 s, when the fire is extinguished

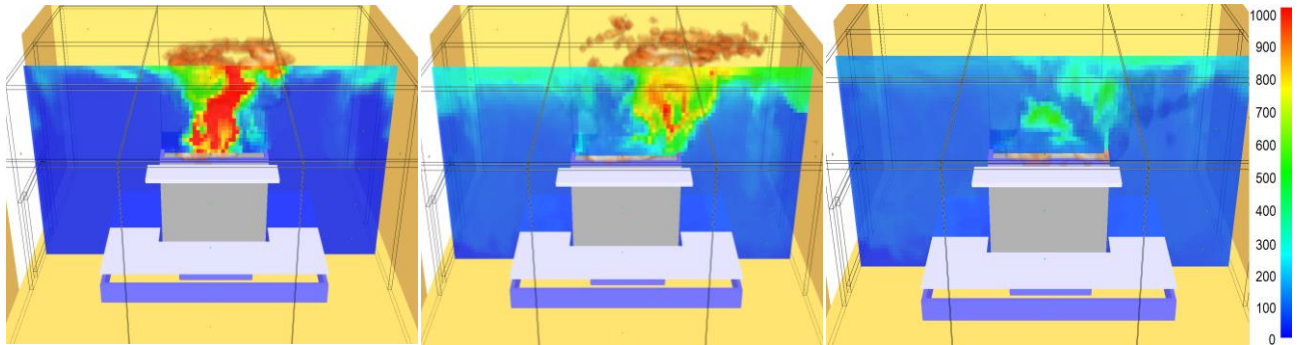


Figure 14. 3D simulation of Test No. 2 with a temperature slice at 6 s (left), 80 s (center), and 120 s (right)

3.1.3 Test No.3 low-pressure concealed diesel spray fire on side of mock-up

This test corresponds to the IMO No. 4 performed at the DFL test hall. Figures 15 and 16 show the temperatures of Tree 1 and Tree 2. Fire consists of diesel spray concealed under an exhaust steel plate of 700 x 3,500 mm, as Figure 1 shows, so the main extinguishment mechanism to consider is indirect flame interaction: global oxygen depletion/inertization and flammable vapor dilution, whereas the secondary effect is the indirect water mist cooling produced by fine droplets of K6 nozzle and the radiation attenuation. The experiment declared an extinguishing time of 120 s with a preburn time of 5 s. In the simulation, the fire is extinguished sooner, which means that FDS may overestimate the aforementioned effects. Nevertheless, the temperatures of the simulation are reasonably close to the real test ones except for the THCP 3.5 m at Tree 1.

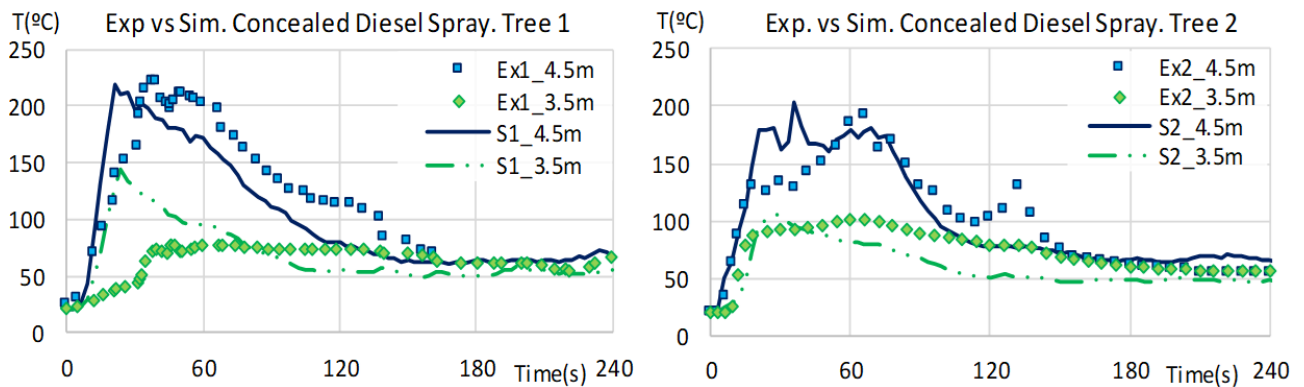


Figure 15. Experiment vs simulation Test No.3 upper temperatures Trees 1 (left) and 2 (right)

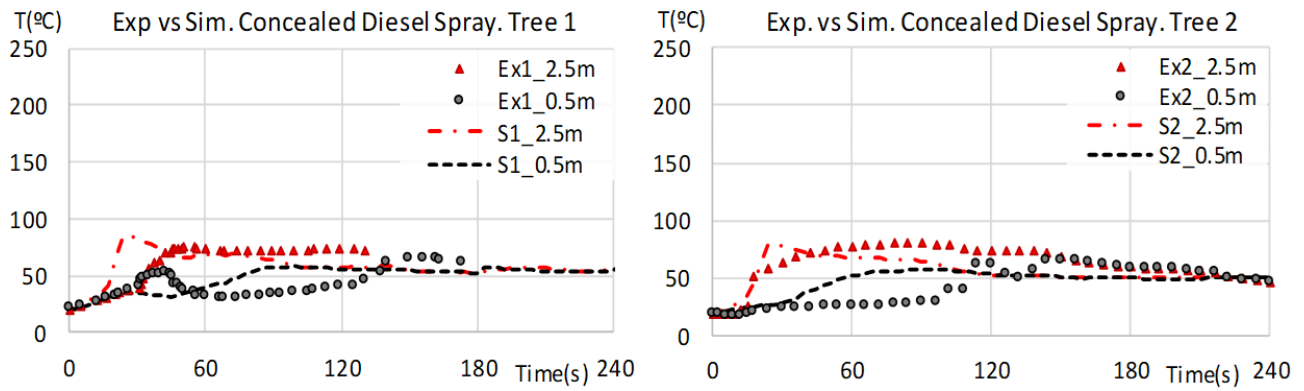


Figure 16. Experiment vs simulation Test No.3 lower temperatures Trees 1 (left) and 2 (right)

Figure 13 (right) shows the HRR, the radiation, and the O_2 evolution of the Test No.3. The O_2 is measured by a sensor placed between the fuel nozzle and the door, 0.3 m from the back of the nozzle. The HRR begins to drop when the O_2 concentration is close to 14.5% which is consistent with the results of other experiments (59) with water mist and obstructed fires in IMO tests machinery space of 500 m³, where the average oxygen concentrations at extinction was 14.5% for spray fires and 13.5% for pool fires.

Figure 17 shows 3D simulation with the temperature slice contours of Test No. 3 at different moments. The image on the left is taken at 8 s, when the HRR of the diesel spray reaches 5.4 MW, which is very close to the maximum value of 5.7 MW; the image in the center is taken at 60 s, when water mist reduces the HRR to 3.4 MW; finally, the image on the right is taken at 120 s when the flames are extinguished.

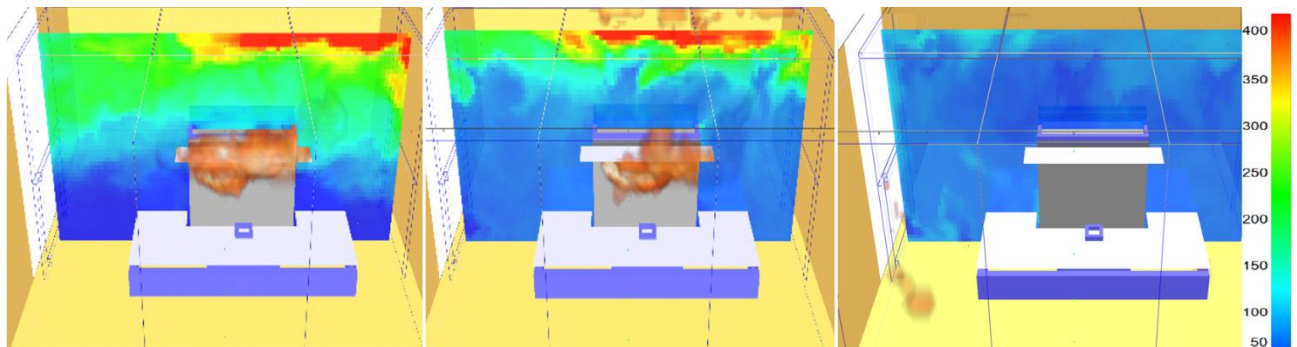


Figure 17. 3D simulation of Test No. 3 with temperature slice at 8 s (left), 60 s (center), 120 s (right)

3.1.4 Test No.4 low-pressure concealed diesel spray fire on side of mock-up

This test corresponds to USCG Test No. 44, based on IMO No. 4 and similar previous No. 3 performed in DFL. The fire is a small low-pressure 1.3 MW diesel spray on side of mock-up and concealed under a steel plate. There are 2 vents of 2 m² each in the room, as Figure 2 (left) shows. The water mist nozzle used is G1, a common swirl full cone nozzle shown in Figure 3 (left), and the preburn time is 60 s.

Figure 18 shows the average room temperatures, in the report of the experiment, the temperature evolution graphs are averaged room temperatures at four heights instead of two THCP trees. The release of water mist reduces the temperatures of the experiment smoothly, while the simulation shows a faster reduction just after the release of water mist, but the THCP located at roof remains near 100°C until 330 s when the oxygen consumption occurs as Figure 18 left shows. The lower THCP at the right show a similar evolution, except for a higher peak at 60 s of the simulation's THCP at 2.5 m height. The mist of these nozzles produces a larger

drop diameter, medium velocity, and thus a high mist momentum. Additionally, 36 nozzles supply the largest flow rate (410 lpm). However, the experiment shows that the fire was controlled at 300 s after the water mist is released (total 360 s), which agrees with the trends already identified in the aforementioned Naval tests (29,51); the obstructed fires were more difficult to extinguish than the exposed fires because less mist reaches the fire in obstructed zones. The total room oxygen depletion mechanism requires a longer time than direct flame cooling and local oxygen depletion. Additionally, small fires are more difficult to extinguish than larger fires in many cases because they consume more oxygen and produce higher temperatures, generating more water vapor.

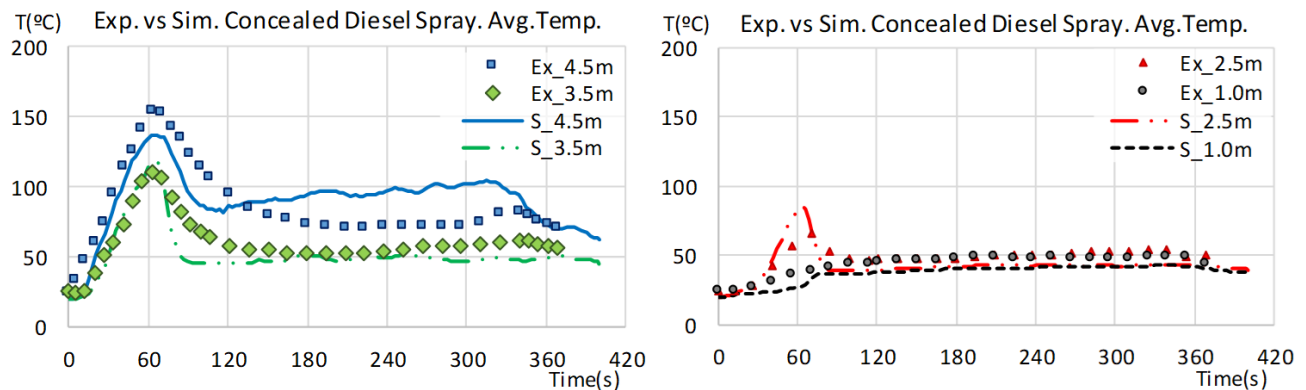


Figure 18. Experiment vs simulation Test No. 4 with 4 m² vent, upper temp. (left) and lower temp. (right)

Figure 19 shows the 3D simulation with a temperature slice. The image on the left is taken at 8 s when the HRR of the diesel spray reaches the highest value of 1.2 MW. The center image is taken at 60 s, when the water mist is released. The image on the right is taken at 380 s, when the fire is extinguished.

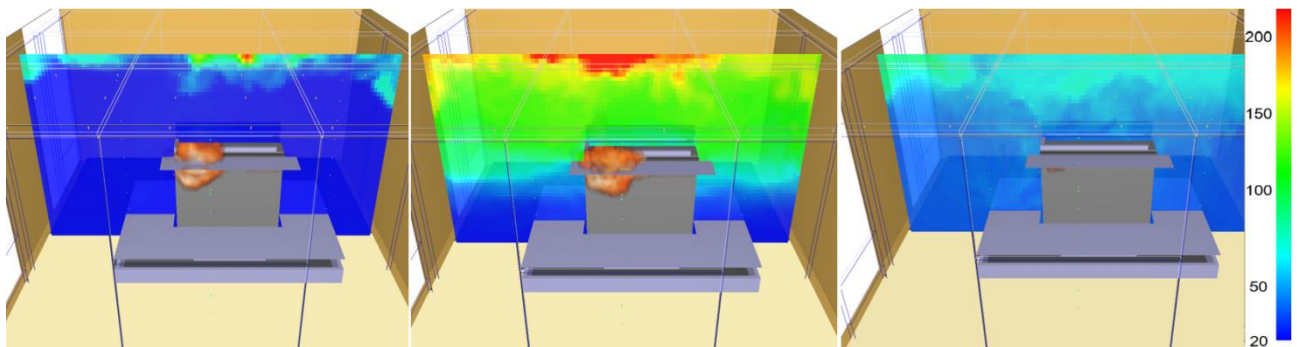


Figure 19. 3D simulation of Test No.4: temperature slice at 8 s (left), 60 s (center), 380 s (right)

Figure 20 shows the average temperature evolution of a test identical to the previous one but with a smaller vent of 1.1 m², which corresponds to the USCG test 46. The fire is controlled earlier than in previous test due to the smaller vents, which causes less oxygen to be available for combustion. As in the previous test, simulations show a faster effect of water mist just after it is released, with the THCPs located at roof remaining near 90°C and the extinguishment time being delayed. It seems that the modeled cooling effect of water mist on the diesel spray is slightly overestimated. Therefore, the burning rate in the simulation is lower than in the experiment, which means that oxygen consumption is delayed.

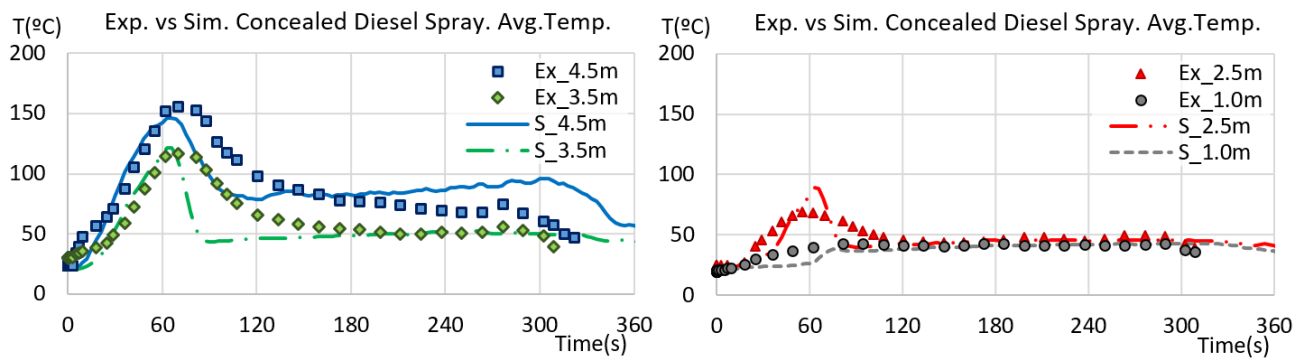


Figure 20. Experiment vs simulation Test No. 4 with 1 m² vent, upper temperatures (left) and lower temperatures (right)

Figure 21 shows the HRR, the radiation, and the O₂ evolution of the two simulations. The results are similar during the first 300 s; however, subsequently, the smaller vent seems to yield a faster reduction in the HRR (Figure 21 right). In the test with a 4 m² vent, the fire is controlled after 370 s as shown in Figure 21 left, which agrees with the experimental results. In the test with a 1.1 m² vent, the fire is controlled earlier due to the smaller vents.

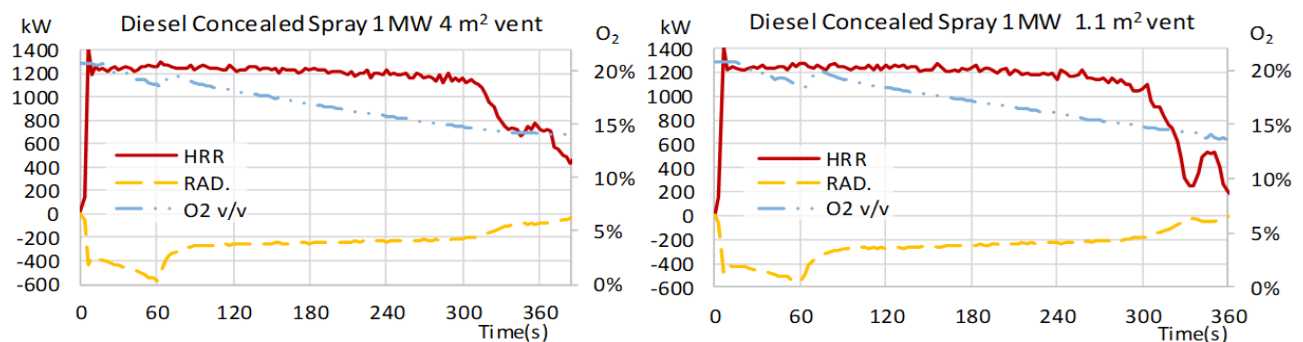


Figure 21. Simulation of HRR, Radiation and O₂ Test No.4 with 4 m² vent (left) and 1.1 m² vent (right)

3.1.5 Test No. 5, local application water mist concealed low-pressure diesel spray fire on side.

This test corresponds to USCG test No. 75, a concealed diesel spray extinguished by local application water mist supplied by 9 (3x3) G3 nozzles at 7 bar, spaced 1 m and located 2 m over the mock-up. The fire is the same as that used in the previous Test No. 4: a diesel spray of 1.3 MW located on the side of the mock-up under the exhaust steel plate as shown in Figure 2 (left). The ventilation conditions are also specified as a 6 m² vent with a 170 m³/min blower located on the roof. The THCPs are located overhead, they are supposed to be at the front, center and back of the compartment as for other USCG tests (Figure 2 left).

Figure 22 (left) shows the front and back overhead temperatures. The release of water mist slightly reduces the temperature of the THCP; however, the THCP temperature then rises to 130°C followed by a consistent reduction at the end of the test but without extinguishment, which may be due to local application water mist limited ability against obstructed fires. Simulation shows also a small reduction in the temperature when the water mist is released followed by a growth and a higher temperature at the end. Figure 22 right shows the evolution of mass loss rate, HRR and oxygen, the fire size was unaffected by the application of the mist following the trends already identified at earlier Naval tests conducted (29,51) where the local application of water mist had any effect in small obstructed fires (1.0-3.0 MW), in contrast, large fires (6.0 MW) were reduced on the order of 10-50% depending on the system. The evolution and final concentration (16%) of the oxygen

in the simulation is very similar to the experiment. Two additional simulations were conducted to study the influence of the air flow in the temperatures and the extinguishing time, but no definitive conclusions can be drawn.

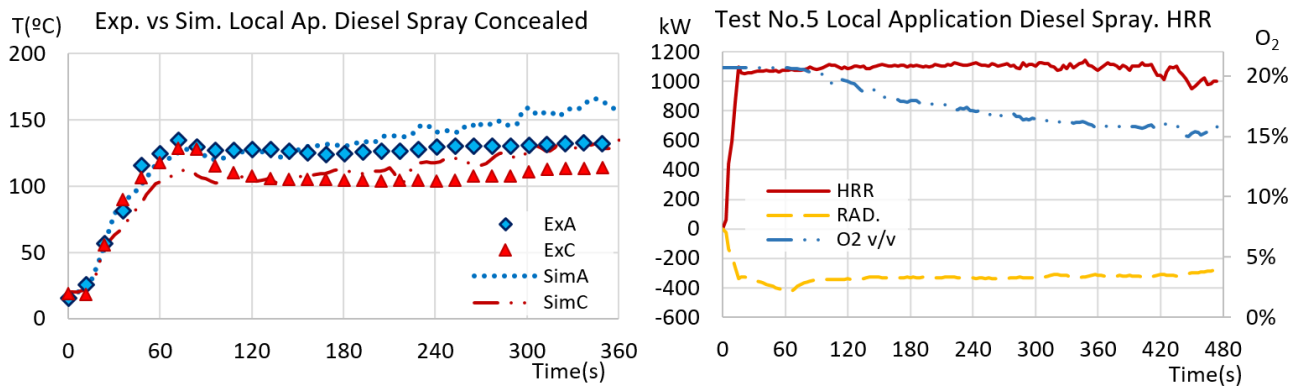


Figure 22. Experiments vs simulation average temperatures (left). HRR, O₂ (right) simulation test No.5

Figure 23 shows the 3D simulation with a temperature slice at three different moments. The image on the left is taken at 8 s, when the HRR of the diesel spray is 1.1 MW, close to the maximum value. The center image is taken at 60 s (preburn time), when water mist has just been released and water from local application nozzles on top of mock-up can be seen. The image on the right is taken at 360 s. Color slices show that the temperatures are higher 300 s after the water mist discharge. It is interesting to note that the 9 nozzles produce a water application rate of 22.8 g/m²s, which is lower than the values estimated by Wighus (58,60,61); thus, the cooling and oxygen depletion/inertization effects are not achieved.

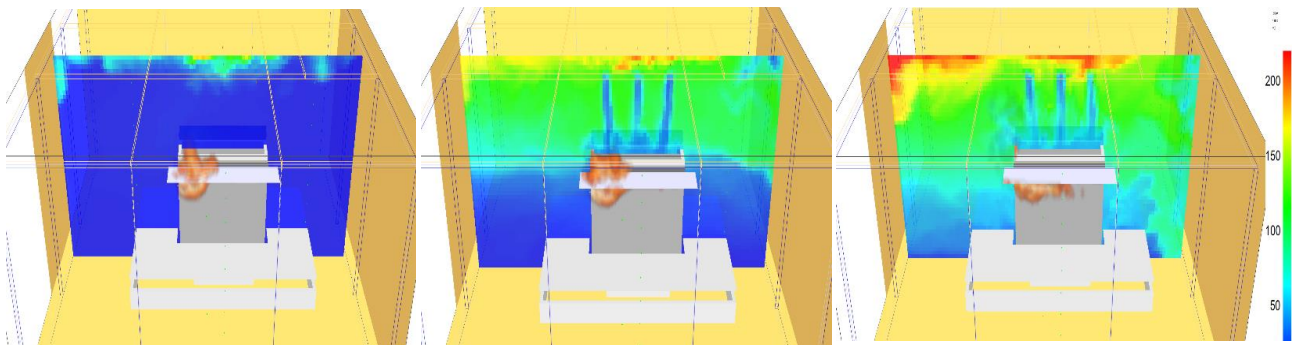


Figure 23. 3D simulation of Test USCG No.75 with temperature slice at 8 s (left), 60 s (center), 360 s (right)

3.2 Pool fire simulations

3.2.1 Test No.6, concealed 3x0.7 m heptane tray on top of bilge at the side of mock-up

This test corresponds to IMO No. 5, Figure 24 shows the higher THCP temperatures of Tree 1 and Tree 2 and Figure 25 the lower THCP. The upper THCP reached 150°C and dropped in less than 2 min after the water mist was released, with a registered extinguishing time of 505 s. Water mist produced by K6 nozzle seems to achieve an adequate combination of small droplets and momentum, thus performs adequately for a concealed pool fire where indirect flame interaction are the primary mechanisms: global oxygen depletion and indirect surface wetting/cooling that reduces the pyrolysis mass loss rate. Considering that the heptane flash point is lower than the ambient temperature, the water mist cooling effect on the pool fire surface has less importance here than in the diesel pool fire of the next Test No.7.

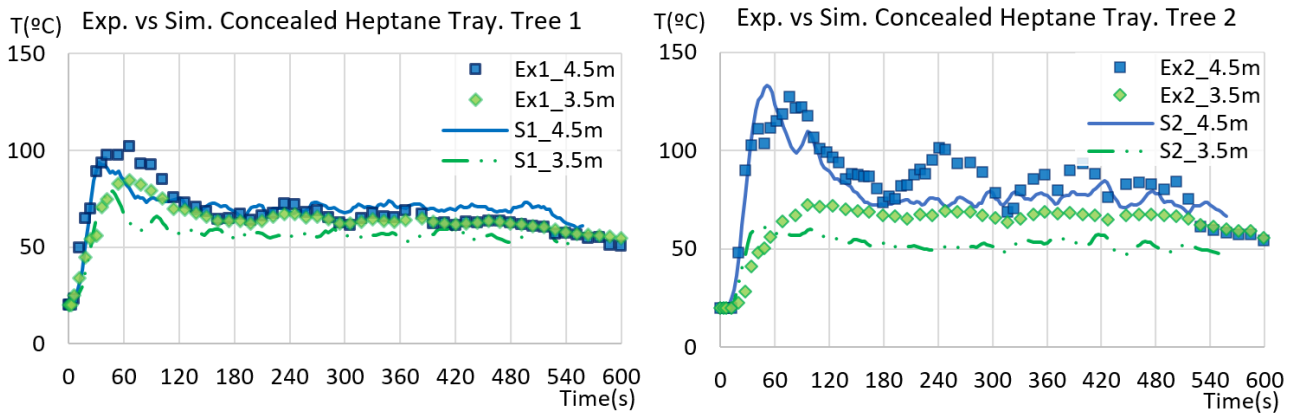


Figure 24. Comparison of upper temperatures experiment vs simulation for Test No. 6.

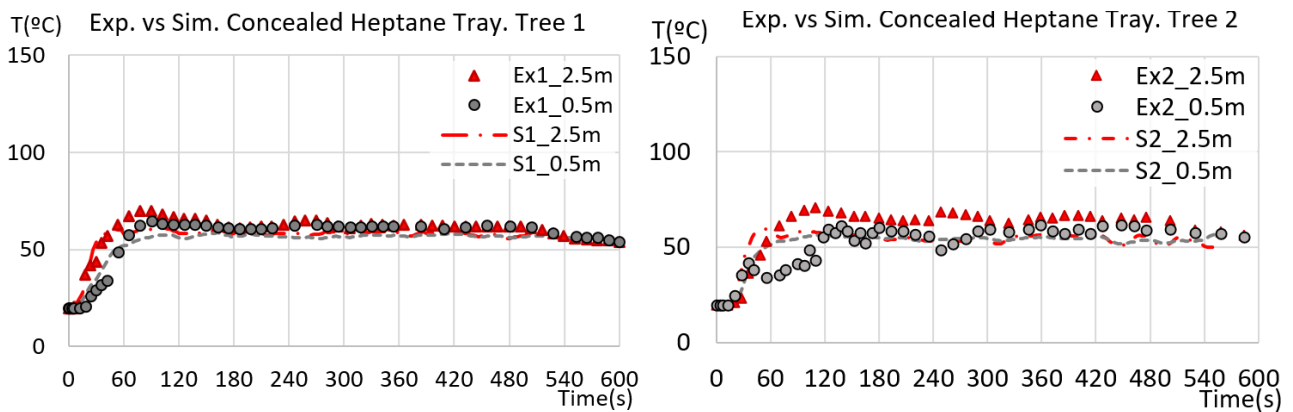


Figure 25. Comparison of lower temperatures experiment vs simulation for Test No.6

The simulation matched the temperature evolution of the experiment quite well for both trees despite the early heating of the pool due to the aforementioned simplification of FDS liquid fuels modeling. Nevertheless, the higher THCPs of the experimental Tree 2 record higher temperatures and there are oscillations that are not reproduced in simulations.

One of the O_2 sensors in the simulation is placed 0.4 m above the heptane pool fire. Figure 26 (left) shows as the O_2 concentration above the pool fire sinks fast from the beginning of the simulation, and stabilizes around 13%, which again is consistent with the Back et al. work that pointed out a concentration of 13.5% for pool fires when the extinguishment occurs.

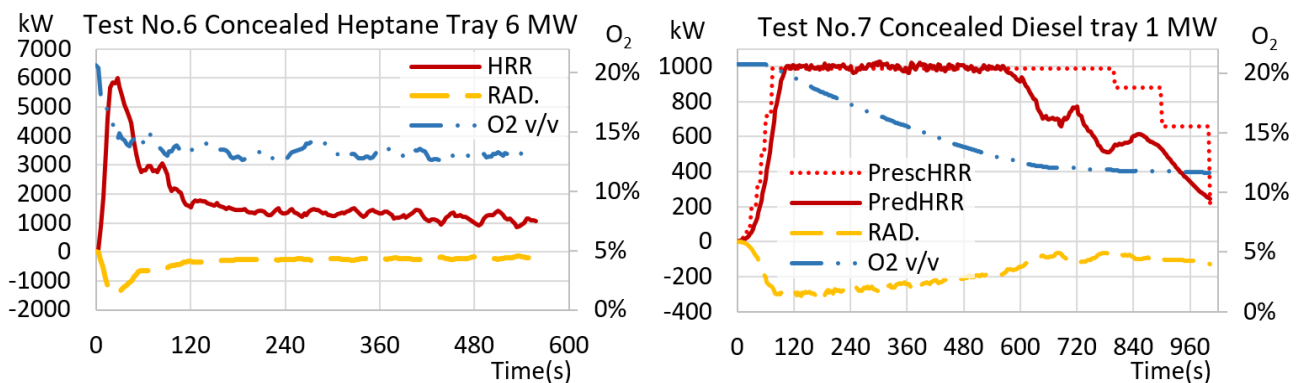


Figure 26. HRR, $\dot{m}$, O_2 for simulation of heptane pool fire Test No.6 (left) and diesel pool Test No.7 (right)

Figure 27 shows the 3D simulation of Test No. 6 with temperature slices. The image on the left is taken at 12 s when the HRR of the heptane pool fire reaches 6.6 MW. The center image was taken at 30 s, the water

mist was released 25 s before and the HRR decreased until 5.3 MW, but the temperatures reached the highest level. The image on the right is taken at 570 s, when temperatures are lower than 60°C in almost the entire room. The experiment reported an extinguishment time of 505 s including a preburn time of 15 s, so the simulation delayed slightly fuel and oxygen consumption. It seems that the modeled cooling effect of water mist in the concealed heptane tray is slightly overestimated, so the fuel combustion rate is slower than in the experiment.

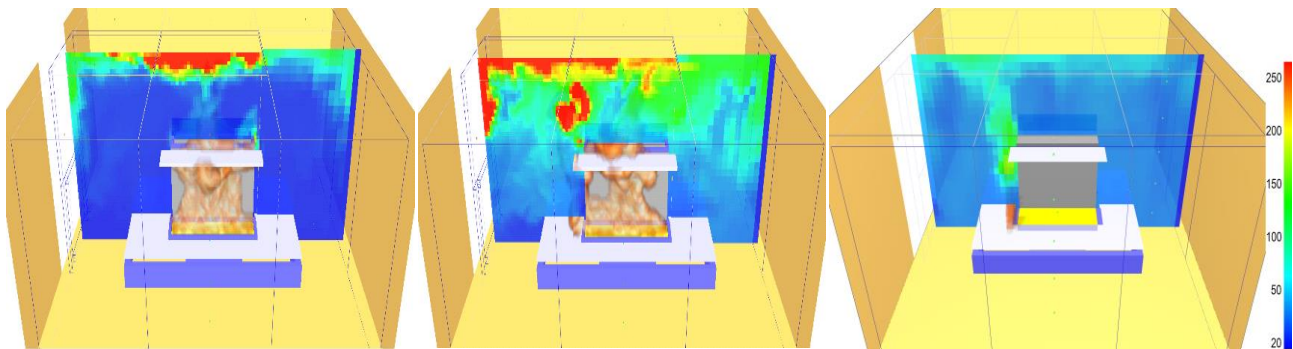


Figure 27. 3D simulation of Test No. 6 with temperature slice at 12 s (left), 30 s (center), 570 s (right)

3.2.2 Test No. 7, diesel tray fire of 1 MW concealed on top of the bilge on side of mock-up.

This test corresponds to USCG Test No. 54. It is based on IMO No. 5 as the previous test No. 6, but the fire pan is smaller and the fuel is diesel instead of heptane. The water mist nozzle used is G1, a full cone swirl (Figure 3 left). The preburn time is 60 s, and there are two vents of 2 m². Figure 28 shows the average room temperature. In the experiment, the water mist succeeds in maintaining the temperatures of the upper THCPs lower than 100°C in the experiment, with an extinguishment time of 852 s including a preburn time of 60 s. The simulation also maintains the temperature of the upper THCPs lower than 100°C. Furthermore, simulations without water mist were performed to verify that the average temperature at the roof was much higher, reaching 250°C.

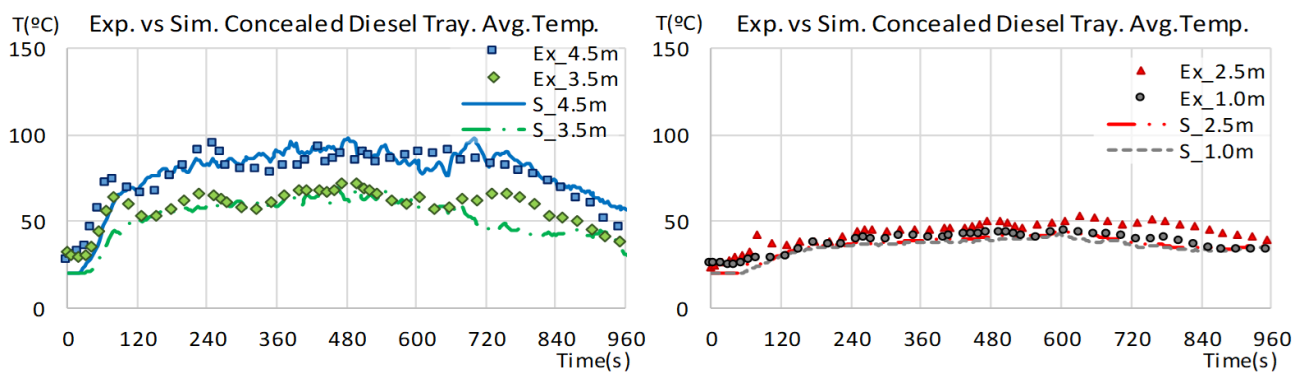


Figure 28. Experiment vs simulation Test No. 7, upper temperatures (left) and lower temperatures (right)

The long time required to extinguish the fire agrees with the trends already identified in the aforementioned Naval tests; small obstructed fires were more difficult to extinguish than larger fires because small fires consume less oxygen, and the hot layer temperatures are lower, generating less water vapor. Total room oxygen depletion mechanism requires both water vapor generation and O₂ consumption, which are low in small fires. Figure 26 (right) shows the HRR prescribed, an input for the model, and the HRR predicted by the simulation,

which remains at 1 MW until 600 s and then drops when the O₂ concentration falls below 14%. The steady state O₂ concentration in the experiment was 15.6%, and the simulation showed a lower concentration.

Figure 29 shows the 3D simulation of Test No. 7 with temperature slices. The image on the left is taken at 120 s when the HRR of the diesel pool fire reaches 1 MW. The center image was taken at 780 s, when the HRR decreased to approximately 500 kW when the fire is controlled, and the temperatures of higher THCPs begin to drop. The image on the right is taken at 930 s when the fire is extinguished, and the temperature of the room has dropped to between 40 and 70°C, as shown in the temperature slice of Figure 29 right. The extinguishment time of the simulation is longer than that of the experiment, however, the simulation matched the temperature evolution of the experiment well. Previous USCG simulations (Test Nos. 4 and 5) show a faster cooling effect combined with a delay of the extinguishment time, which could mean that oxygen depletion and consumption effects of nozzles G1 and G3 with large droplets are not accurately simulated when the hall has large vents or blowers.

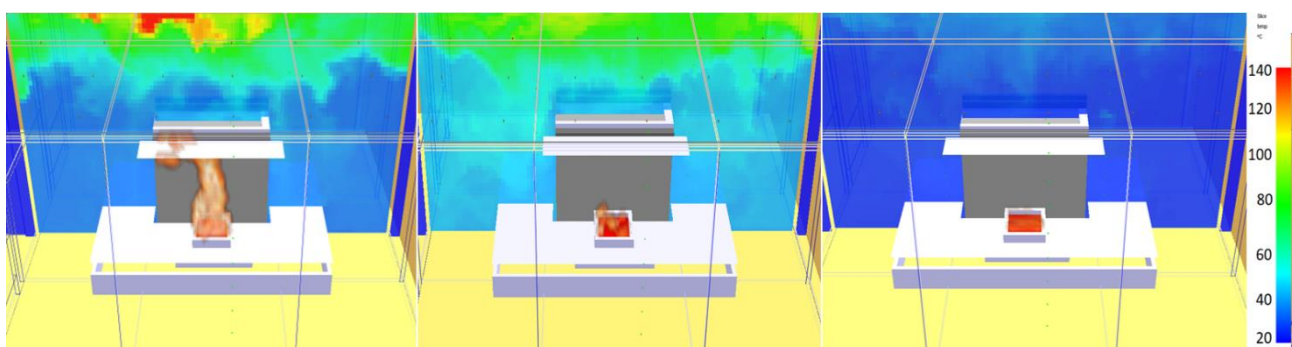


Figure 29. 3D Simulation of Test no.7 with temperature slice at 120s (left), 780s (centre), 930s (right)

3.2.3 Test No. 8, thermal management, obstructed heptane tray

For this test, the nominal fire size must be correlated to the test compartment volume according to a fire size-scaling table 4 of the IMO MSC/Circ1165. The interpolation result for the 500 m³ hall is 1 MW with a 2x2 m conceal steel plate. The requirement for this test is to keep the average temperature of every THCP below 100°C no later than 300 s after activation of the system.

Figure 30 shows the higher thermocouples of Tree 1 and Tree 2 and Figure 31 shows the lower thermocouples of the Trees. Both figure show the thermocouples remaining near 50°C for the entire experiment duration, so the acceptance test is accomplished. As noted above, the water mist produced by the K6 nozzle adequately performs global oxygen depletion and surface wetting/cooling effects. Furthermore, the water mist flow supplied by 9 nozzles in the DFL small hall produces a higher ratio per volume and per area than values estimated by Andersson (57) and Wighus (58), which ensures cooling and oxygen depletion/inertization effects.

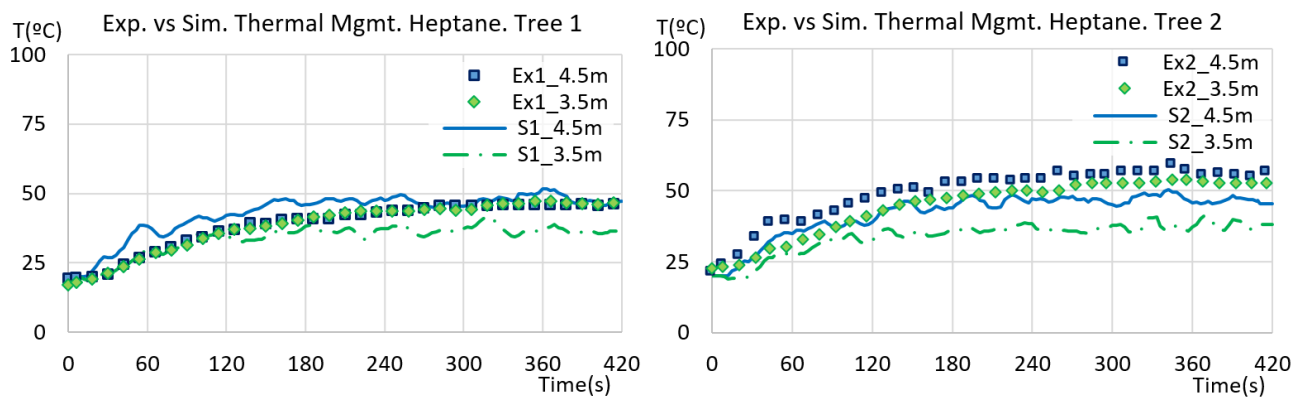


Figure 30. Experiment vs simulation Test No.8 upper temperatures Trees 1 (left) and 2 (right)

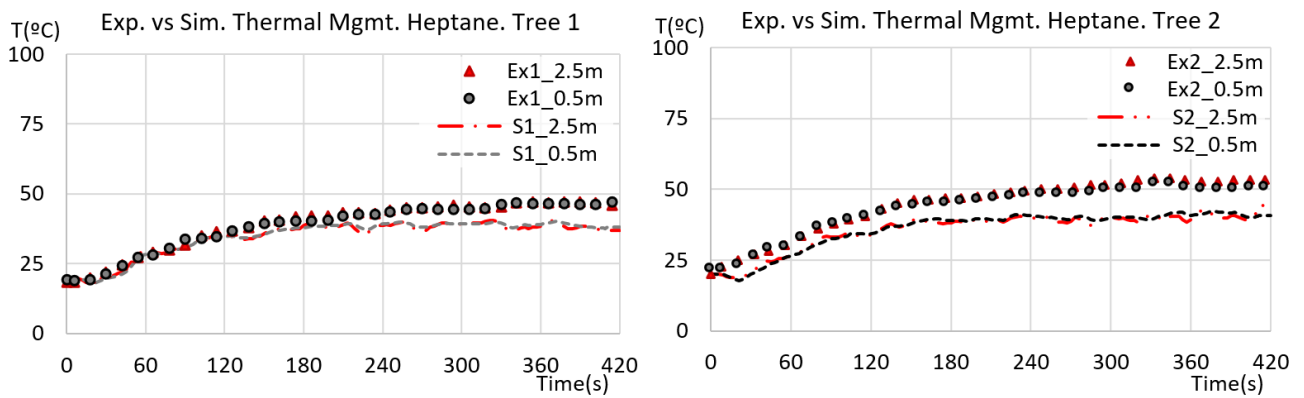


Figure 31. Experiment vs simulation Test No.8 lower temperatures Trees 1 (left) and 2 (right)

3.3 Uncertainty analysis

In this section the uncertainty of the model is analyzed based on a comparison between experimental data and model predictions of temperatures and extinguishment times. In addition, a sensitivity analysis is made to assess the impact of the fuel and water flow uncertainties in the simulation results.

This analysis requires a knowledge of the measurement uncertainties with enough detail, which in this case is not feasible. Therefore, estimations of measurement uncertainty are made according to the summary statistics available in the FDS Validation Guide (8) for the different quantities examined. This includes the following standard deviations σ_E : 0.08 for the burning rate, 0.07 for the Hot Gas Layer (HGL) temperature with natural ventilation, and 0.1 for the extinguishment time.

The uncertainty statistic parameters of the models linked with temperature evolution were determined for each thermocouple tree of the tests: standard deviation σ_M and the bias δ . Table 6 shows the final results of the performed simulations with the extinguishment time and the uncertainty statistics. It can be observed that the predicted extinguishment times only differ significantly from the experiments in the first Test.

Concerning the uncertainties of spray fires, Test No.2 provides poor results due to the complex phenomena of reignition. Predictions of the rest of spray fires are reasonably close to the experimental data, and show better results than the aforementioned set of water mist suppression models included in the FDS Validation Guide with $\sigma_M = 0.47$ and bias $\delta = 0.99$.

The statistics of the pool fire Tests 6, 7 and 8 provided good results. Sikanen and Hostikka (62) stated that heat fluxes of pool fires tend to be under-predicted and the 68% confidence interval for the predictions is $\pm 30\%$ of accuracy. Specifically, heat fluxes resulting from flame impingement were predicted with standard deviation $\sigma_M = 0.37$ and bias $\delta = 0.93$, which are higher than results shown in Table 5.

Table 5
Fire extinguishing tests results and uncertainty

Test No.	Fire Type	Fuel	Experiment* exting. time(s)	Model exting. time(s)	Uncertainty**			
					σ_{M1}	σ_{M2}	δ_1	δ_2
1	Spray on top exposed	Heptane	15 + 20	15 + 45	0.24	0.27	1.10	1.12
2	Spray on top exposed	Diesel	5 + 106	5 + 110	0.37	0.48	1.63	1.87
3	Spray side concealed	Diesel	5 + 120	5 + 115	0.36	0.25	1.13	1.03
4	Spray side concealed	Diesel	60 + 300	60 + 320	0.16		0.92	
5	Spray side concealed	Diesel	No, T=130°C	No, T=120°C	0.11		0.89	
6	Pool fire side concealed	Heptane	15 + 490	15 + 555	0.10	0.16	0.98	0.93
7	Pool fire side concealed	Diesel	60 + 792	60 + 900	0.19		0.87	
8	Pool fire under bilge	Heptane	T<100°C in less than 300s	T<100°C in less than 300s	0.11	0.03	0.95	0.81

* Data from report: preburn + extinguishment time after preburn

** DFL test uncertainty of Tree 1 and Tree 2, for USCG uncertainty of average temperatures at 4.5 m.

Figure 32 shows uncertainty graphics for the most representative spray fires tests: Test No.1 for heptane and Test No.3 for diesel. Figure 33 shows the uncertainty graphics for the most representative pool fires: Test No.6 for heptane and Test No.7 for diesel.

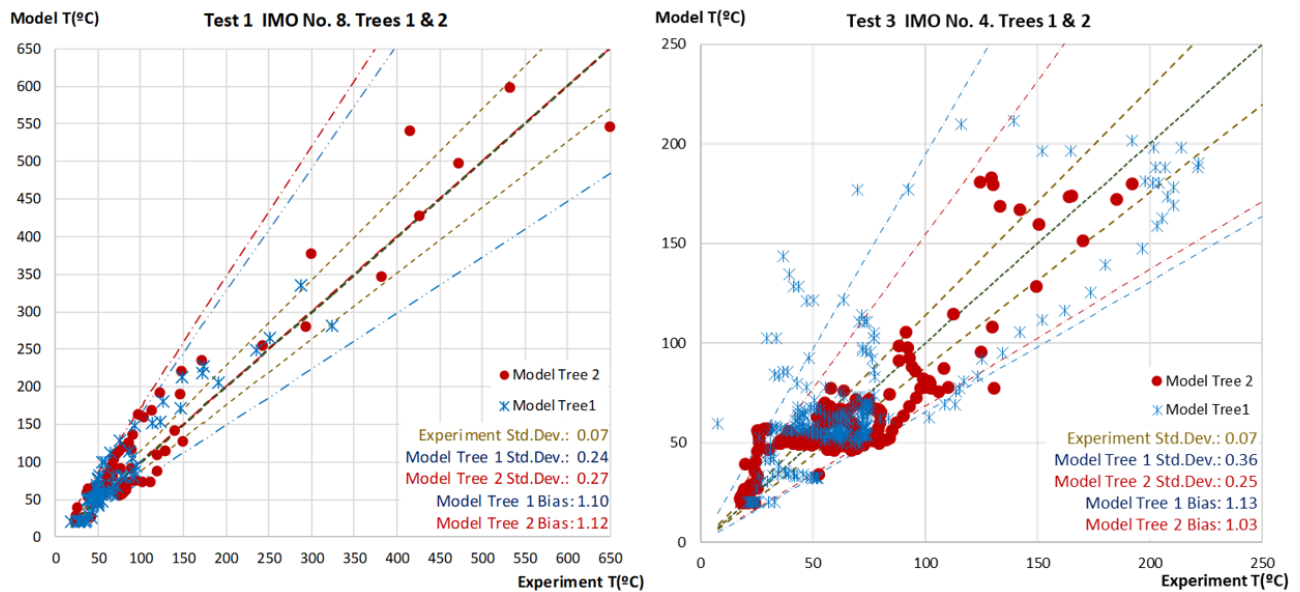


Figure 32. Experiment & Model Uncertainty for spray fires: Test 1 heptane (left), Test 3 diesel (right)

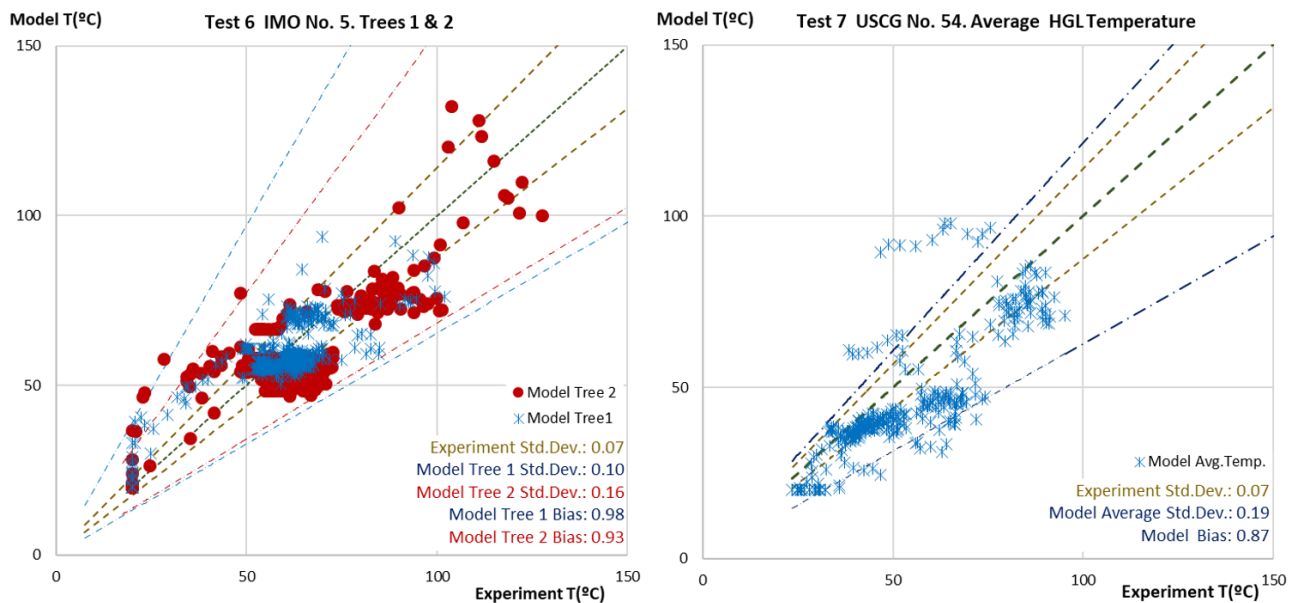


Figure 33. Experiment & Model Uncertainty for pool fires: Test 6 heptane (left), Test 7 diesel (right)

The IMO MSC 1165 establishes a flow of heptane or diesel range of 0.16 ± 0.01 kg/s (9 to 10.2 kg/min ± 0.6 kg/s); this variability of $\pm 6.25\%$ is in line with the accuracy of measurement devices detailed in section 2.3.1. In order to analyze the temperature and HRR sensitivity to this input parameter, a series of simulations with a variation of $\pm 5\%$ of the fuel flow were conducted. Results showed that the average temperature variation over time is slightly under $\pm 5\%$ while the variation of HRR is even lower.

Concerning the water flow sensitivity, simulations were conducted with a variation of $\pm 5\%$ in the water flow. The average temperature difference over time is under 5%, while for the HRR it is even lower.

On the other hand, the IMO MSC 1165 establishes that the average flow constant K of the water nozzles should be verified within $\pm 5\%$ of the manufacturers value. The nozzles tested at DFL were verified, the maximum variation being 4.7% for one nozzle in a sample of 25 units and the measured average variation lower than 3%.

4. Conclusions

A study of low-pressure water mist fire extinguishing in different ship engine room scenarios defined by IMO Circulars was performed by comparing the FDS simulations with the full-scale test results. The simulated fires were based on DFL and USCG tests which included exposed diesel and heptane spray, concealed diesel spray, concealed diesel and heptane tray, and thermal management experiments. The ignition and extinguishment stages were modeled, and the evolution of temperatures was compared against the measured values at different heights in the testing rooms. An analysis of both the experimental and modeled uncertainties was carried out to assess the simulation performance.

The simulation results showed a reasonably good agreement with the experimental results for the heptane unobstructed spray, concealed diesel spray, concealed diesel and heptane tray and thermal management tests in magnitudes, including temperature ranges and reductions during the extinguishment stage. In contrast, differences in the temperature evolution of some thermocouples were identified for the diesel unobstructed spray. The simulations conducted do not accurately describe complex phenomena, such as reignition (deflagrations and backdrafts); this is important for cooled diesel spray fires that loses the ability to burn but then reignites later in the top tray of the mock-up.

Slightly longer extinguishing times were obtained for simulations of concealed diesel and heptane sprays. In these cases, temperature ranges during the heating stage exhibited a slightly faster cooling effect in the model. The same effect seems to occur for concealed diesel and heptane trays, that also presented longer extinguishment times in simulations than those of experiments. However, no conclusions can be drawn about the indirect flame interaction mechanism that caused it but may include oxygen depletion and dilution, and, to a lesser extent radiation attenuation.

A more accurate model configuration and more comprehensive study of FDS extinction modes are needed to obtain results closer to the experiments. As the Technical Reference Guide (36) describes for under-ventilated fire compartment scenarios with a suppression agent such as water mist or CO₂, approximations of the FDS extinction modes can yield insufficient accuracies when modeling the strain between the fuel and oxidizing streams. Thus, the FDS Road Map (63) identifies the modeling of gas phase suppression (flame) phenomena, reignition and backdraft, pyrolysis, water mist atomization and evaporation, interaction of water with flames and smoke, as key areas to be improved. It is also concluded that expertise, skills and more comprehensive input information (e.g. information about nozzle spatial drop size distribution) are essential to obtain accurate results.

On the other hand, both the experiments and simulations show the high performance of the water mist produced with cluster swirl nozzles. Modern low-pressure water mist nozzles have thus achieved a balance between small droplets (for improving oxygen depletion) and momentum (to reach concealed fires and improve surface wetting/cooling effects).

Similarly, the reasonably good agreement between the compartment temperature evolution and fire extinguishment time encourage the application of FDS simulations in a PBD approach to the design of marine fire suppression systems. Designers and engineers can model machinery volumes and nozzle layouts that differ from those tested and approved in the one-story square room of the IMO, and can then predict the temperatures evolution and extinguishing times for a real machinery room for the approval of the authorities.

This study presents consistent simulation results that may serve as a complement to existing validation data sets included in the FDS Validation Guide, while highlighting the challenges of modeling fire suppression by water mist in complex fire scenarios.

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