

Article

Crankcase Explosions in Marine Diesel Engines: A Computational Study of Unvented and Vented Explosions of Lubricating Oil Mist

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Abstract: Accidental crankcase explosions in marine diesel engines are presumably caused by the inflammation of lubricating oil in air followed by flame propagation and pressure buildup. This manuscript deals with the numerical simulation of internal unvented and vented crankcase explosions of lubricating oil mist using the 3D CFD approach for two-phase turbulent reactive flow with finite-rate turbulent/molecular mixing and chemistry. The lubricating oil mist was treated as either monodispersed with a droplet size of 60 μm or polydispersed with a trimodal droplet size distribution (10 μm (10 wt%), 250 μm (10 wt%), and 500 μm (80 wt%)). The mist was partly pre-evaporated with pre-evaporation degrees of 60%, 70%, and 80%. As an example, a typical low-speed two-stroke six-cylinder marine diesel engine was considered. Four possible accidental ignition sites were considered in different linked segments of the crankcase, namely the leakage of hot blow-by gases through the faulty stuffing box, a hot spot on the crankpin bearing, electrostatic discharge in the open space at the A-frame, and a hot spot on the main bearing. Calculations show that the most important parameter affecting the dynamics of crankcase explosion is the pre-evaporation degree of the oil mist, whereas the oil droplet size distribution plays a minor role. The most severe unvented explosion was caused by the hot spot ignition of the oil mist on the main bearing and flame breaking through the windows connecting the crankcase segments. The predicted maximum rate of pressure rise in the crankcase attained 0.6–0.7 bar/s, whereas the apparent turbulent burning velocity attained 7–8 m/s. The rate of heat release attained a value of 13 MW. Explosion venting caused the rate of pressure rise to decrease and become negative. However, vent opening does not lead to an immediate pressure drop in the crankcase: the pressure keeps growing for a certain time and attains a maximum value that can be a factor of 2 higher than the vent opening pressure.

Keywords: marine diesel; crankcase explosion; oil mist; ignition; flame propagation; pressure buildup; explosion venting



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1. Introduction

For a long time, crankcase explosions of the lubricating oil mist in marine diesel engines are considered the most serious accidents [1–7]. The crankcase of a marine diesel engine is known to be filled with liquid lubricating oil, oil mist, blow-by gases, and air. Oil droplets of different size can be produced in several ways: due to the rotation of crankshaft and other engine components [8–10]; due to high flow rates of gas over the lubricant film around the piston ring gaps [11–13]; due to air flow through the piston rings and liner interface [14]; due to oil accumulation and blow-off from oil drain holes and the piston skirt [15]; due to the condensation of oil vapor in cool regions of the crankcase [16–19]; etc. The blow-by gases are the gases leaking to the crankcase mainly from the combustion

chamber through piston rings and defects in the stuffing box mostly during the high-pressure combustion phase [20]. Other sources of blow-by gases are leaky valve stems, turbocharger, etc. The blow-by gases generally composed of air, carbon dioxide, steam, unburned hydrocarbons, and various pollutants present in the combustion chamber can degrade oil properties and condense/freeze in cold ambient conditions. Air is mainly present in the crankcase due to displacing the blow-by gases by a ventilation system. One of the most important duties of the ventilation system is to keep the pressure level in the crankcase as constant as possible and to prevent the buildup of the crankcase pressure. The latter is possible when the mixture of oil mist, blow-by gases, and air is accidentally ignited inside the crankcase, giving rise to internal explosion. Ignition can be promoted by any part of construction heated above 300–400 °C due to poor lubrication or by jets of hot combustion products escaping through piston rings or by electrostatic discharges. For minimizing the effect of explosion, marine engines must be equipped with crankcase relief valves [21–23] and flame arrestors [23,24].

Lubricating oil mist is characterized by a wide spectrum of droplet sizes (from 10^{-4} to 10^{-7} m [25–28]). Recently, Wang et al. [29] injected lubricant droplets of 1 mm size into the combustion chamber of a marine engine and found that these droplets disintegrated into smaller droplets with a size exceeding 40 µm. Dyson et al. [30] measured the size distribution of droplets in the crankcase of a fired engine using a laser diffraction particle sizer and reported three characteristic ranges of droplet size: spray-sized (250–1000 µm); major mist (30–250 µm); and minor mist (0.1–30 µm).

The ignition and combustion of fuel mist is a complex physicochemical phenomenon [31–33]. Contrary to the ignition and combustion of an isolated fuel droplet in an unconfined oxidizing atmosphere [34–36], the processes of ignition and combustion of fuel mist depend on the local instantaneous droplet size, liquid vapor concentration, and distance between droplets [37]. Both experimental and computational studies indicate that evaporating fuel droplets affect differently fuel-lean and fuel-rich laminar flames: they enhance the flames under fuel-lean conditions and suppress the flames under fuel-rich conditions [38]. The presence of evaporating fuel droplets in the overall fuel-lean mist enhances the growth of the flame surface area and the heat is released mostly due to the premixed mode of combustion [39]. As the overall equivalence ratio increases, the contribution of nonpremixed combustion to the overall heat release rate increases. Due to droplet inertia, the flame velocity in fuel mist can oscillate by alternating slow and fast modes of flame propagation [40]. There exist three main modes of quasi-steady mist/spray combustion, namely external combustion, when the flame stands at the periphery of the mist/spray cloud; group combustion, when small pockets of droplets burn inside the mist/spray cloud; and hybrid combustion, combining the two previous modes [41]. There are also transient modes of fuel mist combustion with spontaneous local extinction and re-ignition of flames around individual fuel droplets and droplet groups [42–44]. The flame propagation mechanisms are mainly determined by the heat conductivity and convection in the gas phase, but thermal radiation can also play a significant role [45–47]. Similarly to in gaseous flames, turbulence can exert a significant effect on flame ignition and propagation in fuel mist [48–54].

As mentioned above, marine engines are equipped with explosion relief systems composed of crankcase relief valves, flame arrestors, and sometimes venting ducts to avoid severe internal and external explosions of oil mist. To design explosion relief systems, there is a need in experimental and computational studies of mist flame ignition and propagation in the vented enclosures of complex geometry including linked vessels. In the literature, there are many references dealing with vented gaseous explosions and explosions in linked vessels (see, e.g., [55–59] and references therein). Despite explosion venting proving generally to be a good safety solution [60,61], vented explosions may exhibit a destructive power much higher than unvented explosions or explosions in a single vessel in terms of both peak pressures and rates of pressure rise [62–64]. Maremonti et al. [65] numerically simulated explosions of a methane–air mixture in two linked vessels

and found that the major factor affecting the explosion intensity was the diameter of the connecting duct, which influenced the turbulence induced in the second vessel. Ferrara et al. [66] experimentally studied the interaction between internal and external explosions of propane–air and methane–air mixtures in a vessel vented through a relief duct and found that the pressure rise inside the vessel closely resembled the pressure rise in the duct (“coherent deflagration” [67]). Di Sarli et al. [68] reported the results of numerical simulations of unsteady premixed methane–air flames in a vented vessel with a centered single obstacle of various shape and blockage ratio and attributed the arising overpressure peaks to the competition between the rates of combustion and venting. Recently, Wang et al. [69] reported the results of numerical calculations of internal explosions of methane–air mixture in a single vessel, in a single vessel with a connected duct, and in one of two vessels connected with a duct and found that the explosion intensity was highly affected by the ignition position in the linked system. In the literature, there are several studies comparing the results of CFD calculations with experimental data on medium- and large-scale vented gaseous explosions [70–73] and dust explosions [74–76]. In most cases, explosion vessels of simple geometries were considered. In some cases, the explosion vessels were equipped with regular [70] internal obstacles. To the best of the present authors’ knowledge, there has been no attempt so far to use CFD for simulating internal explosions in the crankcase of a real-size marine diesel.

The objective of this work was to numerically simulate various possible scenarios of unvented and vented internal explosions of oil mist in a crankcase of a real-size marine diesel engine in order to reveal the potentially most severe scenario and estimate the characteristic values of the most important parameters like the maximum rate of pressure rise, rate of heat release, turbulent burning velocity, etc. To follow this objective, the flame tracking (FT) model [77], which traces explicitly the flame in the oil mist–air mixture at the subgrid level, was used for the first time. This objective, approach, and the obtained results are the novel and distinctive features of the present study.

2. Materials and Methods

2.1. Marine Diesel Crankcase

As an example, a typical low-speed two-stroke six-cylinder marine diesel engine was considered (see, e.g., [78,79]). Figure 1 shows the schematic of the crankcase segment of a typical low-speed two-stroke marine diesel engine (Figure 1a) and the simulated crankcase of a 6-cylinder engine (Figure 1b). The adopted overall dimensions of the engine crankcase are a height of 5 m, width of 3 m, and length of 3 m. Three important specific features of crankcase construction are worth mentioning. Firstly, each crankcase segment contains many protruding and moving internal structural elements with blunt body shapes like ladders, cylinder liners, crossheads, crankshafts, etc., which can play the role of turbulence generators in a flow. Secondly, the A-frames separating the segments of individual cylinders possess crankshaft holes with windows connecting the neighboring crankshaft segments. Thirdly, each crankcase segment contains a pressure relief valve for explosion venting. All these features taken together mean that the engine crankcase as a whole must be considered as a system of linked vented vessels of large volume with multiple turbulence-generating internal obstacles. In view of the complexity of the internal geometry of the crankcase, it is worthwhile to simplify it by considering only the largest structural elements of the simplified shape. Moreover, in view of the fast development of internal explosion, it is reasonable to treat all moving elements as unmovable, at least at the first stages of the study.

The crankcase volume was assumed to be filled with air, oil vapor, and the mist of oil droplets of different sizes. Several possible accidental ignition sites are shown in Figure 1a as the numbered red dots. It was assumed that ignition sites #1 to #4 may appear due to the leakage of hot blow-by gases through the faulty stuffing box (#1), due to a hot spot on the crankpin bearing (#2), due to electrostatic discharge in the open space at the A-frame (#3), and due to a hot spot at the main bearing (#4). It is assumed that these ignition sites can occur in any of the 6 crankcase segments numbered by roman numerals I to VI in Figure 1b.

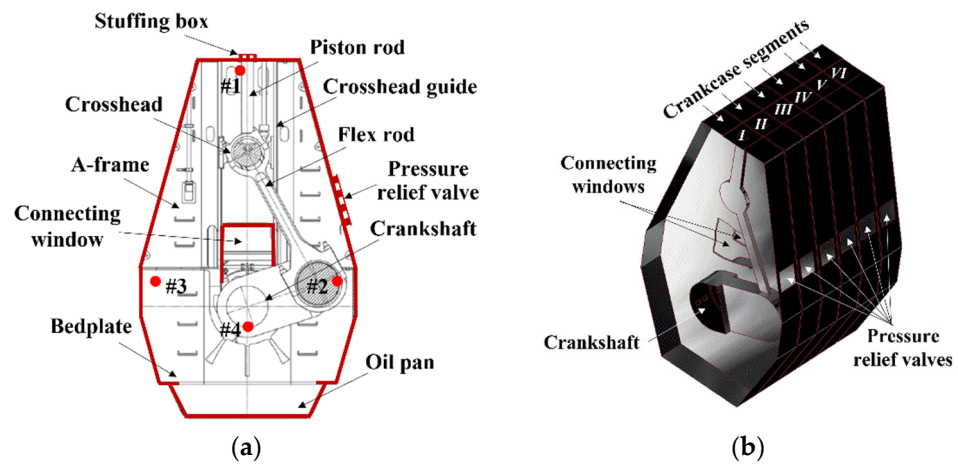


Figure 1. (a) Schematic of the crankcase segment of a marine diesel engine; red points show possible accidental ignition sites. (b) Simulated crankcase of a 6-cylinder engine with internal windows connecting crankcase segments; Roman numerals show the crankcase segment number (engine cylinder).

Following Distaso et al. [80–82], who showed that the chemical and physical characteristics of engine lubricating oils are well reproduced by a single primary hydrocarbon—*n*-hexadecane ($n\text{-C}_{16}\text{H}_{34}$), this hydrocarbon was used as the physical and chemical surrogate of the lubricating oil.

2.2. Mathematical Statement of the Problem

The evolution of crankcase explosion was simulated based on the three-dimensional (3D) unsteady Reynolds-averaged Navier–Stokes (URANS) equations for the two-phase turbulent reacting flow. The URANS equations were supplemented by the ideal gas equations of state, the $k\text{-}\epsilon$ turbulence model, and the FT combustion model [77], as well as by the initial and boundary conditions. The thermophysical parameters of gas and oil were considered variable. Since the characteristic time of explosion was small (less than a fraction of a second), the effects of gravity were neglected.

2.2.1. Gas Flow Equations

The governing equations for the gas phase are [83]:

$$\rho \frac{DU_i}{Dt} = \rho \frac{\partial U_i}{\partial t} + \rho U_j \frac{\partial U_i}{\partial x_j} = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j} [\tau_{ij} - \rho \overline{U'_i U'_j}] + S_m \quad (1)$$

$$\rho \frac{DI}{Dt} = \rho \frac{\partial I}{\partial t} + \rho U_j \frac{\partial I}{\partial x_j} = \dot{Q} + \frac{\partial P}{\partial t} + \frac{\partial}{\partial x_i} (\tau_{ij} U_j) + \frac{\partial}{\partial x_j} \left(\lambda \frac{\partial T}{\partial x_j} \right) + S_r + S_e \quad (2)$$

$$\rho \frac{DY_f}{Dt} = \rho \frac{\partial Y_f}{\partial t} + \rho U_j \frac{\partial Y_f}{\partial x_j} = \dot{r} + \frac{\partial}{\partial x_j} \left(\rho D_f \frac{\partial Y_f}{\partial x_j} - \rho \overline{Y'_f U'_j} \right) + S_Y \quad (3)$$

$$\rho \frac{\partial k}{\partial t} + \rho U_j \frac{\partial k}{\partial x_j} = P_k - \epsilon + \frac{\partial}{\partial x_j} \left(\mu + \frac{\mu_t}{\sigma_k} \frac{\partial k}{\partial x_j} \right) \quad (4)$$

$$\rho \frac{D\epsilon}{Dt} = \left(C_{\epsilon 1} P_k + C_{\epsilon 4} k \frac{\partial U_k}{\partial x_k} - C_{\epsilon 2} \epsilon \right) \frac{\epsilon}{k} + \frac{\partial}{\partial x_j} \left(\frac{\mu_t}{\sigma_\epsilon} \frac{\partial \epsilon}{\partial x_j} \right) \quad (5)$$

$$P = \rho RT \sum_{l=1}^N \frac{Y_l}{W_l} \quad (6)$$

$$H_l = H_l^0 + \int_{T^0}^T c_{p,l} dT \quad (7)$$

where t is time, x_j ($j = 1, 2, 3$) is the coordinate, U_i is the i th component of the mean velocity vector; ρ is the mean density; P is the mean pressure; τ_{ij} is the tensor of viscous stresses; U'_i is the i th component of the pulsating velocity vector; $I = H + 0.5\sum_i U_i^2$ is the mean total enthalpy (H is the mean static enthalpy); λ is the thermal conductivity; T is the mean static temperature; Y_f is the mean mass fraction of fuel; Y'_f is the pulsation of the fuel mass fraction; D_f is the molecular diffusion coefficient of fuel in the mixture; $\dot{r}$ and $\dot{Q}$ are the mean sources of fuel mass and energy due to frontal combustion; $k = \overline{U'_i U'_i}/2$ is the kinetic energy of turbulence; ε is the dissipation rate of k ; $P_k = -\overline{U'_i U'_j} \frac{\partial U_i}{\partial x_j}$ is the mean-strain production term; μ is the dynamic molecular viscosity; $\mu_t = C_\mu \rho k^2 / \varepsilon$ is the dynamic turbulent viscosity; $\sigma_k, \sigma_\varepsilon, C_\mu, C_{\varepsilon 1}, C_{\varepsilon 2}$, and $C_{\varepsilon 4}$ are the coefficients in the standard k - ε model; T^0 is the standard temperature (298 K); H_l and H_l^0 are the mean static enthalpy and standard enthalpy of formation of the l th species in the mixture ($l = 1, \dots, N$, N is the total number of species), respectively; R is the universal gas constant; W_l is the molecular mass of the l th species in the mixture; and Y_l is the mean mass fraction of the l th species in the mixture. Equation (3) implies that the mass fractions of species other than fuel were calculated based on the balances of C, H, and O elements. The source terms S_m , S_e and S_Y describe the interphase interaction of gas and mist droplets, whereas S_r represents gas radiation [84]:

$$S_r = 4\sigma(T - T_\infty)^4 \sum_i P_i a_i \quad (8)$$

Here, σ is the Stefan Boltzmann constant, T_∞ is the ambient temperature, P_i is the partial pressure of the i th radiating species (H_2O and CO_2), and a_i is the corresponding Plank absorption coefficient given by the polynomial:

$$a_i = c_0 + c_1\eta + c_2\eta^2 + c_3\eta^3 + c_4\eta^4 + c_5\eta^5 \quad (9)$$

where $\eta = 1000/T$. The coefficients of the polynomial are presented in Table 1.

Table 1. Coefficients in the Plank absorption polynomial (9).

Species	c_0	c_1	c_2	c_3	c_4	c_5
H_2O	−0.23093	−1.1239	9.4153	−2.9988	0.51382	$−1.8684 \times 10^{-5}$
CO_2	18.741	−121.31	273.5	−194.05	56.31	−5.8169

2.2.2. Source Terms for Interphase Exchange Processes

The mist droplets are represented by parcels each containing M identical droplets of diameter d . Each parcel moves as a single droplet according to equations:

$$\frac{dx_{kd}}{dt} = u_{kd} \quad (10)$$

$$m_d \frac{du_{kd}}{dt} = F_k \quad (11)$$

where index d denotes droplet properties, x_{kd} ($k = 1, 2, 3$) is the droplet Cartesian coordinate, u_{kd} is the k th component of the parcel velocity, m_d is the single droplet mass, and $F_k = F_{kf} + F_{kp}$ is the k th components of the force acting on a single droplet, which is composed of the aerodynamic drag F_{kf} and pressure F_{kp} forces. The forces are calculated using the following relationships:

$$F_{kf} = \frac{1}{2} \rho A_d C_d U_k |U| \quad (12)$$

$$F_{kp} = V_d \nabla P_k \quad (13)$$

where C_d is the aerodynamic drag coefficient; A_d is the droplet midsection area, V_d is the droplet volume, ∇P_k is the k th component of the pressure gradient vector, and U_k is the relative velocity of phases defined as

$$U_k = \overline{u}_k + u'_k - u_{kd} \quad (14)$$

The drag coefficient C_d is determined based on the value of the relative Reynolds number $Re = \frac{\rho|U|d}{\mu}$ based on the length of the relative velocity vector, $|U|$:

$$C_d = \begin{cases} \frac{24}{Re_d} \left(1 + 0.15 Re^{0.687} \right), & Re < 10^3 \\ 0.44, & Re \geq 10^3 \end{cases} \quad (15)$$

It is assumed that during motion the droplets undergo aerodynamic breakup once their Weber number $We = d\rho U^2 / \sigma_d$ exceeds the critical value: $We > We_c \approx 12$. Here σ_d is the liquid surface tension. The breakup phenomenon is described by the WAVE model [85], claiming that during the characteristic breakup time τ_b the droplet radius $r = d/2$ continuously decreases to a certain stable value r_s :

$$\frac{dr}{dt} = -\frac{r - r_s}{\tau_b} \quad (16)$$

The stable radius of the droplet is determined as

$$r_s = \begin{cases} b_0 \Lambda & \text{at } b_0 \Lambda < r \\ \min \left[\left(\frac{3\pi r^2 |U|}{2\Omega} \right)^{1/3}, \left(\frac{3r^2 \Lambda}{4} \right)^{1/3} \right] & \text{at } b_0 \Lambda > r \end{cases} \quad (17)$$

where Λ is the perturbation wavelength, Ω is the rate of perturbation growth, and $b_0 = 0.61$ is the model constant. The characteristic time τ_b is given by the relationship:

$$\tau_b = b_1 \frac{r}{|U|} \sqrt{\frac{\rho_d}{\rho}} \quad (18)$$

where b_1 is the model parameter varying from 10 to 30.

For determining the source terms in the mass and energy conservation equations for the gas phase, the Dukowicz model [86] of droplet heating and evaporation was used. The mass and energy conservation equations for a single droplet in parcel read:

$$\frac{dm_d}{dt} = \dot{Q} \frac{\dot{f}_{vs}}{\dot{q}_s} \quad (19)$$

$$m_d c_d \frac{dT_d}{dt} = \dot{Q} + L \frac{dm_d}{dt} \quad (20)$$

where $\dot{Q}$ is the heat flux to the droplet, c_d is the liquid specific heat, L is the latent heat of vaporization, $\dot{f}_{vs}$ and $\dot{q}_s$ are the specific (per unit surface of the droplet) heat and mass fluxes, and indices s and v denote surface and vapor, respectively.

The unknown fluxes were determined using the following relationships:

$$\dot{Q} = \pi \lambda d u (T - T_d) + q_r \quad (21)$$

$$\frac{\dot{f}_{vs}}{\dot{q}_s} = -\frac{B_Y}{h - h_s - (h_{vs} - h_s)(Y_v - Y_{vs})} \quad (22)$$

$$B_Y = \frac{Y_{vs} - Y_v}{1 - Y_{vs}} \quad (23)$$

$$Nu = 2 + Re^{1/2} Pr^{1/3} \quad (24)$$

$$q_r = q(r_F) \frac{d^2}{16r_F^2} \quad (25)$$

where Nu and Pr are the Nusselt and Prandtl numbers, q_r is the rate of radiation absorption by the droplet, and r_F is the distance from the droplet to the flame front. All thermo-physical parameters of gaseous species on the droplet surface were determined based on the assumption of phase equilibrium. Model [86] shows satisfactory agreement with the results of detailed calculations of droplet heating and evaporation [87], as well as with experiments [88].

To couple mass, momentum, and energy variations in the liquid phase with those in the gas phase, the force F_i , as well as mass $\dot{Q}_{\frac{f_{gs}}{q_s}}$ and heat $\dot{Q}$ fluxes calculated for a single droplet in the parcel, were then taken with the opposite sign, multiplied by M , and entered as source terms S_m , S_e , and S_Y in the momentum and energy conservation equations and species continuity equations for the gas phase. The same procedure was used for other parcels available in the given location at a given time.

2.2.3. Combustion Model

The stages of flame ignition, propagation, and acceleration in the oil mist were simulated using the FT model based on the Huygens principle stating that each point on the flame front is itself the source of spherical wavelets, whereas the secondary wavelets emanating from different points mutually interfere [89]. Within the FT model, the flame is an infinitely thin surface separating the fresh mixture from combustion products and is traced explicitly at the subgrid level. The flame surface is represented by a set of flame elements. Each element moves in the flow at a velocity equal to the sum of the local turbulent flame velocity, u_T , and the local flow velocity, U , determined from Equations (1)–(7). The local turbulent flame velocity is related to the local laminar flame velocity, u_n , through the mass balance equation $F_n u_n = F u_t$ (here, F_n and F are the local instantaneous specific (per unit volume) surface areas of a true wrinkled and averaged planar flame, respectively) and depends on u_n and local turbulence parameters like, e.g., in the Shchelkin model [90]:

$$u_T = A u_n \sqrt{1 + (U'/u_n)^2} \quad (26)$$

where $U' = \sqrt{\frac{2}{3}k}$ is the pulsating flow velocity and $A \approx 1$ is the model parameter. In the FT model, the average planar flame surface is explicitly traced and the specific surface area F is determined in the course of solution. Therefore,

$$\dot{r} = \rho_+ Y_{f+} F_n u_n = \rho_+ Y_{f+} F u_t \quad (27)$$

$$\dot{Q} = Q \dot{r}_s \quad (28)$$

where ρ_+ and Y_{f+} are the local instantaneous density of the fresh mixture and fuel mass fraction immediately ahead of the flame surface, respectively; and Q is the heat effect of combustion. For u_n , the precalculated look-up tables were used that include the dependences $u_n(T, P, \Phi)$, where Φ is the fuel-to-air equivalence ratio in the gas phase. Such look-up tables are developed for mixtures of different alkane, alkene, and alkyne hydrocarbons with air and oxygen based on either detailed or overall reaction mechanisms [91]. Figure 2 demonstrates the accuracy of the look-up tables in predicting the laminar flame velocity in the homogeneous air mixtures of n-hexadecane, which was chosen as a physical and chemical surrogate of lubrication oil, against experimental data [92] at 443 K and 1 bar. It is worth noting that the FT model was previously successfully used in solving problems of flame acceleration and deflagration-to-detonation transition in gaseous and two-phase explosive mixtures [93].

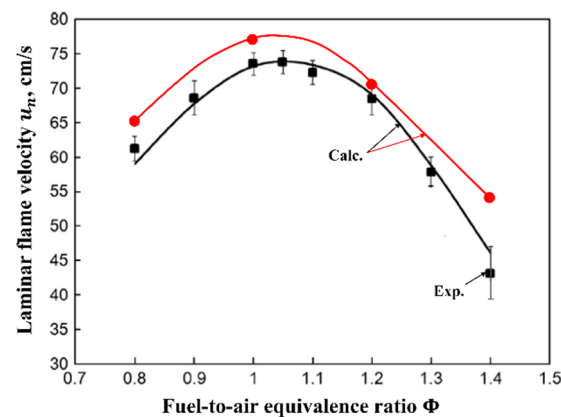


Figure 2. The dependences of the laminar flame speed in n-hexadecane/air mixtures on the fuel-to-air equivalence ratio at 443 K and 1 bar. Curves correspond to calculations by present authors (red curve) and authors of [90] (black curve); black squares correspond to measurements [90].

2.2.4. Initial and Boundary Conditions

To study the dynamics of internal explosions in the crankcase, it is worthwhile to consider the conditions most favorable for explosions. In view of this, it was assumed that initially, the crankcase is uniformly (in average) filled with the quiescent two-phase mixture of air, oil vapor, and oil droplets of the overall fuel-to-air equivalence ratio $\Phi = 1$ at atmospheric pressure ($P = 1$ bar) and elevated temperature of $T_0 = 380$ K [94]. The degree of oil pre-evaporation was taken to be equal to 60%, 70% (baseline case), and 80%. As for the initial droplet size distribution, two cases were considered: one with the monodisperse droplets with the initial diameter $d_0 = 60$ μm (baseline cases) [29] and another with the polydisperse trimodal droplet size distribution with $d_0 = 10$ μm (10 wt%), 250 μm (10 wt%), and 500 μm (80 wt%) [30].

The boundary conditions for the mean flow velocity, pressure, temperature, mean species mass fractions, and turbulent kinetic energy and its dissipation (see Equations (1)–(5)) on the crankcase rigid walls were set using the formalism of wall functions under the assumption that the walls are nonslip, noncatalytic, impermeable, and isothermal (380 K). Each pressure relief valve was assumed to be closed if the internal overpressure at the inner valve surface, ΔP_v , was less than the critical overpressure of valve opening ($\Delta P_c = 0.005$ or 0.05 bar). It was assumed that the pressure relief valves open instantaneously when $\Delta P_v > \Delta P_c$. Since the values of ΔP_v at different pressure relief valves can be different at a given time, the opening time of each valve can be also different. When considering only internal crankcase explosions, a simple constant-pressure ($P = 1$ bar) boundary condition was set at the open pressure relief valve. Table 2 shows the data for all calculation variants presented in this paper.

2.3. Numerical Procedure

The governing equations for the mean flow variables coupled with the turbulence and combustion models are presented in the integral form of the general conservation law and are solved in Cartesian coordinates using collocated variable arrangements using the cell-face based connectivity and interpolation practices for gradients. The rate of variable change is discretized by the first-order Euler scheme. The convective fluxes were treated using a deferred correction approach with the blending factor between UPWIND and MINMOD schemes. The diffusion terms were discretized using the approach of [95] avoiding unphysical oscillations. The arising set of linear algebraic equations was solved using the iterative procedure based on the SIMPLE algorithm [96]. The numerical algorithm was realized in the in-house gas dynamic code coupled with the open-access solvers of linear algebraic equations.

Table 2. Calculation variants.

Variant	Drop Size, μm	Φ	Vap, %	T_0 , K	P , Bar	ΔP_{cr} , Bar	Ignition Site	Crankcase Segment
1	60	1	60	380	1	–	4	IV
2	60	1	60	380	1	–	1	I
3	60	1	60	380	1	–	2	I
4	60	1	60	380	1	–	3	I
5	60	1	60	380	1	–	4	I
6	60	1	60	380	1	–	1	IV
7	10–250–500	1	60	380	1	–	4	IV
8	60	1	70	380	1	–	4	IV
9	60	1	80	380	1	–	4	IV
10	60	1	60	380	1	0.05	4	IV
11	60	1	60	380	1	0.005	4	IV

The baseline computational mesh consists of 1,000,000 cells with the characteristic computational cell size of 4–5 mm. The integration time step was varied between 10^{-5} and 10^{-4} s to ensure the CFL number ≤ 3 . The effect of the dimensions of the computational cells on the flow structure in the crankcase was checked using additional calculations on the finer mesh containing 6,500,000 cells with a characteristic size of 2–3 mm. It must be noted that regardless of the spatial resolution of the computational mesh, the flame in the adopted combustion model is traced explicitly at the subgrid level.

The calculation begins from mixture ignition at a certain ignition site, which is a circle 1 mm in radius (see Figure 1a). The ignition procedure implies that the flame from the initial ignition site expands spherically with an apparent velocity $U_{\text{ign}} = \alpha u_t$, where α is the expansion coefficient of combustion products. When the size of the ignition source reaches 5 mm, the main combustion calculation module based on the FT model is switched on.

2.4. Validation Test Cases

2.4.1. Validation Test Case I

The first validation test case is based on experimental work [97]. In [97], experimental results on the combustion of homogeneous propane–air mixtures of different composition in a vertical cylindrical vessel at normal pressure and temperature (NPT) and normal gravity conditions were reported. The vessel was 172 mm in inner diameter and 360 mm high. It was equipped with an igniter at the bottom or in the center.

In calculations, a uniform structured mesh with a cell size of 2 mm was used. This cell size is close to that used in the calculations of internal explosions in the crankcase enclosure of a considerably larger size. The initial flame kernel is a circle 1 mm in radius with a center located in the middle of the cylinder bottom. Figure 3a shows the snapshot of the calculated flame shape and position at the combustion of fuel-rich propane–air mixture with an equivalence ratio of $\Phi = 1.13$. The flame front is elongated in vertical direction and touches the wall. Figure 3b compares the predicted and measured pressure histories in the vessel for two fuel-rich propane–air mixtures. Despite combustion in calculations being seen to be somewhat more intense than in the experiments, the agreement between the predicted and measured results can be treated as satisfactory in terms of the characteristic explosion time and pressure [77].

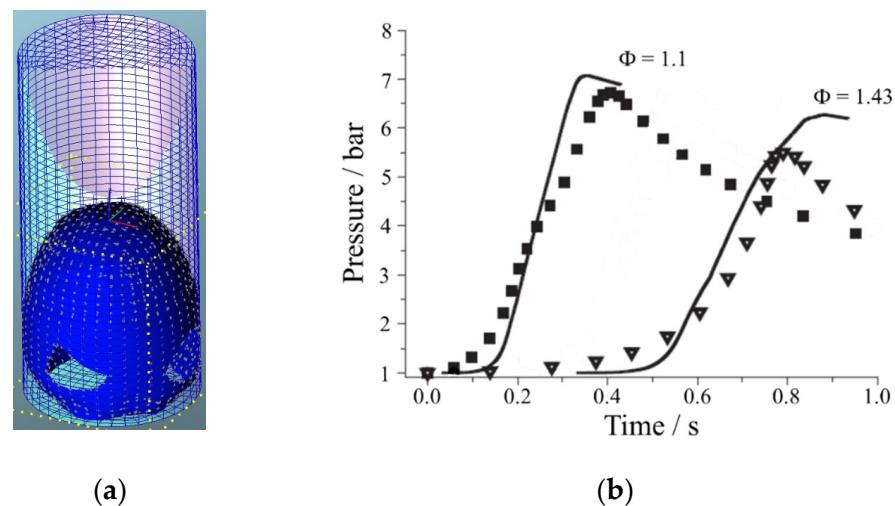


Figure 3. (a) The snapshot of the calculated flame shape and position at combustion of fuel-rich propane–air mixture with an equivalence ratio of $\Phi = 1.13$ at time 65 ms after ignition; and (b) comparison of predicted (curves) and measured (symbols [97]) pressure histories in the cylindrical vessel for two fuel-rich propane–air mixtures with $\Phi = 1.13$ and 1.43.

2.4.2. Validation Test Case II

The second validation test case is based on experimental work [98]. In [98], flame ignition and propagation in the initially quiescent stoichiometric propane–air mixture was studied in a straight tube of circular cross-section with one closed and another open end under NPT conditions. The tube diameter and length were 152 mm and 3.1 m, respectively. The tube was equipped with regular identical orifice plates with blockage ratios of 0.43, 0.6, and 0.75 installed with a pitch equal to the tube diameter. The blockage ratio was defined as the ratio of orifice plate-to-tube cross-section area. A weak spark discharge was used to ignite the mixture.

In calculations, a uniform structured mesh with a cell size of 2 mm was used. Figure 4 compares the calculated and measured dependences of the apparent velocity of the flame leading edge on the travelled distance for the three values of the blockage ratio. The calculated flame velocity is seen to follow the experimental points very well until the flame velocity reaches the value of 300–400 m/s. At larger flame velocities, the calculated values overpredict the measured ones: the maximum calculated values reach 750–900 m/s, whereas the maximum measured values reach 650–700 m/s. At the initial stage, flame accelerates faster in a tube with orifice plates of a larger blockage ratio. This is due to a higher level of turbulent pulsations occurring near obstacles. However, as the flame speed increases, the loss of momentum on the obstacles increases, and the acceleration of the flame at the obstacles with higher blockage ratio slows down. These trends are the same in both calculations and measurements. Oscillations in the flame velocity are associated with flow acceleration in the cross-section contraction in orifice plates (flame velocity increases) and flow deceleration in cross-section expansion between orifice plates (flame velocity decreases). Contrary to the calculations, the measured points show the gradual increase in the flame velocity, which is caused by averaging the flame velocity over the measuring segments of finite length.

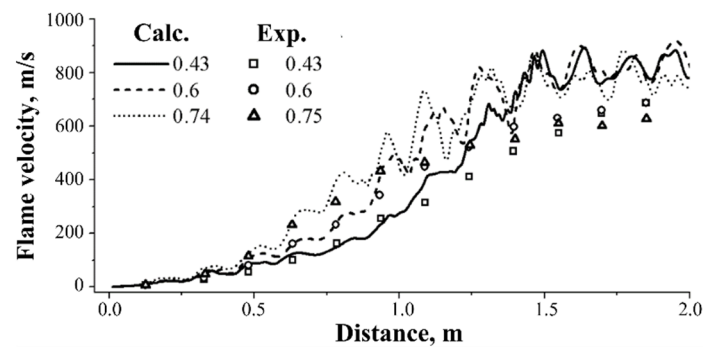


Figure 4. Comparison of calculated (curves) and measured (symbols [98]) dependences of the apparent flame velocity on the traveled distance.

2.4.3. Validation Test Case III

The third validation test case refers to experimental studies [99] on the propagation of laminar flame in the droplet suspension of partly pre-evaporated diesel oil and heavy fuel oil in air. Experiments were conducted in a setup with a free-falling combustion chamber of a square cross-section 70×70 mm. A suspension of partly pre-evaporated droplets was first supplied to the chamber. The chamber was separated into two parts by the flame-extinguishing permeable partition. The mixture was ignited in the central part of the chamber below the valve, so that the flame could move downwards, whereas the combustion products could expand upwards. The propagation of the flame front was recorded by the high-speed video camera at a length of 150 mm. The measured flame velocities in droplet suspensions of diesel oil and heavy fuel oil were very close to each other at similar experimental conditions.

In calculations, the full geometry of the combustion chamber with dimensions $200 \times 70 \times 70$ mm was used with the uniform structured computational mesh and a cell size of 2 mm. The oils applied in the experiments are represented by n-hexadecane. Figure 5 shows a schematic of the computational domain. At the left end of the chamber, the boundary conditions of a constant static pressure of 1 bar were set. All other chamber walls were adiabatic. To simulate the permeable flame-extinguishing partition, chemical reactions are deactivated in a chamber section 50 mm long from the left border (marked in orange in Figure 5). In this case, combustion products can flow freely through this region.

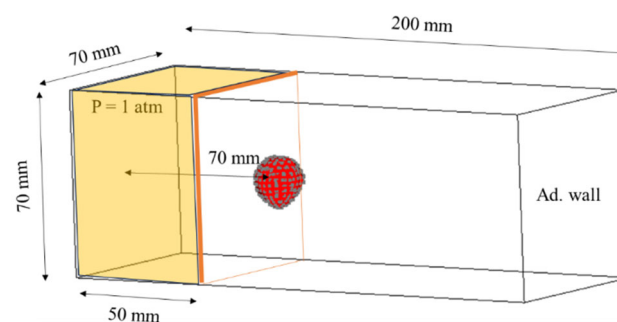


Figure 5. Schematic of the computational domain in the validation test case III. Red ball is the flame kernel. Chemical reactions are deactivated in a chamber section 50 mm long from the left border (marked in orange).

Initially, the entire region was uniformly filled with a quiescent mixture with a uniform random distribution of droplets in the computational domain. Fuel droplets, regardless of their initial size, were modeled using 100,000 parcels.

To check the sensitivity of the calculation results to changes in channel geometry and the resolution of the computational mesh, two additional calculations were made: one for a channel 400 mm long with a cell size of 2 mm and another for a channel 200 mm long

with a cell size of 1 mm. The results of additional calculations virtually repeated the data obtained in the calculations for the basic geometry. When modeling radiation, it is assumed that only half of the thermal radiation energy can be absorbed by droplets, while the other half of the thermal radiation energy escapes through the left boundary.

Figure 6 shows the calculated distributions of gas and droplet temperatures during combustion of the stoichiometric suspension of partly pre-evaporated (50%) n-hexadecane droplets (initial diameter 30 μm) in air at time instants from 0.1 to 0.4 s. It is seen that the flame is curved, the droplet number density is nonuniform across the flame, and droplets rapidly disappear in the flame front. The thermal radiation of H_2O and CO_2 is seen to contribute to droplet heating and evaporation ahead of the flame with the evident decay of the radiation heat flux with the distance from the flame. As a result, the flame propagates in the nonhomogeneous droplet suspension in terms of the droplet number density, size, and temperature. The flame propagation velocity was determined as the mass averaged value.

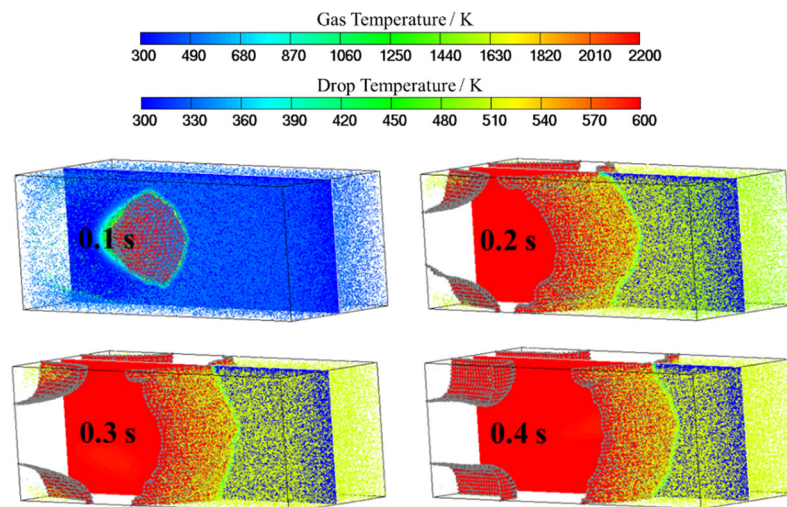


Figure 6. Calculated temperature distributions of gas (ground color) and droplets (the color of dots) during combustion of the stoichiometric suspension of partly pre-evaporated (50%) n-hexadecane droplets in air; initial droplet diameter is 30 μm .

Figure 7 compares the calculated (lines) and measured (symbols [99]) dependences of the mass averaged flame propagation velocity, u_f , on the initial droplet size and the degree of fuel pre-evaporation. Satisfactory qualitative and quantitative agreement between the predicted and measured data was obtained.

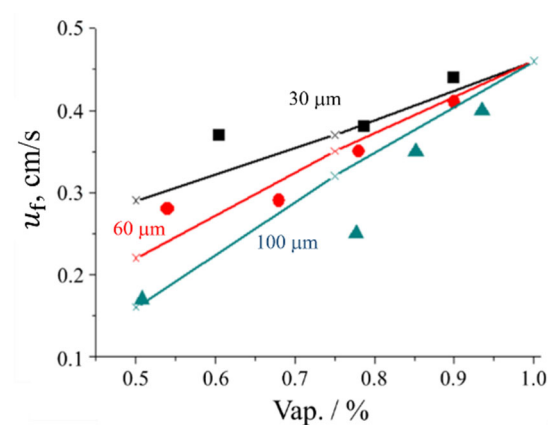


Figure 7. Comparison of calculated (lines) and measured (symbols [99]) dependences of the flame propagation velocity u_f on the initial droplet diameter and the degree of fuel pre-evaporation (Vap).

2.4.4. Conclusions on Validation Test Cases

The three validation cases aimed at checking the capability of the model to predict a constant-volume gaseous explosion (validation test case I), premixed flame acceleration in the volume with regular internal obstacles (validation test case II), and laminar flame propagation in a mixture of partly pre-evaporated lubricating oil mist with air (Validation test case III). Test case 1 aimed at checking the characteristic explosion time and pressure. Test case 2 aimed at checking flame acceleration at internal obstacles of different blockage ratio. Finally, test case 3 aimed at checking flame propagation in oil mist with droplets of different initial size and pre-evaporation degree. The important point is that all validation cases used computational meshes of the same quality as that applied below in the simulation of a large-scale crankcase explosion of oil mist. Since all validation cases passed the tests successfully, there are solid grounds to treat the obtained results as reliable.

3. Results and Discussion

Let us now consider the various possible scenarios of accidental internal explosions of oil mist in the crankcase of a marine diesel engine of Figure 1b and reveal the potentially most severe scenarios by placing the ignition site (positions #1 to #4 in Figure 1a) in different crankcase segments (segments I to VI in Figure 1b). We started simulations from checking the mesh sensitivity of the calculation results for unvented explosions (i.e., explosions with closed pressure relief valves). Then we show the results of calculations for unvented explosions and checked the effects of droplet size distribution in the oil mist and the pre-evaporation degree of the oil mist on the dynamics of pressure rise in the crankcase in the course of unvented explosion. Thereafter, we discuss the results of calculations for vented explosions with two different values of the critical overpressure of valve opening, namely, 0.05 and 0.005 bar. In view of explosion venting, all calculations presented below in this paper are limited by a relatively low explosion overpressure of $\Delta P = 0.1$ bar.

3.1. Unvented Explosion with Ignition in Site #4 in Crankcase Segment IV (Variant 1): Mesh Sensitivity Study

The mesh sensitivity of the calculation results was studied using one of the most severe scenarios (see below), when oil mist was presumably ignited by the hot spot on the main bearing in crankcase segment I. Figure 8 shows snapshots of gas temperature, oil mist droplet diameter, and flame surface in the crankcase at different time instants calculated at the baseline (left column) and fine (right column) meshes. From now on, the gas temperature in the figures is shown by colored fields in the longitudinal cross-section of the crankcase; the droplet size is shown by colored dots in the crankcase volume and superimposed on the temperature field; and the flame is shown by the brown-color surface. The corresponding scales are shown on the top of the figures. After ignition in crankcase segment IV, the flame is seen to spread mostly horizontally toward the neighboring crankcase segments, and in 0.4 s after ignition, it breaks into the outer segments I and VI. Such fast flame propagation is caused by turbulence generated by the windows in the A-frame. As seen, the flame surface becomes highly wrinkled and oil droplets are accumulated near the rigid walls. In calculations with both meshes, the explosion dynamics are evidently very similar.

Figure 9 shows the local flow structure in the vicinity to the flame surface at a time instant of 0.4 s after ignition. The oil droplets are seen to disappear (evaporate) immediately behind the flame surface.

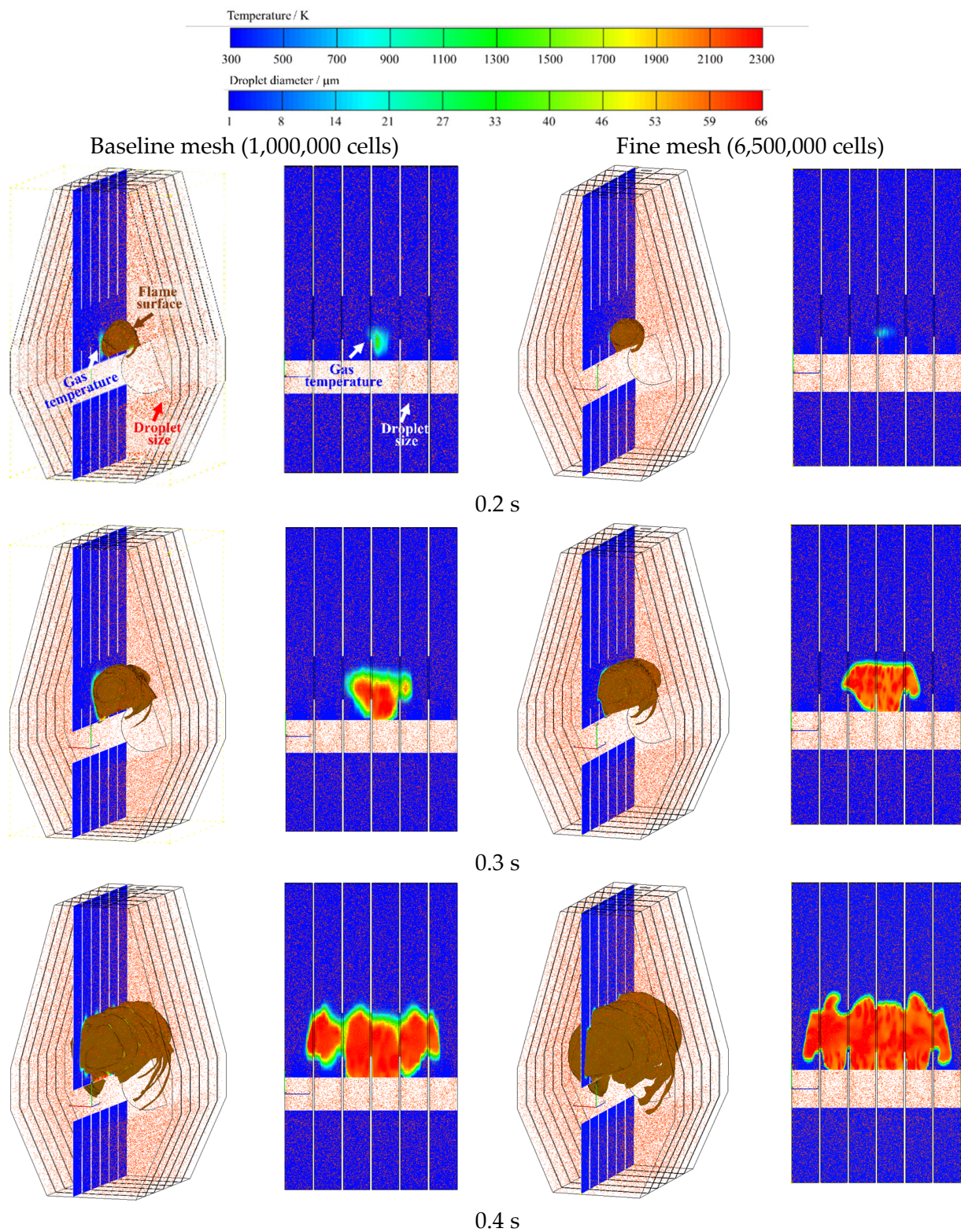


Figure 8. Snapshots of gas temperature (fields), droplet diameter (dots), and flame surface evolution in the course of unvented crankcase explosion calculated at the baseline (left column) and fine (right column) meshes. Initial oil droplet diameter $d_0 = 60 \mu\text{m}$; the degree of oil pre-evaporation is 70%; time is counted from the start of ignition.

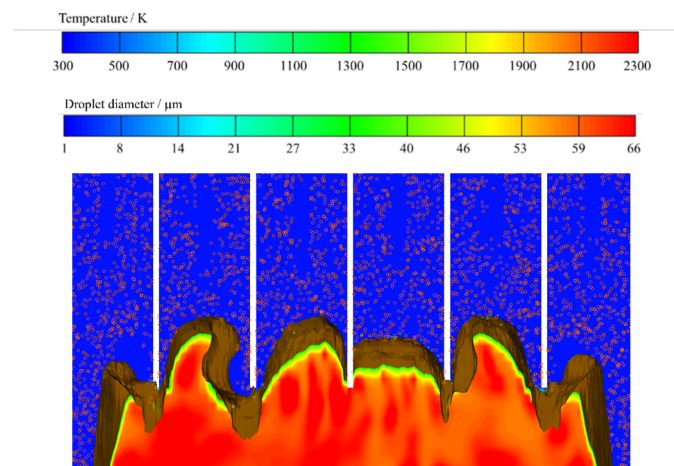


Figure 9. The exploded view of local gas temperature and droplet diameter distributions, as well as the flame surface at a time instant of 0.4 s after ignition. Initial oil droplet diameter $d_0 = 60 \mu\text{m}$; the degree of oil pre-evaporation is 70%.

Figure 10 compares the pressure histories in the crankcase calculated using the baseline and fine meshes. The limiting value of overpressure ($\Delta P = 0.1 \text{ bar}$) was attained in approximately 0.39 s (fine mesh) and 0.4 s (baseline mesh) after ignition, i.e., the difference is 2.5%. This means that the baseline mesh can be readily used for further calculations.

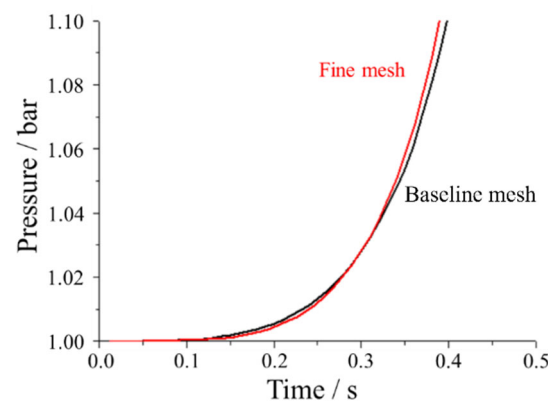


Figure 10. Comparison of pressure histories in the course of unvented crankcase explosion calculated using the baseline and fine meshes.

3.2. Unvented Crankcase Explosion

This section presents the results of some selected calculations for unvented crankcase explosions. Firstly, we show the differences in the explosion dynamics when ignition occurs in the same crankcase segment (segment I) but in different sites (##1, 2, 3, and 4 in Figure 1a). Secondly, we compare the explosion dynamics when the same ignition site (site #1) appears in different crankcase segments, e.g., in segment I and in segment IV.

3.2.1. Unvented Explosion in Crankcase Segment I: Ignition Site #1 (Variant 2)

Figure 11 shows snapshots of gas temperature, oil mist droplet diameter, and flame surface evolution in the course of unvented crankcase explosion presumably due to the leakage of hot blow-by gases through the faulty stuffing box (ignition site #1 in Figure 1a) in crankcase segment I. After ignition, the flame propagates predominantly downward toward the window in the A-frame. After reaching the window, it breaks through four adjacent crankcase segments in a very short time ($\sim 0.05 \text{ s}$) forming a highly turbulent flame core expanding mainly upward and in lateral directions.

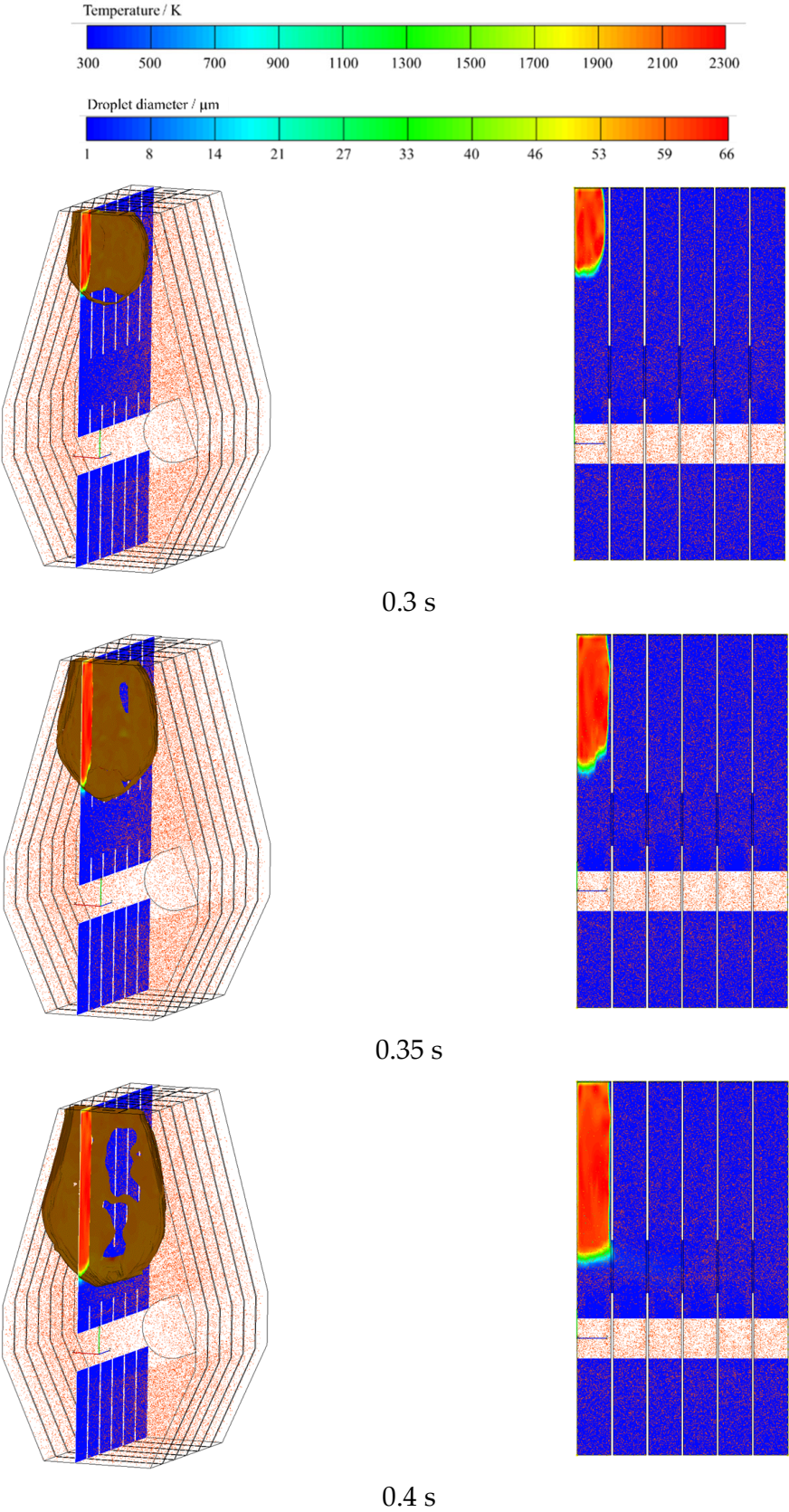


Figure 11. Cont.

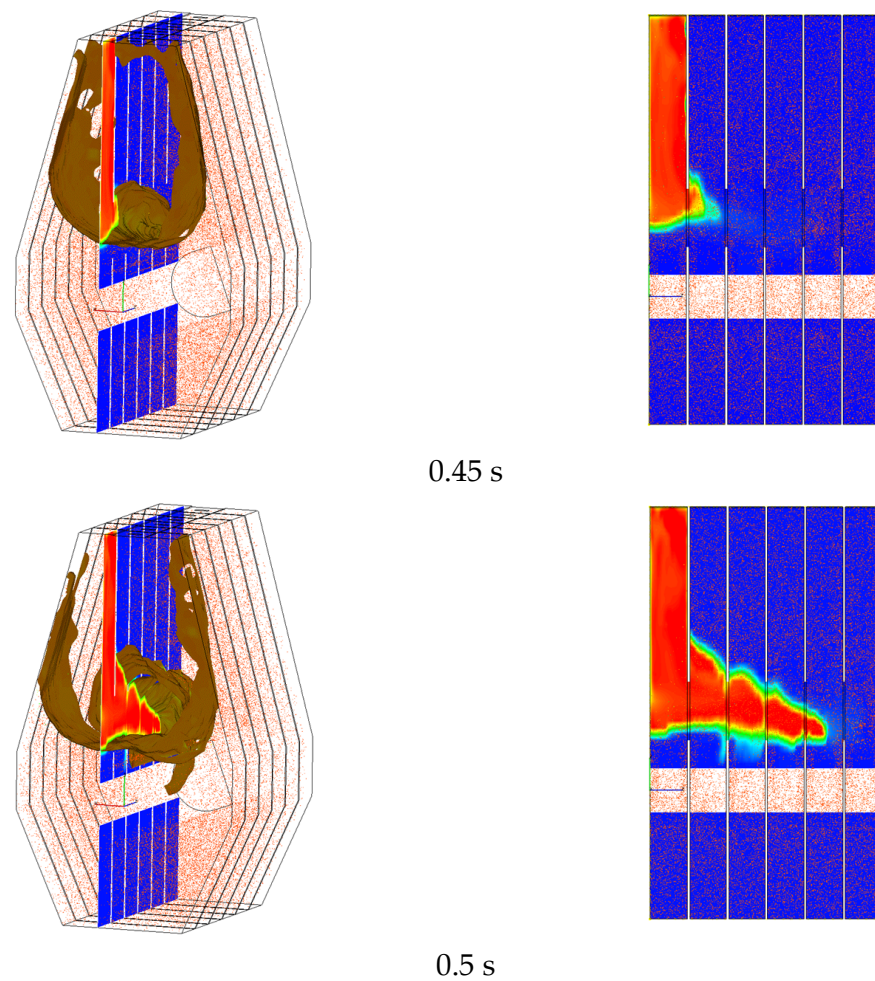


Figure 11. Calculated snapshots of gas temperature (fields), droplet diameter (dots), and flame surface evolution in the course of unvented crankcase explosion, ignition site #1 in crankcase segment I. Initial oil droplet diameter $d_0 = 60 \mu\text{m}$; the degree of oil pre-evaporation is 70%; time is counted from the start of ignition.

3.2.2. Unvented Explosion in Crankcase Segment I: Ignition Site #2 (Variant 3)

Figure 12 shows snapshots of gas temperature, oil mist droplet diameter, and flame surface evolution in the course of unvented crankcase explosion presumably triggered by a hot spot on the crankpin bearing (ignition site #2 in Figure 1a) in crankcase segment I. After ignition, the flame propagates in all directions increasingly stretching toward the window between crankcase segments. After reaching the window, it breaks through all five adjacent crankcase segments, forming a highly turbulent flame core expanding in all directions inside individual segments.

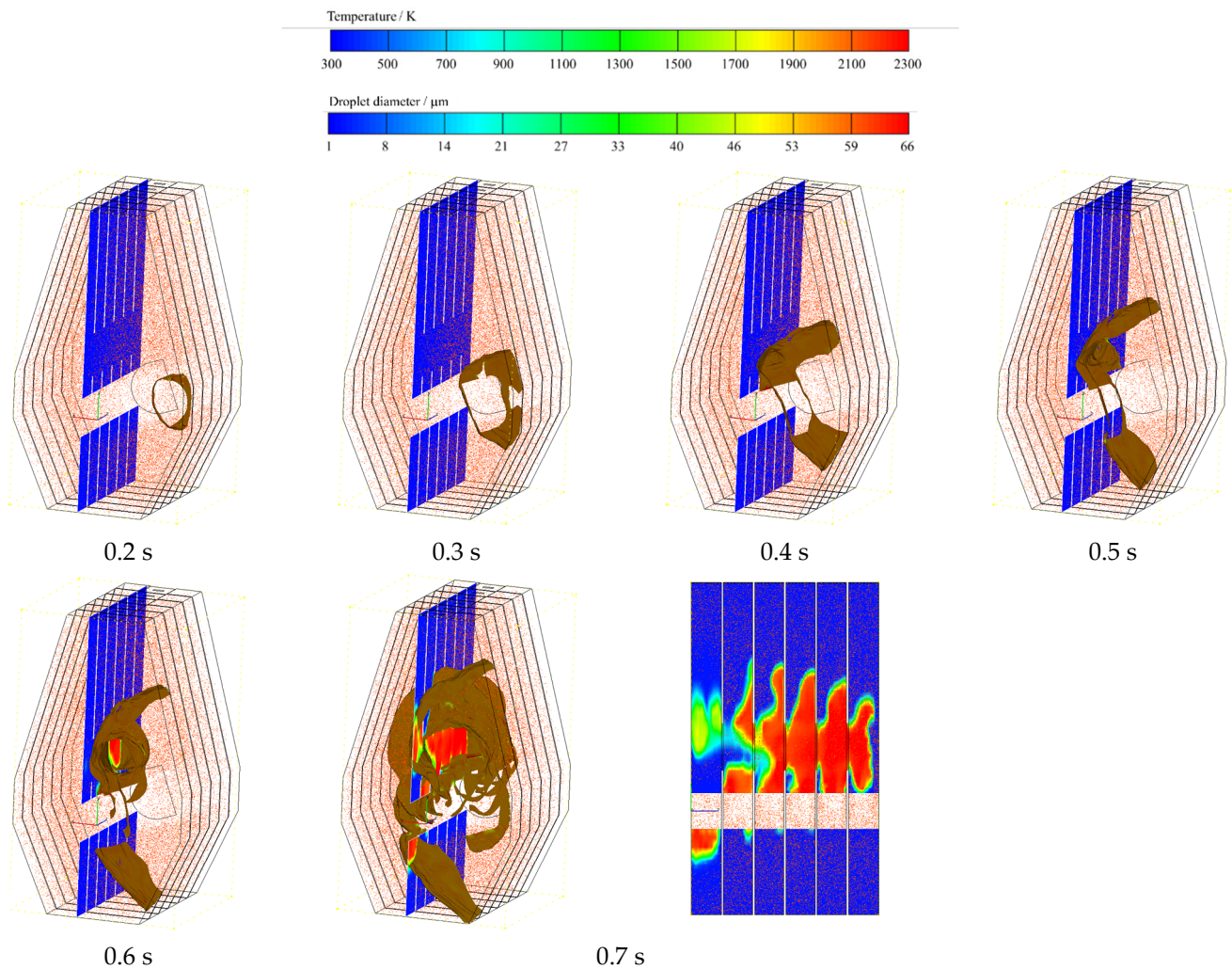


Figure 12. Calculated snapshots of gas temperature (fields), droplet diameter (dots), and flame surface evolution in the course of unvented crankcase explosion, ignition site #2 in crankcase segment I. Initial oil droplet diameter $d_0 = 60 \mu\text{m}$; the degree of oil pre-evaporation is 70%; time is counted from the start of ignition.

3.2.3. Unvented Explosion in Crankcase Segment I: Ignition Site #3 (Variant 4)

Figure 13 shows snapshots of gas temperature, oil mist droplet diameter, and flame surface evolution in the course of an unvented crankcase explosion presumably triggered by an electrostatic discharge in the open space at the A-frame (ignition site #3 in Figure 1a) in crankcase segment I. This test case is similar to that considered in the previous section, but ignition occurs in the open space rather than in the space blocked by the crankshaft. After ignition, the flame stretches toward the window. After reaching the window, it also breaks through all five adjacent crankcase segments and forms a highly turbulent flame core expanding in all directions inside individual segments. However, as compared to the previous test case, the burned volume is significantly larger 0.7 s after ignition.

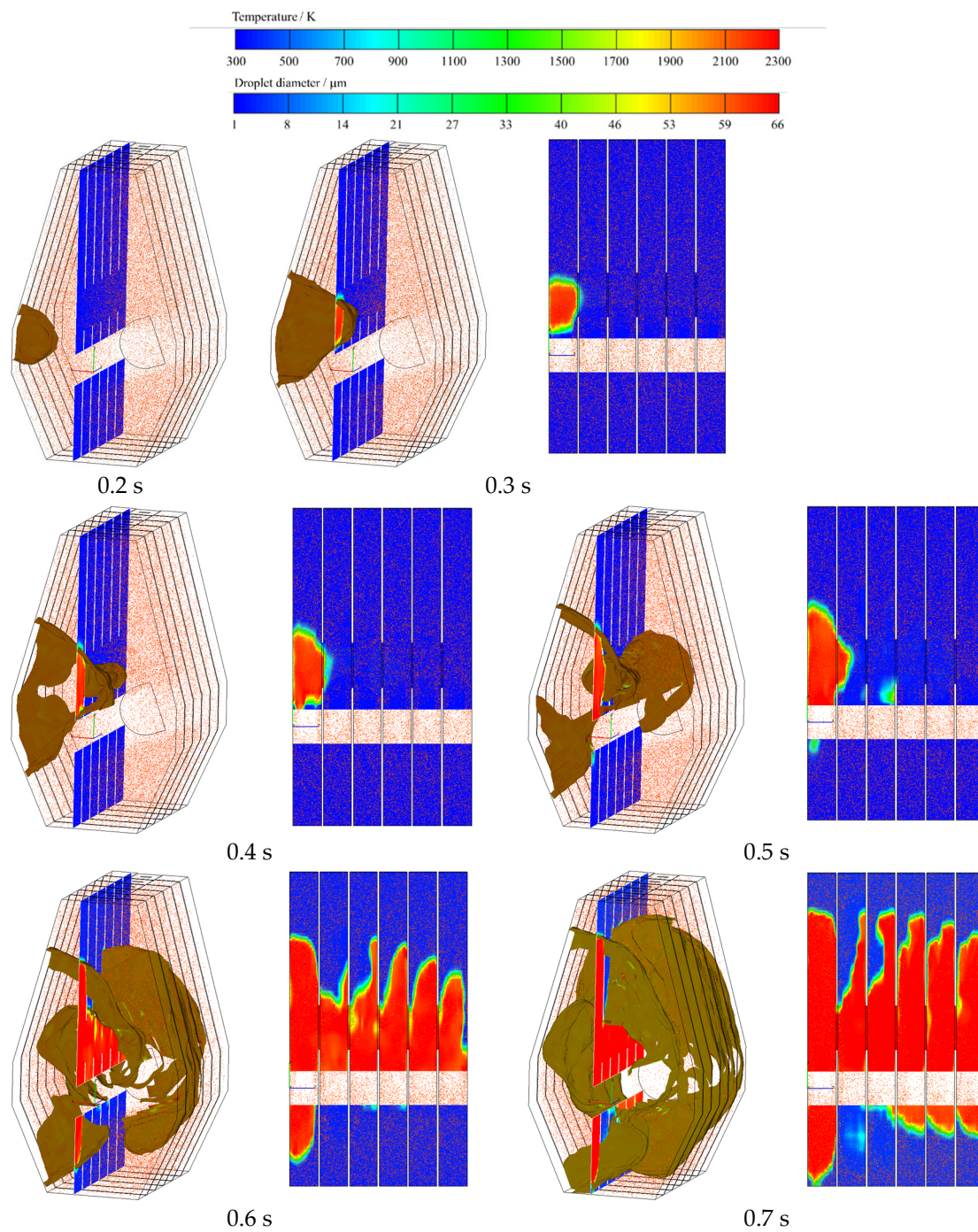


Figure 13. Calculated snapshots of gas temperature (fields), droplet diameter (dots), and flame surface evolution in the course of unvented crankcase explosion, ignition site #3 in crankcase segment I. Initial oil droplet diameter $d_0 = 60 \mu\text{m}$; the degree of oil pre-evaporation is 70%; time is counted from the start of ignition.

3.2.4. Unvented Explosion in Crankcase Segment I: Ignition Site #4 (Variant 5)

Figure 14 shows snapshots of gas temperature, oil mist droplet diameter, and flame surface evolution in the course of an unvented crankcase explosion presumably triggered by a hot spot on the main bearing (ignition site #4 in Figure 1a) in crankcase segment I. This test case can be compared with the test case considered in Section 3.1 where the same ignition site was applied to crankcase segment IV. Contrary to ignition in segment IV, where the flame propagated horizontally into the two neighboring crankcase segments,

in the current test case, the flame spread into the single neighboring segment II and then successively into the segments III to VI. This sequential nature of flame propagation from segment to segment slows down the overall process of explosion evolution compared to that discussed in Section 3.1.

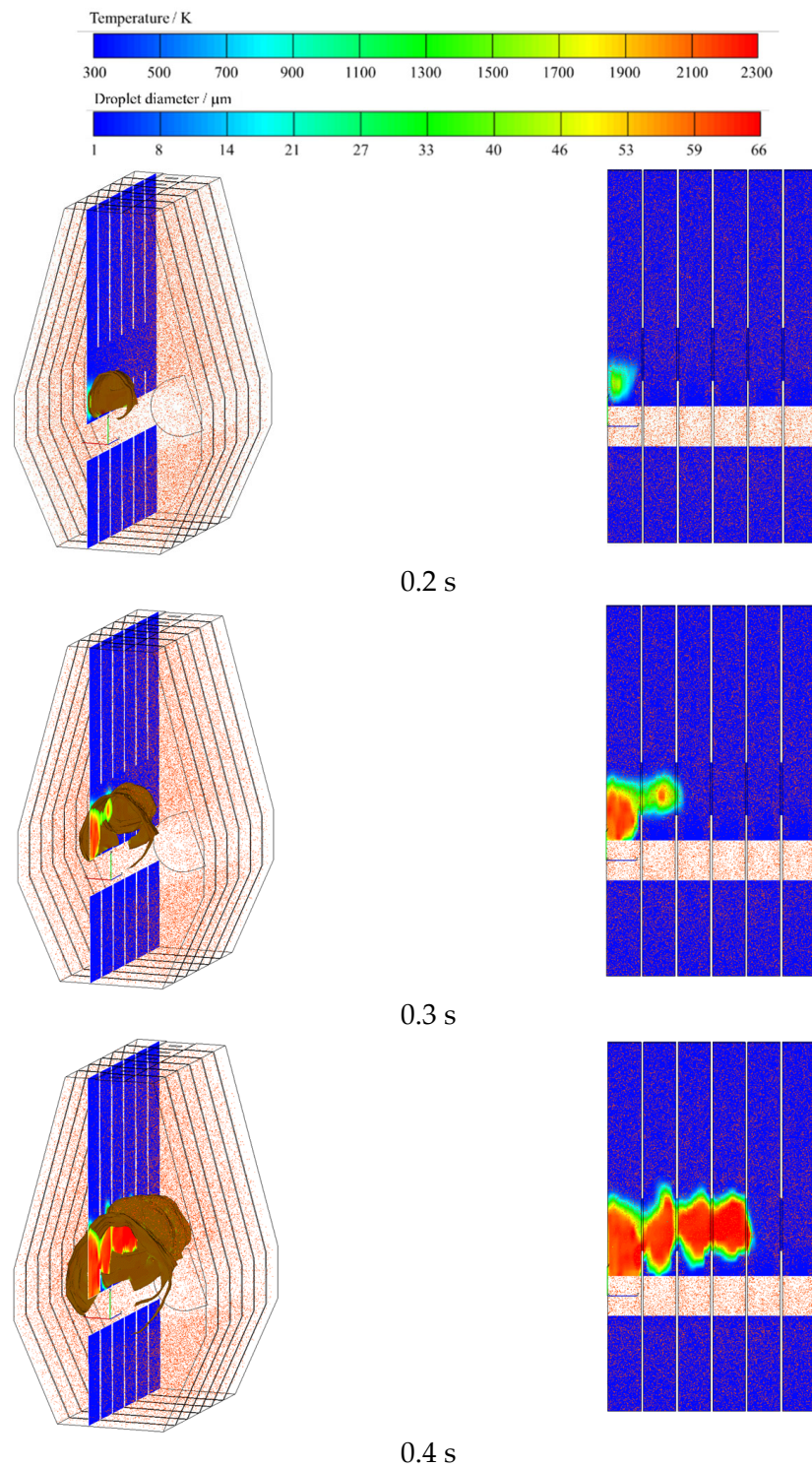


Figure 14. Cont.

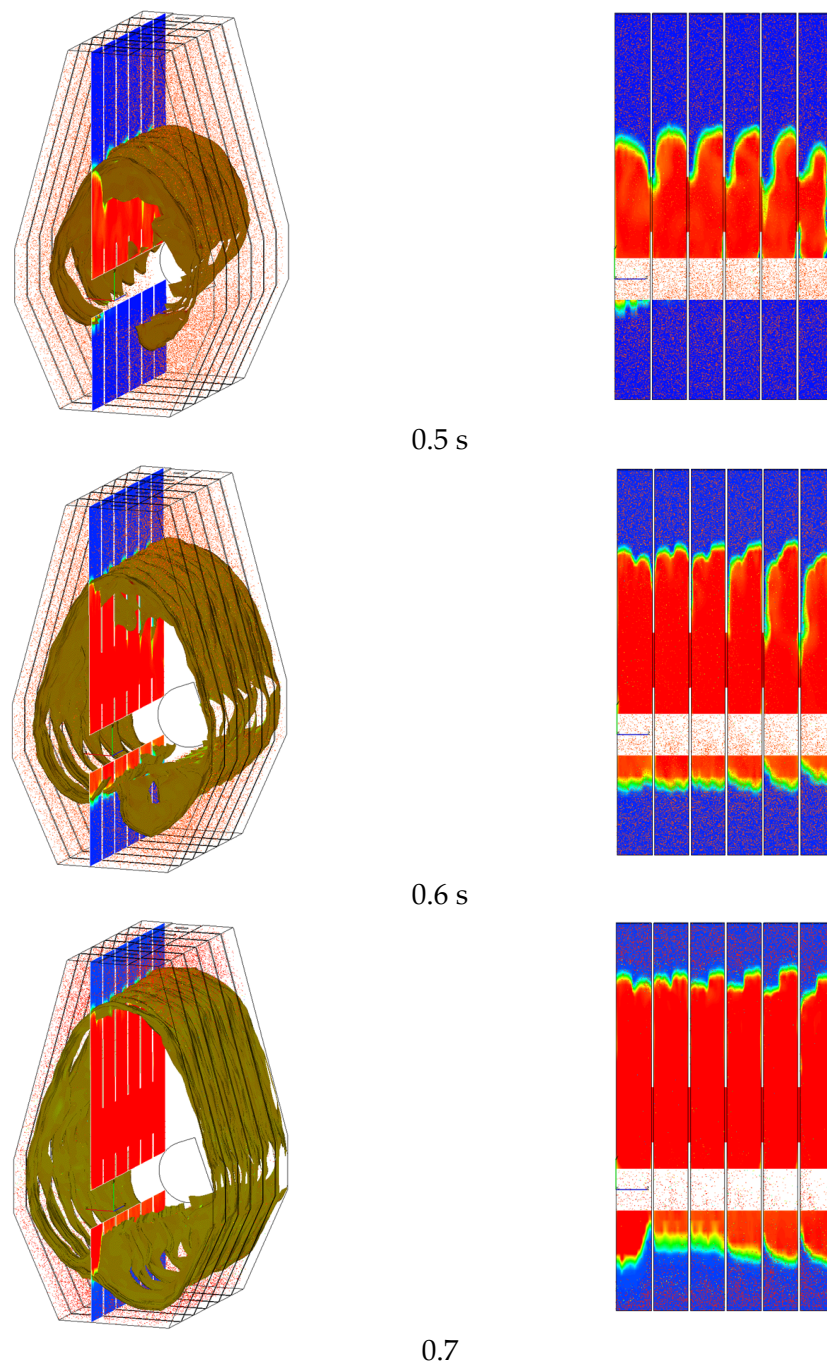


Figure 14. Calculated snapshots of gas temperature (fields), droplet diameter (dots), and flame surface evolution in the course of unvented crankcase explosion; ignition site #4 in crankcase segment I. Initial oil droplet diameter $d_0 = 60 \mu\text{m}$; the degree of oil pre-evaporation is 70%; time is counted from the start of ignition.

3.2.5. Unvented Explosion in Crankcase Segment IV: Ignition Site #1 (Variant 6)

The results discussed in Section 3.2.4 mean that ignition in the middle crankcase segments looks more dangerous than in the outer segments I or VI. To see whether this is true for ignition sites other than #4, let us compare the explosion dynamics in cases when the same ignition site (site #1) appears in different crankcase segments, e.g., in segment I and in segment IV. Figure 15 shows snapshots of gas temperature, oil mist droplet diameter, and flame surface evolution in the course of an unvented crankcase explosion presumably triggered by the leakage of hot blow-by gases through the faulty stuffing box (ignition

site #1 in Figure 1a) in crankcase segment IV. This test case can be compared with the test case considered in Section 3.2.1 where the same ignition site was applied to crankcase segment I. This comparison shows that the volume occupied by burned gas in the test under consideration is smaller than in the test considered in Section 3.2.1, i.e., flame accelerates faster when ignition occurs in crankcase segment I. This is probably caused by a faster pressure relief in segment IV through two windows in the A-frame than in segment I through a single window in the A-frame. Anyway, this result indicates that explosion evolution is highly dependent on the ignition site.

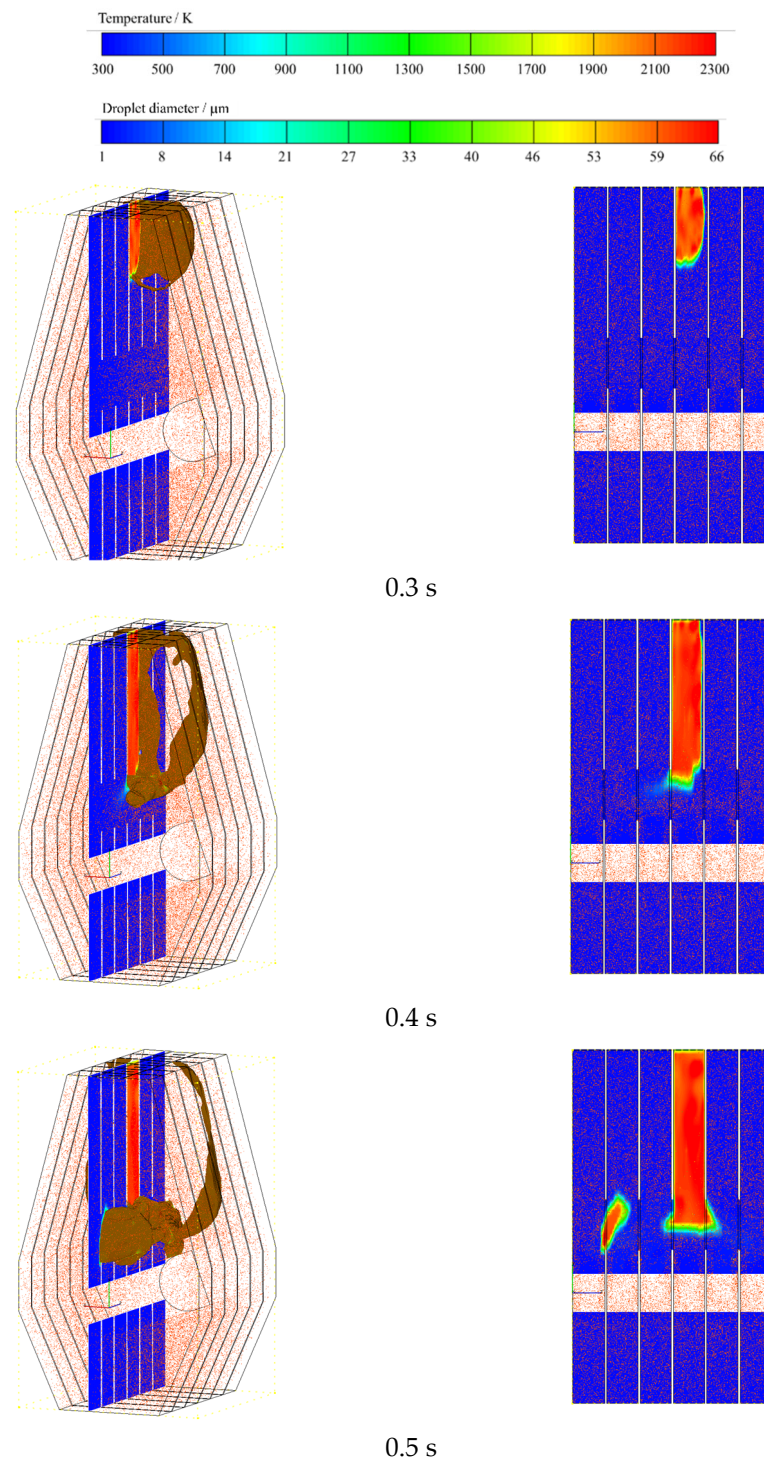


Figure 15. Cont.

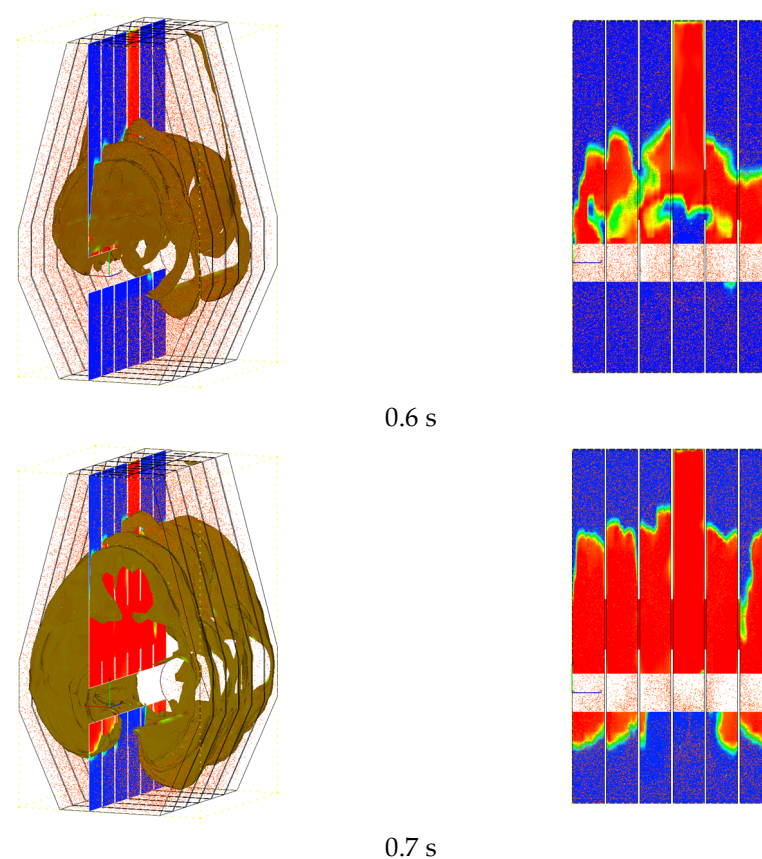


Figure 15. Calculated snapshots of gas temperature (fields), droplet diameter (dots), and flame surface evolution in the course of unvented crankcase explosion; ignition site #1 in crankcase segment IV. Initial oil droplet diameter $d_0 = 60 \mu\text{m}$; the degree of oil pre-evaporation is 70%; time is counted from the start of ignition.

3.2.6. Unvented Explosions in the Crankcase: Pressure Histories (Variants 1 to 6)

To summarize the results of calculations for unvented explosions, let us consider Figure 16 comparing the calculated pressure histories in the course of unvented crankcase explosions with different ignition scenarios. The pressure curves in the figure are marked as X(Y)u, where X is the ignition site from Figure 1a, Y is the crankcase segment from Figure 1b, and u relates to “unvented” explosion. The most dangerous explosions occur when ignition is presumably triggered by the hot spot on the main bearing, i.e., near the windows between the crankcase segments (curves 4(IV) and 4(I)). In these cases, the arising flame quickly breaks through the windows between the crankcase segments like it happens when the premixed flame propagates in a tube with orifice plates with sharp edges (see, e.g., Section 2.4.2) and in a vented vessel with internal obstacles [70]. The rapid breakthrough of the flame forms an extended cylindrical core of highly perturbed turbulent flame, which leads to the subsequent spread of the flame throughout the entire volume of the crankcase. It is worth noting that the scenario with accidental flame ignition at site #3 in the open space at the A-frame in crankcase segment I (see curve 3(I)) is also potentially very dangerous: initially, the rate of pressure rise in this case is the largest among all considered test cases. Accidental ignition by hot blow-by gases near the stuffing box (curves 1(I) and 1(IV)) initially leads to a fairly slow spread of the flame downward, but when the flame enters the window between the crankcase segments it accelerates so quickly that the rate of pressure rise in the crankcase reaches the highest value of 0.6–0.7 bar/s (see the arrows in Figure 16).

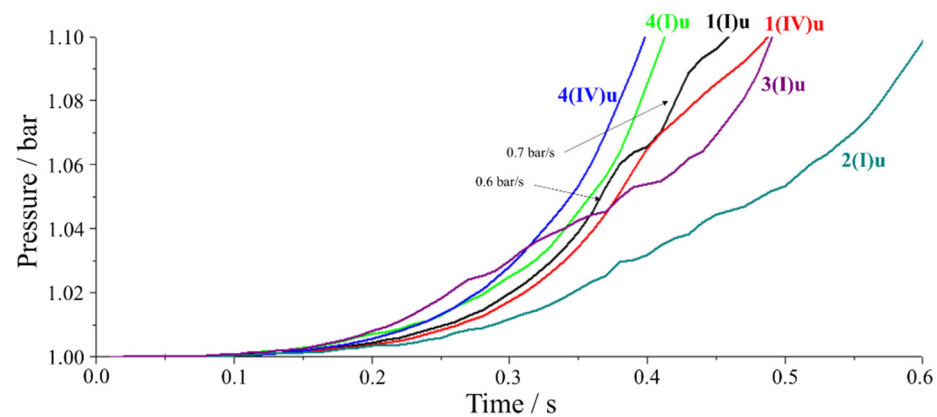


Figure 16. Comparison of calculated pressure histories in the course of unvented crankcase explosions. The pressure curves are marked as X(Y)u, where X is the ignition site from Figure 1a, Y is the crankcase segment from Figure 1b, and u denotes “unvented” explosion. Initial oil droplet diameter $d_0 = 60 \mu\text{m}$; the degree of oil pre-evaporation is 70%.

Figure 17 shows the calculated time histories of the overall rate of heat release and mass-averaged burning rate in the crankcase for the same scenarios of unvented explosions as in Figure 16. The maximum turbulent burning velocity is achieved at the stage when the flame makes its way into the window between the crankcase segments. For example, for the case 1(I)u, the apparent turbulent burning velocity at this stage attains 30–40 m/s, while the normal turbulent burning velocity is about 2–3 m/s in the open space and up to 7–8 m/s near obstructions. At a time of 0.4 s after ignition, the rate of heat release attains a huge value of 13 MW.

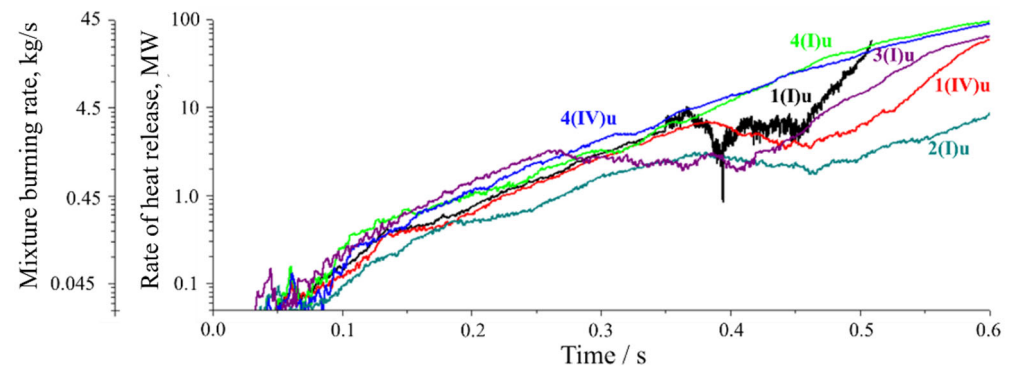


Figure 17. The calculated time histories of the overall rate of heat release and the overall mass-averaged burning rate in the course of unvented crankcase explosions. The curves are marked as X(Y)u, where X is the ignition site from Figure 1a, Y is the crankcase segment from Figure 1b, and u denotes “unvented” explosion. Initial oil droplet diameter $d_0 = 60 \mu\text{m}$; the degree of oil pre-evaporation is 70%.

3.2.7. Unvented Explosions in the Crankcase: Effect of Particle Size Distribution (Variants 1 and 7)

Let us check the effect of the droplet size distribution in the oil mist on the dynamics of pressure rise in the crankcase in the course of the seemingly most severe unvented explosion 4(IV)u. Two calculations were performed: one with the monodisperse mist with the initial droplet diameter $d_0 = 60 \mu\text{m}$ (baseline cases), and another with the polydisperse mist with the trimodal droplet size distribution: $d_0 = 10 \mu\text{m}$ (10 wt%), $250 \mu\text{m}$ (10 wt%), and $500 \mu\text{m}$ (80 wt%). Figure 18 compares the corresponding calculated pressure histories in the course of unvented crankcase explosions of oil mist with different droplet size distributions. As seen, the droplet size distribution plays a minor role in the development of crankcase explosion.

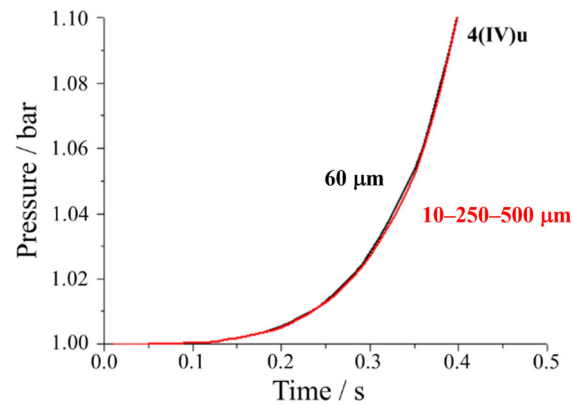


Figure 18. Comparison of calculated pressure histories in the course of unvented crankcase explosions of oil mist with different droplet size distributions (monodispersed mist with $d_0 = 60 \mu\text{m}$ and polydisperse mist with $d_0 = 10 \mu\text{m}$ (10 wt%), $250 \mu\text{m}$ (10 wt%), and $500 \mu\text{m}$ (80 wt%); the degree of oil pre-evaporation is 70%.

3.2.8. Unvented Explosions in the Crankcase: Effect of Mist Pre-Evaporation Degree (Variants 1, 8, and 9)

Finally, let us check the effect of the pre-evaporation degree of the oil mist on the dynamics of pressure rise in the crankcase in the course of the seemingly most severe unvented explosion 4(IV)u. Figure 19 compares three calculations for the oil mist with the same overall fuel-to-air equivalence ratio $\Phi = 1$ but with different pre-evaporation degrees: 60%, 70% (baseline case), and 80%. The decrease in the pre-evaporation degree from 70% to 60% is seen to significantly slow down the pressure rise in the crankcase. At a time of 0.4 s after ignition, the overpressure becomes a factor of 5 lower: 0.02 bar vs. 0.1 bar. An increase in the pre-evaporation degree from 70% to 80% leads to a significant acceleration in the rate of pressure rise. At a time of 0.3 s after ignition, the overpressure becomes a factor of over 3 higher: 0.1 bar vs. 0.03 bar. Thus, the pre-evaporation degree appears to be the strongest parameter affecting the rate of pressure rise. This finding corresponds well with the findings reported in [39] indicating that the heat in a fuel-lean droplet suspension is released mostly due to the premixed mode of combustion.

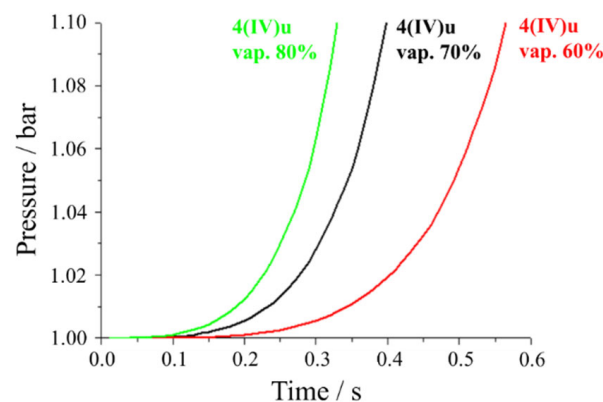


Figure 19. Comparison of calculated pressure histories in the course of unvented crankcase explosions of oil mist with different pre-evaporation degree. All curves correspond to ignition at site #4 in crankcase segment IV. Initial oil droplet diameter $d_0 = 60 \mu\text{m}$.

3.3. Vented Crankcase Explosions

Let us now consider the results for vented explosions with two different values of the critical overpressure of valve opening, namely, 0.05 and 0.005 bar.

3.3.1. Vented Explosion in Crankcase Segment IV: Ignition Site #4; Vent Opening Overpressure 0.05 Bar (Variant 10)

For understanding the effect of explosion venting, we chose the seemingly most severe explosion corresponding to curve 4(IV) in Figure 16. In the first test case, the vent opening overpressure was set at the level of 0.05 bar. Figure 20 shows snapshots of gas temperature, oil mist droplet diameter, and flame surface evolution in the course of a vented crankcase explosion presumably triggered by the hot spot on the main bearing (ignition site #4 in Figure 1a) in crankcase segment IV. This test case can be compared with the test case considered in Section 3.1 (see Figure 8) where the same ignition scenario was applied to the unvented explosion. At a time of 0.4 s after ignition, the vented explosion exhibits a flame that is less developed in the longitudinal direction but stretched toward the pressure relief valve in crankcase segment IV. Note that the pressure relief valves of all crankcase segments are already open by this time.

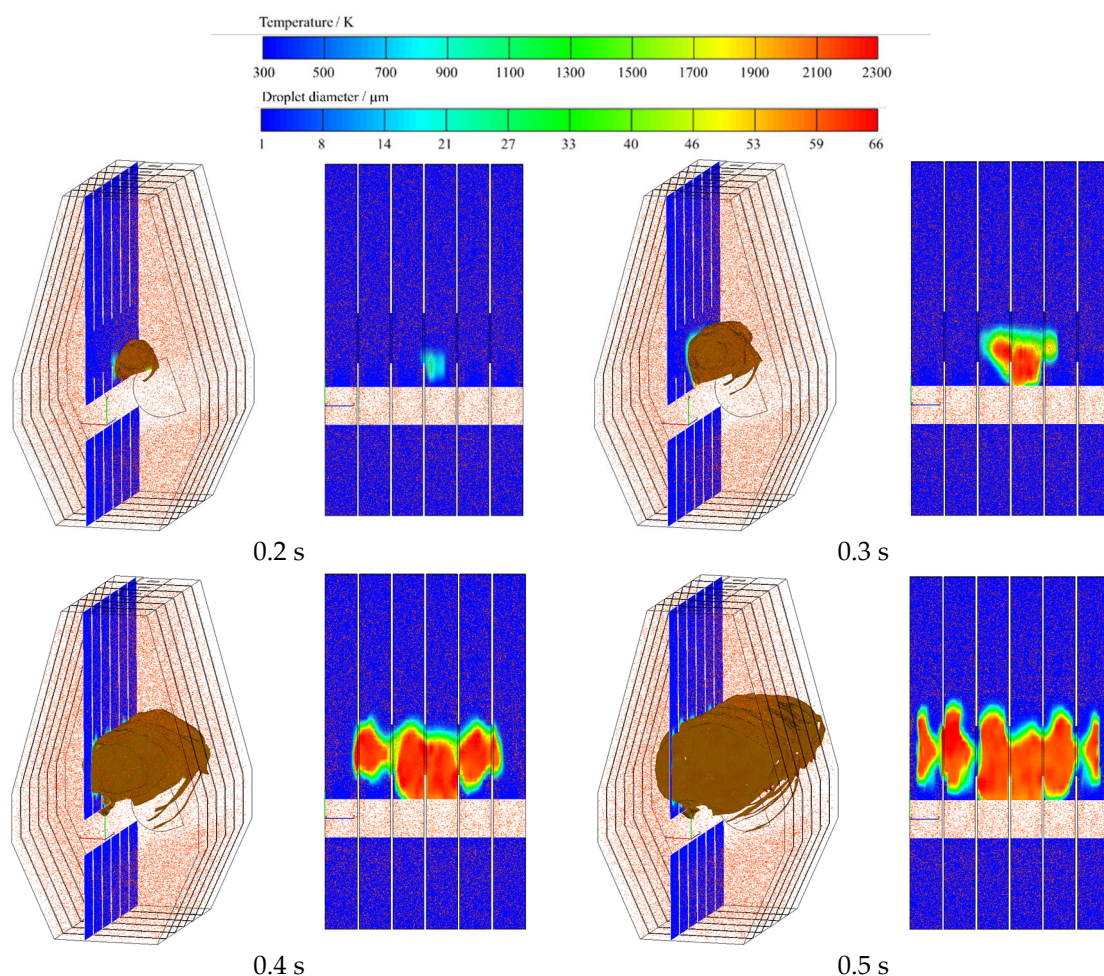


Figure 20. Calculated snapshots of gas temperature (fields), droplet diameter (dots), and flame surface evolution in the course of vented crankcase explosion; ignition site #4 in crankcase segment IV; vent opening overpressure is 0.05 bar. Initial oil droplet diameter $d_0 = 60 \mu\text{m}$; the degree of oil pre-evaporation is 70%; time is counted from the start of ignition.

3.3.2. Vented Explosion in Crankcase Segment IV: Ignition Site #4; Vent Opening Overpressure 0.005 Bar (Variant 11)

In the second test case, the vent opening overpressure is set at the lower level of 0.005 bar. Figure 21 shows snapshots of gas temperature, oil mist droplet diameter, and flame surface evolution in the course of a vented crankcase explosion presumably triggered by the hot spot on the main bearing (ignition site #4 in Figure 1a) in crankcase segment IV.

This test case can be compared with the test cases considered in Section 3.1 (see Figure 8) and in Section 3.3.1 (see Figure 20) where the same ignition scenario was applied to the unvented explosion and to the vented explosion with a higher value of vent opening overpressure (0.05 bar), respectively. At a time of 0.4 s after ignition, the vented explosion under consideration exhibits a flame that is less developed in the longitudinal direction but is still stretched toward the pressure relief valve in crankcase segment IV.

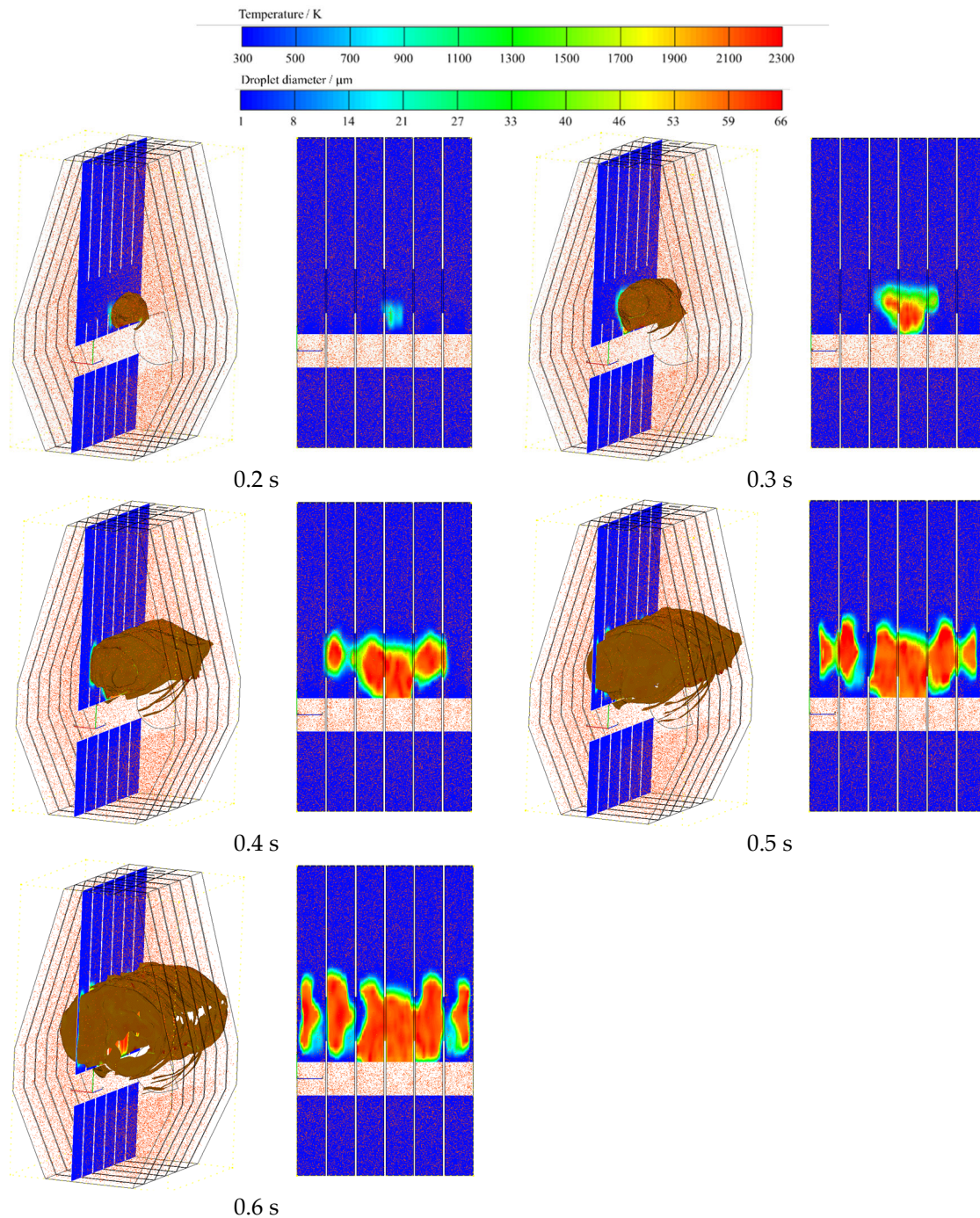


Figure 21. Calculated snapshots of gas temperature (fields), droplet diameter (dots), and flame surface evolution in the course of vented crankcase explosion; ignition site #4 in crankcase segment IV; vent opening overpressure is 0.005 bar. Initial oil droplet diameter $d_0 = 60 \mu\text{m}$; the degree of oil pre-evaporation is 70%; time is counted from the start of ignition.

Figure 22 compares the pressure histories in the course of unvented explosion on the one hand and two vented explosions on the other hand. The curves are marked as X(Y)u or X(Y)vZ, where X is the ignition site from Figure 1a, Y is the crankcase segment from Figure 1b, u and v correspond to “unvented” and “vented” explosions, respectively, and Z corresponds to the vent opening overpressure in bar. The arrows show two levels of vent opening overpressure, namely, 0.05 and 0.005 bar. As seen, explosion venting results in the dramatic change in the pressure histories: the rate of pressure rise decreases and even becomes negative, i.e., the crankcase pressure drops down. In both considered vented explosions, vent opening does not lead to an immediate pressure drop in the crankcase: the pressure keeps growing for a certain time and attains a maximum value that can be 40% higher (0.07 bar vs. 0.05 bar for the curve 4(IV)v0.05) and a factor of 2 higher (0.01 bar vs. 0.005 bar for the curve 4(IV)v0.005) than the vent opening pressure. In addition, Figure 23 shows the calculated time histories of the overall rate of heat release in the crankcase for the same scenarios of vented explosions as in Figure 22. For both vented explosions, vent opening results in a drastic increase in the overall rate of heat release. Thus, for the vent opening overpressure of 0.05 bar, by the time of 0.4 s, it attains a huge value of 23 MW instead of 13 MW for the unvented explosion. For the vent opening overpressure of 0.005 bar, by the time of 0.4 s, it attains a value of 10 MW, which is only 23% less than that for the unvented explosion. Such an increase in the overall rate of heat release is caused by the turbulence induced by crankcase depressurization and the corresponding increase in the turbulent flame surface and turbulent burning velocity.

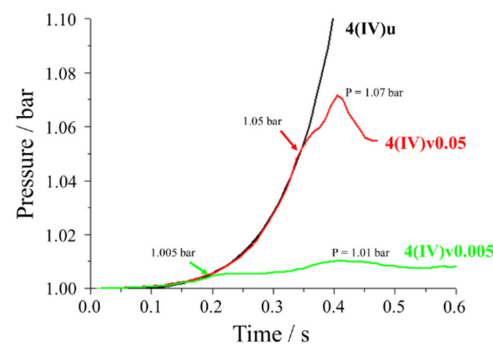


Figure 22. Comparison of calculated pressure histories in the course of vented crankcase explosions. The pressure curves are marked as X(Y)u and X(Y)vZ, where X is the ignition site from Figure 1a, Y is the crankcase segment from Figure 1b, u and v denote “unvented” and “vented” explosions, and Z denotes the vent opening overpressure. Initial oil droplet diameter $d_0 = 60 \mu\text{m}$; the degree of oil pre-evaporation is 70%.

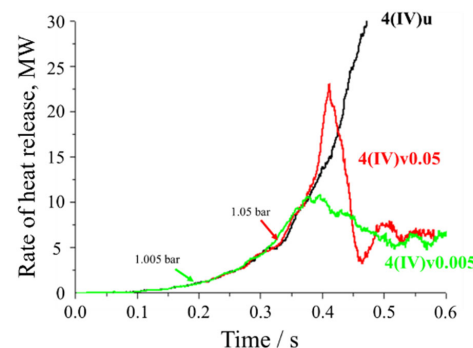


Figure 23. The calculated time histories of the overall rate of heat release in the course of vented crankcase explosions. The curves are marked as X(Y)u and X(Y)vZ, where X is the ignition site from Figure 1a, Y is the crankcase segment from Figure 1b, u and v denote “unvented” and “vented” explosions, and Z denotes the vent opening overpressure. Initial oil droplet diameter $d_0 = 60 \mu\text{m}$; the degree of oil pre-evaporation is 70%.

4. Conclusions

Unvented and vented crankcase explosions of the lubricating oil mist were numerically simulated using a 3D CFD approach for a two-phase turbulent reactive flow with finite-rate turbulent/molecular mixing and chemistry. A typical low-speed two-stroke six-cylinder marine diesel engine was considered as an example. Four possible accidental ignition sites were considered in different linked segments of the crankcase. The calculations are limited by a relatively low explosion overpressure of 0.1 bar in view of possible explosion venting. As a result of our calculations, the following findings are worth emphasizing:

- (1) The most important parameter affecting the dynamics of crankcase explosion is the pre-evaporation degree of the oil mist, whereas the oil droplet size distribution is shown to play a minor role. This means that the major role in the pressure build-up in these conditions is played by the combustion of pre-evaporated oil.
- (2) Explosion dynamics are highly dependent on the accidental ignition site. Calculations of unvented crankcase explosions reveal a seemingly most severe explosion scenario caused by the hot spot ignition of the oil mist on the main bearing and flame breaking through the windows connecting the crankcase segments.
- (3) The predicted maximum rate of pressure rise in the unvented crankcase attains 0.6–0.7 bar/s. This places strict requirements on the response time of pressure relief valves for explosion venting.
- (4) The apparent turbulent burning velocity in the unvented crankcase attains 7–8 m/s and the rate of heat release attains a huge value of 13 MW.
- (5) Explosion venting causes the rate of pressure rise to decrease and become negative. However, vent opening does not lead to an immediate pressure drop in the crankcase: the pressure keeps growing for a certain time and attains a maximum value which can be a factor of 2 higher than the vent opening pressure.
- (6) Vent opening results in a drastic increase in the overall rate of heat release caused by the turbulence induced by crankcase depressurization. Thus, for a vent opening overpressure of 0.05 bar, the overall rate of heat release attains the value of 23 MW instead of 13 MW for the unvented explosion by the same time after accidental ignition.

Future work will be focused on considering both internal and external vented crankcase explosions while accounting for the turbulence generated by moving engine parts.

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References

1. Lamb, J. Explosions in enclosed crankcases of reciprocating engines: Their cause, effect, and possible remedy. *Proc. Inst. Mech. Eng.* **1952**, *166*, 327–349. [\[CrossRef\]](#)
2. Burgoyne, J.H.; Newitt, D.M. Crankcase explosions in marine engines. *Nav. Eng. J.* **1956**, *68*, 122–128. [\[CrossRef\]](#)
3. Freeston, H.G.; Roberts, J.D.; Thomas, A. Crankcase explosions: An investigation into some factors governing the selection of protective devices. *Proc. Inst. Mech. Eng.* **1956**, *170*, 811–824. [\[CrossRef\]](#)
4. Woodyard, D. *Pounder's Marine Diesel Engines and Gas Turbines*, 9th ed.; Butterworth-Heinemann Elsevier Ltd.: Oxford, UK, 2009; pp. 893–907. [\[CrossRef\]](#)
5. Atherton, J.G. The Scientific investigation of marine fires and explosions. *Mar. Technol. Soc. J.* **2012**, *46*, 129–141. [\[CrossRef\]](#)

6. Ünver, B.; Gürgen, S.; Sahin, B.; Altın, İ. Crankcase explosion for two-stroke marine diesel engine by using fault tree analysis method in fuzzy environment. *Eng. Fail. Anal.* **2019**, *97*, 288–299. [\[CrossRef\]](#)
7. Chybowski, L. Study of the relationship between the level of lubricating oil contamination with distillation fuel and the risk of explosion in the crankcase of a marine trunk type engine. *Energies* **2023**, *16*, 683. [\[CrossRef\]](#)
8. Johnson, B.T.; Hargrave, G.K.; Reid, B.A.; Page, V.J.; Wagstaff, S. *Optical Analysis and Measurement of Crankcase Lubricant Oil Atomization*; SAE Technical Paper 2012-01-0882; Society of Automotive Engineers (SAE): Warrendale, PA, USA, 2012.
9. Michalek, D.J.; Hii, W.W.-S.; Sun, J.; Gunter, K.L.; Sutherland, J.W. Experimental and analytical efforts to characterize cutting fluid mist formation and behavior in machining. *Appl. Occup. Environ. Hyg.* **2003**, *18*, 842–854. [\[CrossRef\]](#)
10. Nguyen, T.; Zhang, L.C. Modelling of the mist formation in a segmented grinding wheel system. *Int. J. Mach. Tools Manufact.* **2005**, *45*, 21–28. [\[CrossRef\]](#)
11. Thirouard, B.; Tian, T.; Hart, D.P. Investigation of oil transport mechanisms in the piston ring pack of a single cylinder diesel engine, using two dimensional laser induced fluorescence. *J. Fuels Lubr.* **1998**, *107*, 2007–2015.
12. Gamble, R.J.; Priest, M.; Taylor, C.M. Detailed analysis of oil transport in the piston assembly of a gasoline engine. *Tribol. Lett.* **2002**, *14*, 147–156. [\[CrossRef\]](#)
13. Day, L.; Dunaevsky, V.; McCormick, H. Critical factors affecting oil consumption and deposit formation in engines and compressors come to light from research in two disciplines. *Tribol. Lubr. Technol. STLE* **2008**, *64*, 31–39.
14. Eckhoff, R.K. *Generation, Ignition, Combustion and Explosion of Sprays and Mists of Flammable Liquids in Air: A literature Survey*; Offshore Technology Report—OTN 95 260; Health and Safety Executive: Bootle, UK, 1995.
15. Przesmitzki, S.; Tian, T. Oil transport inside the power cylinder during transient load changes. *J. Engines* **2007**, *116*, 652–667.
16. Hayashi, S.; Kumagai, S. Flame propagation in fuel droplet-vapor-air mixtures. *Proc. Combust. Inst.* **1975**, *15*, 445–452. [\[CrossRef\]](#)
17. Hayashi, S.; Kumagai, S.; Sakai, T. Propagation velocity and structure of flames in droplet-vapor-air mixtures. *Combust. Sci. Technol.* **1977**, *15*, 169–177. [\[CrossRef\]](#)
18. Cameron, L.R.J.; Bowen, P.J. Novel cloud chamber design for “transition range” aerosol combustion studies. *Process Saf. Environ. Prot.* **2001**, *79*, 197–205. [\[CrossRef\]](#)
19. Bradley, D.; Lawes, M.; Liao, S.; Saat, A. Laminar mass burning and entrainment velocities and flame instabilities of i-octane, ethanol and hydrous ethanol/air aerosols. *Combust. Flame* **2014**, *161*, 1620–1632. [\[CrossRef\]](#)
20. Fayad, M.A.; Martos, F.J.; Herreros, J.M.; Dearn, K.D.; Tsolakis, A. Characterisation of soot agglomerates from engine oil and exhaust system for modern compression ignition engines. *Int. J. Engine Res.* **2023**, *24*, 4297–4304. [\[CrossRef\]](#)
21. Mansfield, W.P. Crankcase explosions: Development of new protective devices. *Proc. Inst. Mech. Eng.* **1956**, *170*, 825–862. [\[CrossRef\]](#)
22. International Association of Classification Societies. *Type Testing Procedure for Crankcase Explosion Relief Valves*; IACS: Kolkata, India, 2008; pp. 1–8.
23. Kong, K.-J.; Kang, S.-W.; Kim, J.-H.; Jang, H.-L. Numerical analysis for performance and the combustion reactants of the crankcase explosion relief valve. *J. Mar. Sci. Eng.* **2022**, *10*, 1340. [\[CrossRef\]](#)
24. Jiang, L.; Zhang, S.; Shu, X.; Hao, W.; Ren, Y.; Chen, Z. Innovative design of leeward surface of pin fin in flame arresters fitted in explosion relief valve. In *Advances in Mechanical Design, ICMD*; Tan, J., Ed.; Mechanisms and Machine Science; Springer: Singapore, 2019; Volume 77. [\[CrossRef\]](#)
25. Dasch, J.M.; D’Arcy, J.B.; Kinare, S.S.; Yin, Y.; Kopple, R.G.; Salmon, S.C. Mist generation from high-speed grinding with straight oils. *Tribol. Trans.* **2008**, *51*, 381–388. [\[CrossRef\]](#)
26. Behn, A.; Feindt, M.; Matz, G.; Krause, S. *Fuel Transport across the Piston Ring Pack: Measurement System Development and Experiments for Online Fuel Transport and Oil Dilution Measurements*; SAE Technical Paper 2015-01-2535; Society of Automotive Engineers (SAE): Warrendale, PA, USA, 2015.
27. Dollmeyer, T.A.; Vittorio, D.A.; Grana, T.A.; Katzenmeyer, J.R.; Charlton, S.J.; Clerc, J.; Morphet, R.G.; Schwandt, B.W. *Meeting the US 2007 Heavy-Duty Diesel Emission Standards—Designing for the Customer*; SAE Technical Paper 2007-01-4170; SAE International: Warrendale, PA, USA, 2007.
28. Begg, S.M.; De-Sercey, G.; Miche, N.D.D.; Heikal, M.R.; Gilchrist, R.; Noda, Y.; Tsuruoka, Y.; Mamiya, Y. Experimental investigation of the phenomenon of oil breakup in an engine crankcase. *At. Sprays* **2010**, *20*, 801–819. [\[CrossRef\]](#)
29. Wang, Z.; Yi, P.; Qu, W.; Feng, L.; Gong, Z. Numerical simulation of monodisperse lube oil multiple droplet evaporation and autoignition under nonconstant cylinder conditions of low-speed two-stroke gas engines. *ACS Omega* **2021**, *6*, 33568–33582. [\[CrossRef\]](#) [\[PubMed\]](#)
30. Dyson, C.J.; Priest, M.; Lee, P.M. The flow of lubricant as a mist in the piston assembly and crankcase of a fired gasoline engine. *Tribol. Lett.* **2023**, *71*, 12. [\[CrossRef\]](#)
31. Gant, S. *Generation of Flammable Mists from High Flashpoint Fluids: Literature Review*; HSE Research Report RR980; Health and Safety Executive: Bootle, UK, 2013.
32. Yuan, S.; Ji, C.; Monhollen, A.; Kwon, J.S.-I.; Mashuga, C. Experimental and thermodynamic study of aerosol explosions in a 36 L apparatus. *Fuel* **2019**, *245*, 467–477. [\[CrossRef\]](#)
33. El-Zahlanieh, S.; Dos Santos, I.S.; Sivabalan, S.; Brunello, D.; Tribouilloy, B.; Vignes, A.; Dufaud, O. Finding a way through the “misty” evaluation of the flammability and explosivity of kerosene aerosols. *Fuel* **2022**, *328*, 125275. [\[CrossRef\]](#)

34. Jackson, G.S.; Avedisian, C.T. Modeling of spherically symmetric droplet flames including complex chemistry: Effect of water addition on n-heptane droplet combustion. *Combust. Sci. Technol.* **1996**, *115*, 125–149. [\[CrossRef\]](#)
35. Marchese, A.; Dryer, F.; Nayagam, V. Numerical modeling of isolated n-alkane droplet flames: Initial comparisons with ground and space-based microgravity experiments. *Combust. Flame* **1999**, *116*, 432–459. [\[CrossRef\]](#)
36. Frolov, S.M.; Basevich, V.Y. Simulation of low-temperature oxidation and combustion of n-dodecane droplets under microgravity conditions. *Fire* **2023**, *6*, 70. [\[CrossRef\]](#)
37. Suganuma, Y.; Ikeyama, N.; Nomura, H.; Ujiie, Y. Droplet array combustion experiments on effect of initial droplet diameter on flame spread characteristic time. *Trans. Japan Soc. Aero. Space Sci.* **2017**, *60*, 127–131. [\[CrossRef\]](#)
38. Li, Q.; Zhang, H.; Shu, C. Propagation of heterogeneous and homogeneous planar flames in fuel droplet mists. *Int. J. Multiph. Flow* **2020**, *133*, 103452. [\[CrossRef\]](#)
39. Erol, G.O.; Hasslberger, J.; Klein, M.; Chakraborty, N. A direct numerical simulation investigation of spherically expanding flames propagating in fuel droplet-mists for different droplet diameters and overall equivalence ratios. *Combust. Sci. Technol.* **2019**, *191*, 833–867. [\[CrossRef\]](#)
40. Atzler, F.; Demoulin, F.X.; Lawes, M.; Lee, Y. Oscillations in the flame speed of globally homogeneous two-phase mixtures. In Proceedings of the 18th International Colloquium on the Dynamics of Explosions and Reactive Systems, Seattle, WA, USA, 29 July–3 August 2001; p. 83.
41. Paulhiac, D.; Cuenot, B.; Riber, E.; Esclapez, L.; Richard, S. Analysis of the spray flame structure in a lab-scale burner using Large Eddy Simulation and Discrete Particle Simulation. *Combust. Flame* **2020**, *212*, 25–38. [\[CrossRef\]](#)
42. Chigier, N.A. Instrumentation techniques for studying heterogeneous combustion. *Prog. Energy Combust. Sci.* **1977**, *3*, 175–189. [\[CrossRef\]](#)
43. Law, C.K. Recent advances in droplet vaporization and combustion. *Prog. Energy Combust. Sci.* **1982**, *8*, 171–201. [\[CrossRef\]](#)
44. Sirignano, W.A. Fuel droplet vaporization and spray combustion theory. *Prog. Energy Combust. Sci.* **1983**, *9*, 291–322. [\[CrossRef\]](#)
45. Kikuchi, M.; Arai, T.; Yoda, S.; Tsukamoto, T.; Umemura, A.; Uchida, M.; Kakei, M.; Niioka, T. Numerical study on flame propagation of a fuel droplet array in a high-temperature environment under microgravity. *Proc. Combust. Inst.* **2002**, *29*, 2611–2619. [\[CrossRef\]](#)
46. Fujita, A.; Watanabe, H.; Kurose, R.; Komori, S. Two-dimensional direct numerical simulation of spray flames—Part 1: Effects of equivalence ratio, fuel droplet size and radiation, and validity of flamelet model. *Fuel* **2013**, *104*, 515–525. [\[CrossRef\]](#)
47. Haruki, Y.; Pillai, A.L.; Kitano, T.; Kurose, R. Numerical investigation of flame propagation in fuel droplet arrays. *At. Sprays* **2018**, *28*, 357–388. [\[CrossRef\]](#)
48. Mizutani, Y.; Nakajima, A. Combustion of fuel vapor-drop-air systems: Part II—Spherical flames in a vessel. *Combust. Flame* **1973**, *20*, 351–357. [\[CrossRef\]](#)
49. El Banhawly, Y.; Whitelaw, J.H. Experimental study of the interaction between a fuel spray and surrounding combustion air. *Combust. Flame* **1981**, *42*, 253–275. [\[CrossRef\]](#)
50. Myers, G.D.; Lefebvre, A.H. Flame propagation in heterogeneous mixtures of fuel drops and air. *Combust. Flame* **1986**, *66*, 193–210. [\[CrossRef\]](#)
51. Richards, G.A.; Lefebvre, A.H. Turbulent flame speeds of hydrocarbon fuel droplets in air. *Combust. Flame* **1989**, *78*, 299–307. [\[CrossRef\]](#)
52. Stårner, S.H.; Gounder, J.; Masri, A.R. Effects of turbulence and carrier fluid on simple, turbulent spray jet flames. *Combust. Flame* **2005**, *143*, 420–432. [\[CrossRef\]](#)
53. Lawes, M.; Saat, A. Burning rates of turbulent iso-octane aerosol mixtures in spherical flame explosions. *Proc. Combust. Inst.* **2011**, *33*, 2047–2054. [\[CrossRef\]](#)
54. De Oliveira, P.M.; Allison, P.M.; Mastorakos, E. Ignition of uniform droplet-laden weakly turbulent flows following a laser spark. *Combust. Flame* **2019**, *199*, 387–400. [\[CrossRef\]](#)
55. Bartknecht, W. *Explosions—Course, Prevention, Protection*; Springer: Berlin, Germany, 1981.
56. Phylaktou, H.; Andrews, G.E. Gas explosions in linked vessels. *J. Loss Prev. Process Ind.* **1993**, *6*, 15–19. [\[CrossRef\]](#)
57. Cashdollar, K.L.; Zlochower, I.A.; Green, G.M.; Thomas, R.A.; Hertzberg, M. Flammability of methane, propane, and hydrogen gases. *J. Loss Prev. Process Ind.* **2000**, *13*, 327–340. [\[CrossRef\]](#)
58. Cammarota, F.; Di Benedetto, A.; Russo, P.; Salzano, E. Experimental analysis of gas explosions at non-atmospheric initial conditions in cylindrical vessel. *Process Saf. Environ. Prot.* **2010**, *88*, 341–349. [\[CrossRef\]](#)
59. Salzano, E.; Cammarota, F.; Di Benedetto, A.; Di Sarli, V. Explosion behavior of hydrogen–methane/air mixtures. *J. Loss Prev. Process Ind.* **2012**, *25*, 443–447. [\[CrossRef\]](#)
60. Bradley, D.; Mitcheson, A. The venting of gaseous explosions in spherical vessels. I—Theory. *Combust. Flame* **1978**, *32*, 221–236. [\[CrossRef\]](#)
61. Bradley, D.; Mitcheson, A. The venting of gaseous explosions in spherical vessels. II—Theory and experiment. *Combust. Flame* **1978**, *32*, 237–255. [\[CrossRef\]](#)
62. Van Wingerden, C.J.M.; Zeeuwen, J.P. On the role of acoustically driven flame instabilities in vented gas explosions and their elimination. *Combust. Flame* **1983**, *51*, 109–111. [\[CrossRef\]](#)
63. Cooper, M.G.; Fairweather, M.; Tite, J.P. On the mechanisms of pressure generation in vented explosions. *Combust. Flame* **1986**, *65*, 1–14. [\[CrossRef\]](#)

64. Frolov, S.M.; Gelfand, B.E.; Tsyganov, S.A. A possible mechanism for the onset of pressure oscillation during venting. *J. Loss Prev. Process Ind.* **1990**, *3*, 64–67. [\[CrossRef\]](#)
65. Maremonti, M.; Russo, G.; Salzano, E.; Tufano, V. Numerical simulation of gas explosions in linked vessels. *J. Loss Prev. Process Ind.* **1999**, *12*, 189–194. [\[CrossRef\]](#)
66. Ferrara, G.; Willacy, S.K.; Phylaktou, H.N.; Andrews, G.E.; Di Benedetto, A.; Salzano, E.; Russo, G. Venting of gas explosion through relief ducts: Interaction between internal and external explosions. *J. Hazard. Mater.* **2008**, *155*, 358–368. [\[CrossRef\]](#) [\[PubMed\]](#)
67. Molkov, V.; Makarov, D.; Puttock, J. The nature of coherent deflagrations. In Proceedings of the Fifth International Symposium on Hazards, Prevention and Mitigation of Industrial Explosions, Krakow, Poland, 10–14 October 2004.
68. Di Sarli, V.; Di Benedetto, A.; Russo, G. Using Large Eddy Simulation for understanding vented gas explosions in the presence of obstacles. *J. Hazard. Mater.* **2009**, *169*, 435–442. [\[CrossRef\]](#) [\[PubMed\]](#)
69. Wang, Q.; Sun, Y.; Li, X.; Shu, C.-M.; Wang, Z.; Jiang, J.; Zhang, M.; Cheng, F. Process of natural gas explosion in linked vessels with three structures obtained using numerical simulation. *Processes* **2020**, *8*, 52. [\[CrossRef\]](#)
70. Bauwens, C.R.; Chaffee, J.; Dorofeev, S.B. Vented explosion overpressures from combustion of hydrogen and hydrocarbon mixtures. *Int. J. Hydrogen Energy* **2011**, *36*, 2329–2336. [\[CrossRef\]](#)
71. Keenan, J.J.; Makarov, D.V.; Molkov, V.V. Rayleigh-Taylor instability: Modelling and effect on coherent deflagrations. *Int. J. Hydrogen Energy* **2014**, *39*, 20467–20473. [\[CrossRef\]](#)
72. Vyazmina, E.; Jallais, S. Validation and recommendations for CFD and engineering modeling of hydrogen vented explosions: Effects of concentration, stratification, obstruction and vent area. *Int. J. Hydrogen Energy* **2016**, *41*, 15101–15109. [\[CrossRef\]](#)
73. Talias, I.C.; Stewart, J.R.; Newton, A.; Keenan, J.; Makarov, D.; Hoyes, J.R. Numerical simulations of vented hydrogen deflagrations in a medium-scale enclosure. *J. Loss Prev. Process Ind.* **2018**, *52*, 125–139. [\[CrossRef\]](#)
74. Li, H.; Chen, X.; Deng, J.; Shu, C.-M.; Kuo, C.-H.; Yu, Y.; Hua, X. CFD analysis and experimental study on the effect of oxygen level, particle size, and dust concentration on the flame evolution characteristics and explosion severity of cornstarch dust cloud deflagration in a spherical chamber. *Powder Technol.* **2020**, *372*, 585–599. [\[CrossRef\]](#)
75. Islasa, A.; Fernandez, A.R.; Betegon, C.; Martínez-Panedac, E.; Pandala, A. Biomass dust explosions: CFD simulations and venting experiments in a 1 m³ silo. *J. Loss Prev. Process Ind.* **2023**, *83*, 105029. [\[CrossRef\]](#)
76. Portarapillo, M.; Trofa, M.; Sanchirico, R.; Di Benedetto, A. CFD simulations of the effect of dust diameter on the dispersion in the 1 m³ explosion vessel. *Chem. Eng. Trans.* **2021**, *86*, 343–348. [\[CrossRef\]](#)
77. Frolov, S.M.; Ivanov, V.S.; Basara, B.; Suffa, M. Numerical simulation of flame propagation and localized preflame autoignition in enclosures. *J. Loss Prev. Process Ind.* **2013**, *26*, 302–309. [\[CrossRef\]](#)
78. MAN B&W G95ME-C9.5 IMO Tier II Project Guide. Available online: <https://www.dieselduck.info/machine/01%20prime%20movers/2015%20MAN%20G95ME-C9%20Project%20Guide.pdf> (accessed on 21 November 2023).
79. Crankcase Explosions in Two-stroke Diesel Engines. Available online: <https://www.dieselduck.info/machine/06%20safety/2003%20MAN%20Crankcase%20explosions.pdf> (accessed on 8 November 2023).
80. Zaccardi, J.-M.; Escudié, D. Overview of the main mechanisms triggering low-speed pre-ignition in spark-ignition engines. *Int. J. Engine Res.* **2014**, *16*, 152–165. [\[CrossRef\]](#)
81. Distaso, E.; Amirante, R.; Calò, G.; De Palma, P.; Tamburrano, P.; Reitz, R.D. Predicting lubricant oil induced pre-ignition phenomena in modern gasoline engines: The reduced GasLube reaction mechanism. *Fuel* **2020**, *281*, 118709. [\[CrossRef\]](#)
82. Petersen, E.L.; Mathieu, O.; Thomas, J.C.; Cooper, S.P.; Teitge, D.S.; Juárez, R.; Gutierrez, N.; Mashuga, C.V. Combustion and oxidation of lube oils at gas turbine conditions: Experimental methods. In Proceedings of the ASME Turbo Expo 2021: Turbomachinery Technical Conference and Exposition, Virtual, 7–11 June 2021. [\[CrossRef\]](#)
83. Williams, F.A. *The Combustion Theory*; CRC Press: Boca Raton, FL, USA, 1994.
84. Available online: <https://tnfworkshop.org/radiation/> (accessed on 21 November 2023).
85. Reitz, R.D. Mechanism of atomization processes in high-pressure vaporizing sprays. *At. Spray Technol.* **1987**, *3*, 309–337.
86. Dukowicz, J.K. Quasi-steady droplet change in the presence of convection. In *Rep. LA7997-MS*; Los Alamos Scientific Laboratory: Santa Fe, NM, USA, 1979.
87. Frolov, S.M.; Frolov, F.S.; Basara, B. Simple model of transient drop vaporization. *J. Russ. Laser Res.* **2006**, *27*, 562–574. [\[CrossRef\]](#)
88. Basevich, V.Y.; Frolov, S.M.; Ivanov, V.S.; Frolov, F.S.; Semenov, I.V. The effects of multistage fuel-oxidation chemistry, soot radiation, and real gas properties on the operation process of compression ignition engines. *Eng* **2023**, *4*, 2682–2710. [\[CrossRef\]](#)
89. Anderson, F.L. Huygens' principle geometric derivation and elimination of the wake and backward wave. *Sci. Rep.* **2021**, *11*, 20257. [\[CrossRef\]](#) [\[PubMed\]](#)
90. Shchelkin, K.I. *Fast Combustion and Spin Detonation of Gases*; Voenizdat Publishing: Moscow, Russia, 1949.
91. Belyaev, A.A.; Basevich, V.Y.; Frolov, S.M. Database for calculating laminar and turbulent combustion of aviation kerosene—Air mixtures. *Combust. Explos.* **2015**, *8*, 29–36.
92. Li, B.; Zhang, H.; Egolfopoulos, F.N. Laminar flame propagation of atmospheric iso-cetane/air and decalin/air mixtures. *Combust. Flame* **2014**, *161*, 154–161. [\[CrossRef\]](#)
93. Frolov, S.M. Acceleration of the deflagration-to-detonation transition in gases: From Shchelkin to our days. *Combust. Explos. Shock Waves* **2012**, *48*, 258–268. [\[CrossRef\]](#)

94. Fogliarino, M. Crankcase Pressure Control in an Internal Combustion Engine: GT-Power Simulation. Master's Thesis, University of Windsor, Windsor, ON, Canada, 2014; p. 5218. Available online: <https://scholar.uwindsor.ca/etd/5218> (accessed on 21 November 2023).
95. Marthur, S.R.; Murthy, J.Y. A pressure-based method for unstructured meshes. *Numer. Heat Transf. B* **1997**, *31*, 195–215. [[CrossRef](#)]
96. Patankar, S.V.; Spalding, D.B. A calculation procedure for heat, mass and momentum transfer in three-dimensional parabolic flows. *Int. J. Heat Mass Transf.* **1972**, *15*, 1510–1520. [[CrossRef](#)]
97. Jarosinski, J.; Podfilipski, J.; Gorczakowski, A.; Veyssiere, B. Experimental study of flame propagation in propane-air mixture near rich flammability limits in microgravity. *Combust. Sci. Technol.* **2002**, *174*, 21–48. [[CrossRef](#)]
98. Ciccarelly, G. Effects of obstacle size and spacing on the initial stage of flame acceleration in an obstacle laden tube. In Proceedings of the 19th International Colloquium on the Dynamics of Explosions and Reactive Systems (ICDERS), Hakone, Japan, 27 July–1 August 2003.
99. Ballal, D.R.; Lefebvre, A.H. Flame propagation in heterogeneous mixtures of fuel droplets, fuel vapor and air. *Proc. Combust. Inst.* **1981**, *18*, 321–328. [[CrossRef](#)]

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