A MODEL FOR PREDICTING THE HAZARDS FROM LARGE SCALE COMPARTMENT JET FIRES

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SYNOPSIS

Prediction of the hazards associated with jet fires is an important part of the Safety Cases submitted for offshore platforms. The effects of confinement on the behaviour of jet fires has, however, received little attention to date. Recent experimental studies have suggested that in addition to the well-known hazards of high heat fluxes, there exist several potentially new hazards including external flaming, increased smoke, and increased CO levels [1]. These experiments also suggest that the confined fire approaches steady state quite quickly, in the order of 5-10 minutes in most scenarios. This paper presents a physically based zone model to enable the hazards to be estimated at steady state and at a scale representative of offshore modules. For engineering purposes the model has been implemented in a spreadsheet form which is quick to run and easy to use.

Keywords: Compartment Fire, Jet Fire, Fire Hazards, Offshore Safety

INTRODUCTION

When a fire burns inside a compartment, the combustion starts as though the fire were in the open. There is enough air already present to satisfy the stoichiometric requirement for complete combustion and the fire is fuel controlled. The hot combustion products rise to the ceiling and spread out as a ceiling jet. There is a net outflow of material through the compartment openings as the gases heat up and expand. After a short time a hot gas layer of combustion products builds up in the upper part of the compartment which grows and descends as gases continue to flow into it. A relatively well defined interface normally forms between the upper hot layer and cool air below. When this interface descends below an opening there is a sudden outflow of smoke, combustion products or flame. If the compartment openings are small, the fire may not be able to entrain enough air for complete combustion of the fuel inside the compartment. The fire is then said to be ventilation controlled.

The major hazards associated with compartment fires include all those normally associated with open fires. However, additional hazards exist due to the effect of confinement. For personnel these include:

- The extent of external flaming
- Impaired visibility along escape routes through smoke obscuration
- Increased hazard from carbon monoxide (CO)
- Explosion hazard from unburned fuel if the fire extinguishes due to insufficient oxygen

Heat loading onto vessels, pipework and structures can be greater in compartment fires due to the effects of increased soot formation during ventilation controlled conditions, and additional heat radiated from the hot surroundings/walls. The amount of thermal radiation externally radiated from the fire is very sensitive to the amount of soot present. Generally, the greater the degree of ventilation control, the greater is the soot concentration, and the greater is the radiation hazard. However in heavily under ventilated fires the reduced combustion intensity can result in lower plume temperatures and the soot produced can also obscure radiative heat from the flame.

The primary physical parameters affecting compartment fire behaviour include:

1. Geometry and size of compartment, position and size of vents, and degree of thermal insulation present on the boundaries.

2. Type of fire (jet or pool) and fuel involved.

3. Release conditions of fuel - mass flow rate, orifice/nozzle diameter or pool diameter, flow regime, orientation of release for jet fire.

4. The mass transfer from the fire feeding the smoke layer and losses to the outside environment through vents.

   The amount of air present and available for combustion is the controlling factor in determining soot and CO production and flammability of the smoke layer.

5. Radiation and convection heat fluxes to impinged roof, walls and objects.

   The radiation heat flux is closely linked to soot volume fraction and flame chemistry, particularly the acetylene (C_2H_2) and CO reaction kinetics in the smoke layer. The convection heat flux is usually much lower than the radiation except where a jet fire impinges directly. The amount of heat lost through the roof and walls depends on the effectiveness of insulation. This in turn affects internal temperatures and therefore the combustion behaviour.

6. Mitigating circumstances, such as passive fire protection coatings on objects inside the compartment and/or the roof and walls.

7. Water deluge, if present.

The processes discussed above are not independent, but are coupled together such that a change in any one process can have an immediate effect on all others. In a rigorous analysis of compartment fire behaviour, all pertinent physical and chemical processes need to be considered. However, physically based zone models incorporating empirical correlations for the chemistry and soot behaviour can allow good prediction of the fire hazards and their evolution with time. Recently, experimental work has enabled a much simplified approach to be adopted for estimating the steady state behaviour of compartment fires. The formulation presented is suitable for implementation in a computer spreadsheet and gives estimates of global smoke layer properties at steady state, such as depth and temperature, radiative heat and mass transfer losses, and the extent of external flaming.

THE STEADY STATE MODEL

The compartment geometry is approximated by an idealised box structure of length L, height H, and width W, and can have either one or two open vents at one end (see Figures 1 and 2). The vent areas and positions are defined by a height h_v, width w_v, and distance between vent base and floor level, z_v. The compartment itself may be thermally insulated or totally uninsulated. The mass flow rate of fuel released into the compartment, m_f, is an input parameter allowing jet fires to be considered and, if the total mass burning rate is known, pool fires also can be considered. Heat transfer between the compartment surface and outside environment is assumed to be purely radiative and the effects of convection are neglected. This is a reasonable approximation for large scale modules in which the fire velocity near to boundaries can be assumed small (neglecting direct jet impingement), and where wind cooling effects on the outside surfaces are ignored in a worst case analysis. A steady state heat balance is then performed by matching the heat released during ventilation controlled combustion with the total heat loss, Q_{loss}, from the system. The latter is represented by radiative losses from the compartment walls and through the vent areas in addition to heat loss due to mass flow of hot combustion products to the outside.

Two cases are considered:

1. A compartment with one vent having a neutral plane which intersects the vent plane at steady state (Figure 1).

2. A compartment with two vents having a neutral plane which lies in between the vents at steady state (Figure 2).

The governing equations for the steady state mass and heat transfer processes, for each of these cases, are as follows:

Mass Transfer

For both cases it is assumed that the gases in the upper smoke layer are well stirred, which is a good approximation for large compartments. Hot gases/smoke exit the compartment above the neutral plane and cold air enters below it. The depth to which the neutral plane extends from the ceiling or 'height' of the neutral plan is denoted by h_n and is related to the distance above floor level, z_n by:

$$h_n = H - z_n \tag{1}$$

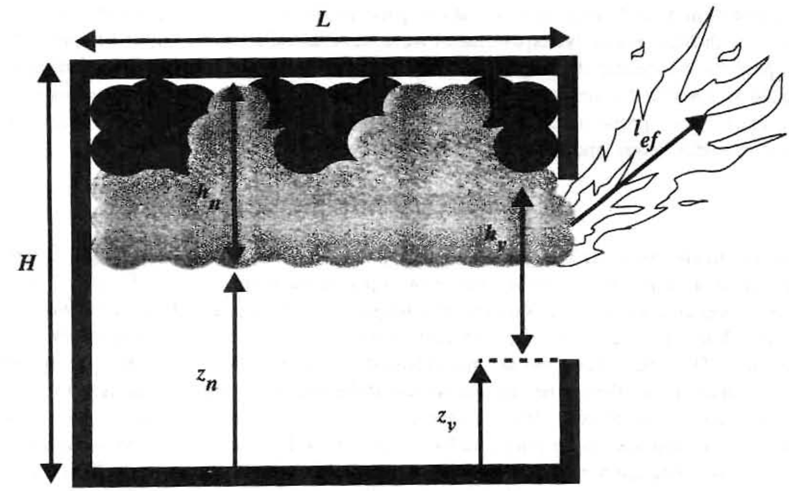


Figure 1 - Single vent compartment (Case 1)
Width of compartment = w
Width of vent = w_v

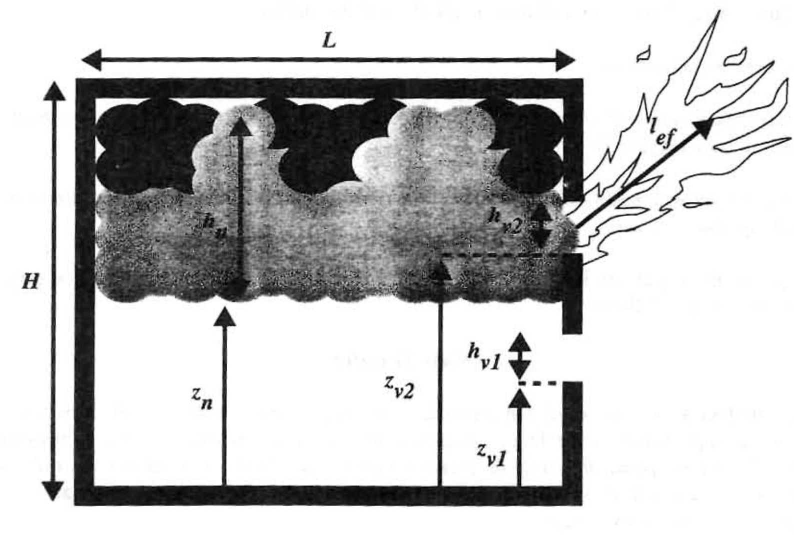


Figure 2 - Double vent compartment (Case 2)
Width of compartment = w
Width of lower vent = w_{v1}
Width of upper vent = w_{v2}

The stoichiometry or equivalence ratio, ϕ, is an important parameter defining the overall state of ventilation and is defined by:

$$\phi = \frac{rm_f}{m_a} \tag{2}$$

where r is the stoichiometric mass ratio for complete combustion and m_a is the mass flow rate of air supplied to the fire through the vents. For $\phi<1$ the fire is fuel controlled or over ventilated, $\phi=1$ corresponds to stoichiometric burning, and for $\phi>1$ the fire is ventilation controlled. Assuming the flow of gases is driven by buoyancy forces and there is no mixing of the inflowing and outflowing gases, then the pressure difference between the inside and outside of the compartment controls the flow. By integrating Bernoulli's equation over the area of vent overlapping with the smoke layer above the neutral plane, the mass flow rate of gas out of the compartment is calculated by:

$$m_o = \frac{2}{3}C_d\left[2g\rho_s(\rho_a-\rho_s)\right]^{1/2}\left[W_v(h_v+z_v-z_n)^{3/2}\right] \tag{3a}$$

for one vent (Case 1) and

$$m_o = \frac{2}{3}C_d\left[2g\rho_s(\rho_a-\rho_s)\right]^{1/2}W_{v2}\left[(h_{v2}+z_{v2}-z_n)^{3/2}-(z_{v2}-z_n)^{3/2}\right] \tag{3b}$$

for two vents (Case 2). The expression in Equation 3b assumes that the neutral plane lies in between the upper and lower vents (Figure 2). The density of the smoke layer/combustion gases is assumed to behave as an ideal gas and can be approximated by:

$$\rho_s = \rho_a\frac{T_a}{T_s} \tag{4}$$

where the temperature of the smoke layer, T_s, is initially set to 1200 K in the spreadsheet and determined more accurately in the heat transfer calculation.

Similarly for the mass flow rate of air into the compartment:

$$m_a = \frac{2}{3}C_d\left[2g\rho_a(\rho_a-\rho_s)\right]^{1/2}\left[W_v(z_n-z_v)^{3/2}\right] \tag{5a}$$

for the single vent case and:

$$m_a = \frac{2}{3}C_d\left[2g\rho_a(\rho_a-\rho_s)\right]^{1/2}W_{v1}\left[(z_n-z_{v1})^{3/2}-(z_n-z_{v1}-h_{v1})^{3/2}\right] \tag{5b}$$

for the double vent case.

At steady state $m_o = m_a+m_f$ and Equations 1, 3, 4 and 5 can be solved to determine the depth of the smoke layer, h_n. Variations in T_s have a small effect on the calculated mass flows and in practice little iteration is required to match the mass flows into and out of the compartment while simultaneously satisfying the steady state heat transfer requirement (next section).

Heat Transfer

Steady state conditions require that the internal heat generated through combustion, ΔE_{rel}, matches the total heat losses from the compartment resulting in no net change in temperature. Neglecting convection, the net heat power generation within an uninsulated compartment is:

$$Q_{net} = \Delta E_{rel} - \left(Q_{rad\to wall} + Q_{rad\to vent} + Q_{rad\to floor} + Q_{mass\,flow}\right) = 0 \tag{6}$$

and for an insulated compartment is:

$$Q_{net} = \Delta E_{rel} - \left(Q_{rad\to vent} + Q_{rad\to floor} + Q_{mass\,flow}\right) = 0 \tag{7}$$

It is assumed that the floor always acts as a heat sink. The heat release rate due to combustion depends on whether the fire is fuel or ventilation controlled. For fuel controlled fires:

$$\Delta E_{rel} = \Delta H_c m_f \qquad (\text{for } \phi \le 1) \tag{8}$$

and for ventilation controlled fires:

$$\Delta E_{rel} = \frac{\Delta H_c m_f}{\phi} \qquad (\text{for } \phi > 1) \tag{9}$$

Considering each term in brackets in Equations 6 and 7:

$$Q_{rad\to wall} = \varepsilon_s \varepsilon_w \sigma \left(\frac{T_s}{1.2}\right)^4 \left[LW + 2h_n(L+W) - W_v(h_v + z_v - z_n)\right] \tag{10a}$$

for the uninsulated single vent compartment and:

$$Q_{rad\to wall} = \varepsilon_s \varepsilon_w \sigma \left(\frac{T_s}{1.2}\right)^4 \left[LW + 2h_n(L+W_v) - W_{v2}h_{v2}\right] \tag{10b}$$

for the uninsulated two vent compartment. In Equations 10a and 10b it is assumed that the compartment walls and ceiling are composed of thin sheets of steel which radiate as grey bodies from both sides, with an emissivity ε_w. Multiple reflections between the smoke layer and walls are neglected which is a reasonable assumption given that the smoke layer is, in reality, a volume emitter with negligible reflectivity. For large compartments it also reasonable to assume the smoke layer radiates as a perfect black body. The heat capacity of the walls is assumed to be small compared with the thermal conductivity such that there are no temperature gradients within the walls or ceiling at steady state. Background radiation contribution from all sources at ambient temperature are small and therefore neglected in this analysis. With these assumptions, the equilibrium temperature of the uninsulated walls and ceiling is approximated simply as $T_{wall}=T_s/1.2$.

Neglecting view factors with the bottom of the hot smoke layer, the radiative heat loss directly through the vent is determined by the spatial overlap between the smoke layer and open vent area. For the single vent:

$$Q_{rad\rightarrow vent} = \varepsilon_s \sigma T_s^4 W_v \left(h_v + z_v - z_n\right) \tag{11a}$$

and for the double vent:

$$Q_{rad\rightarrow vent} = \varepsilon_s \sigma T_s^4 W_{v2} h_{v2} \tag{11b}$$

Heat radiated from the bottom of the smoke layer can be important and if it is assumed that the floor acts as a black body (e.g. coated in soot) and emits radiation at temperature T_{floor}, then for all cases:

$$Q_{rad\rightarrow floor} = \varepsilon_s \sigma \left(T_s^4 - T_{floor}^4\right) LW \tag{12a}$$

In reality the floor in an uninsulated compartment often comprises of high heat capacity material or steel in contact with an effective heat sink. In offshore modules the floor may also be a steel grating and thus mainly transparent to thermal radiation. Therefore it is reasonable to neglect T_{floor} from Equation 12 in these cases, giving:

$$Q_{rad\rightarrow floor} = \varepsilon_s \sigma T_s^4 LW \tag{12b}$$

Equation 12b effectively assumes that the floor is at a temperature of 0K which is a good approximation for uninsulated compartments where back radiation from the floor is small.

For a perfectly insulated compartment, the net radiative loss to the floor at steady state is simply:

$$Q_{rad\rightarrow floor} = 0 \tag{12c}$$

However, a perfectly insulated floor rarely occurs in practice and it can be assumed that the floor always remains at a sufficiently low temperature such that Equation 12a or Equation 12b applies.

The final term in Equations 6 and 7 accounts for the most significant heat loss process in practice, arising from the flow of hot combustion products to the outside through the vent. For the single and double vent cases the appropriate expression is:

$$Q_{mass\,flow} = \left(m_f + m_a\right) c_p \left(T_s - T_a\right) \tag{13}$$

where it is assumed that the fuel is released at ambient temperature, has an isobaric specific heat capacity equivalent to that for air at ambient temperature, and the fuel is either vapour or vapourises immediately upon entering the compartment (without a change in enthalpy). These assumptions are reasonable given that the heat release through combustion dominates over the heat required for liquid fuel vapourisation. The heat capacity of the combustion products/smoke layer is also assumed to be equivalent to that of air.

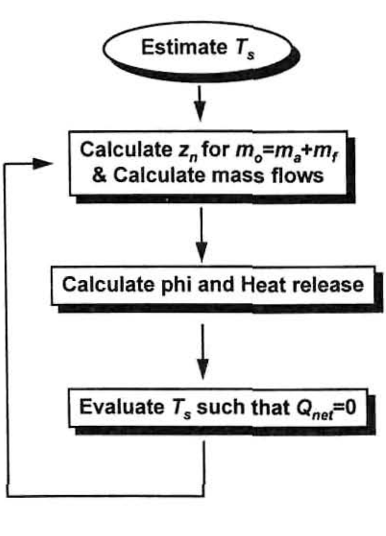
External Flame

Unless the flame is directly oriented toward or is deflected through a vent, fuel controlled compartment fires burn within the confines of the compartment. However for ventilation controlled fires ($\phi>1$) the hot soot and partial combustion products can burn on mixing with the air outside the vent. The vertical length of the external flame can be estimated from the correlation of Yokoi, recommended by Thomas and Law [2]. Also, noting that external flames measured in large scale experiments by Chamberlain [1] rise at an angle of about 60° to the horizontal in still air, the total extent of external flaming simplifies to:

$$l_{ef} = 0.01\left(\frac{T_a}{g}\right)^{1/3}\left[\frac{\Delta H_c m_f\left(1-1/\phi\right)}{c_p \rho_a W_v}\right]^{2/3} \qquad (14)$$

Equations 1-14 have been coded into a spreadsheet program allowing values for T_s, T_{wall}, ϕ, and l_{ef} to be determined from the compartment geometry and mass flow release rate of the fuel. In practice an initial value of T_s is estimated and entered into the spreadsheet to determine a value of z_n which obeys $m_o = m_a + m_f$. A value for ϕ is then calculated allowing ΔE_{rel} to be evaluated. A value for T_s, such that $Q_{net} = 0$, is then obtained from the spreadsheet. This new value for T_s is then used to optimise the values of z_n and m_o (&m_a). A flow chart showing the procedure used is shown in Figure 3. In practice a single iteration is required to satisfy all requirements for steady state conditions.

Figure 3. Flow chart showing calculation procedure used in the computer spreadsheet



RESULTS

The steady state model was tested against programmes of experiments carried out at two scales. One programme was carried out by Shell Research Limited at large scale, in a 135m^3 steel compartment sited at the Norwegian Fire Laboratory, operated by SINTEF at Trondheim. Experimental details can be found in reference [1]. The other programme was recently carried out by Shell Research Limited at small scale, in a 33m^3 steel compartment sited at the Sheffield University Test Site in Buxton. Further details of these experiments will be published in due course. All the large scale (Chamberlain) tests were conducted in an insulated compartment with one wall vent and used gaseous propane fuel. The small scale (Persaud) tests were carried in an uninsulated compartment with either one or two vents in one wall, again using gaseous propane fuel. The height and orientation of the fuel release were different for the tests, but these differences are not treated in the model. A summary of the experimental parameters used in the model is given in Table 1.

Table 1. A summary of Test Parameters used in the Steady State Model

TEST ID	Compartment H(m),W(m), L(m)	Insulated or Uninsulated	Vent h_v(m), W_v(m), z_v(m)	m_f (kg s^{-1})	Fuel Release / Nozzle Orientation
CHAMBERLAIN 1b 1 vent	3.54, 3.91, 9.8	Insulated	2.0, 2.5, 0.0	0.21	sonic propane vertical
CHAMBERLAIN 27 1 vent	3.54, 3.91, 9.8	Insulated	2.0, 2.5, 0.0	0.32	subsonic propane vertical
CHAMBERLAIN 2 1 vent	3.54, 3.91, 9.8	Insulated	2.0, 2.5, 0.0	0.25	subsonic propane vertical
PERSAUD 27 1 vent	2.41, 2.355, 5.925	Uninsulated	0.539, 2.355, 0.0	0.054	subsonic propane vertical
PERSAUD 28 1 vent	2.41, 2.355, 5.925	Uninsulated	0.539, 2.355, 0.0	0.048	subsonic propane horizontal
PERSAUD 17 2 vents	2.41, 2.355, 5.925	Uninsulated	0.539, 2.355, 0.0 0.539, 2.355, 1.871	0.098	subsonic propane vertical
PERSAUD 18 2 vents	2.41, 2.355, 5.925	Uninsulated	0.539, 2.355, 0.0 0.539, 2.355, 1.871	0.138	subsonic propane vertical

Model predictions based on the compartment geometry, vent size and fuel mass flow rate are shown in Table 2 with the experimental values taken at, or near to, steady state.

Table 2. Experimental and predicted values for mean smoke layer temperature, wall temperature, roof temperature, stoichiometry and external flame lengths. Approximate floor temperatures derived from test data are also shown.

1 Steady state model with floor temperature effectively set to 0K using Equation 12b (no back radiation).

2 Steady state model with floor temperature made equal to experimental value at end of test using Equation 12a.

3 Steady state model with floor temperature set to final equilibrium value assuming perfect insulation using Equation 12c.

+ denotes temperature still rising at termination of the test

*** estimated value of "steady" flame length (peak intermittent values approximately ×2 greater)**

 # values for maximum observed flame length

TEST ID	Experiment or Theory	T_s (°C)	T_{wall} (°C)	T_{roof} (°C)	T_{floor} (°C)	ϕ	l_{ef} (m)
CHAMBERLAIN 1b insulated, 1 vent	Experiment	1150	900	1100	830+	1.1	3.6#
	Theory[1]	970	970	970	0 K	0.8	0
	Theory[2]	1100	1100	1100	830	0.8	0
	Theory[3]	1740	1740	1740	1740	0.9	0
CHAMBERLAIN 27 insulated, 1 vent	Experiment	1100	750+	950	580+	1.5	no data
	Theory[1]	1060	1060	1060	0 K	1.3	3.3
	Theory[2]	1100	1100	1100	580	1.3	3.3
CHAMBERLAIN 2 insulated, 1 vent	Experiment	1200	990+	1200	930+	1.5	4.7#
	Theory[1]	1050	1050	1050	0 K	1.0	0
	Theory[2]	1200	1200	1200	930	1.0	0.5
PERSAUD 27 uninsulated, 1 vent	Experiment	400	~200	~200	80+	1.7	1.0*
	Theory[1]	620	470	470	0 K	1.6	1.5
PERSAUD 28 uninsulated, 1 vent	Experiment	430	250	210	80+	1.9	0.5*
	Theory[1]	620	470	470	0 K	1.4	1.1
PERSAUD 17 uninsulated, 2 vents	Experiment	810	750	710	300	0.4	<0.5*
	Theory[1]	790	610	610	0 K	0.5	0
PERSAUD 18 uninsulated, 2 vents	Experiment	850	720	630	<250	0.8	<0.5*
	Theory[1]	930	730	730	0 K	0.8	0

The results show that the steady state model predictions are in good agreement with measurement, given the limitations of the spreadsheet model. Using Equation 12b to describe the heat transfer to the floor, then the smoke temperatures and wall surface temperatures are generally in good agreement with the observed values, although the agreement is better for uninsulated than for insulated compartments. However, if the actual floor temperatures are used in the model (Equation 12a) for the insulated compartments, then the agreement is excellent. This observation gives confidence to the overall methodology employed in the analysis. The zero back radiation assumption does not significantly affect the accuracy of predictions for the uninsulated compartments because the floor remains relatively cool and the T^4 emission is negligible.

In practice offshore modules have minimal thermal insulation. Given that the thermophysical properties of the floor and heat sinks may not be easily quantified, the assumption of back radiation emission from the floor at either zero or ambient temperature remains a reasonable one for all cases, even for insulated compartments.

Values for overall flame stoichiometry are also well predicted for most tests and provides a useful basis for further calculations in more complex models in which the combustion chemistry can be treated to give CO or soot concentration.

Values for external flame lengths show fairly good agreement with experiment. It should be noted that in practice the flames were often attached to the vertical surfaces of the compartment (e.g. PERSAUD Test 27 and Test 28) resulting in longer visible flame lengths than expected, but this intermittent effect is not included in the values given in Table 2. Also, in the large scale tests [1], values quoted for l_{ef} represent the maximum observed extent of external flaming.

Overall the model predictions are remarkably good given the underlying simplicity of the theory. Future work will concentrate on developing a fully time dependent zone model for use in the field, incorporating correlations based on the global combustion chemistry and soot evolution. Such a model will be capable of predicting CO and smoke source terms in addition to the hazards considered here.

SUMMARY

A physically based zone model has been developed enabling the hazards of compartment fires to be estimated at steady state and at a scale representative of offshore modules. For engineering purposes the model has been implemented in a spreadsheet form which is quick to run, transparent and easy to use. For a given scenario the model successfully estimates global smoke layer properties at steady state, such as depth and temperature, radiative heat and mass transfer losses, and the extent of external flaming. The results compare favourably with experiments to a level of precision sufficient for comparison and ranking of offshore hazard scenarios.

REFERENCES

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2. G.A. Chamberlain, 1994, "An experimental study of large-scale compartment fires", Trans I.Chem.E, 72, Part B, 211-219.

NOMENCLATURE

Symbol	*Units*	*Description*
C_d	dimensionless	Discharge coefficient
c_p	J kg^{-1} K^{-1}	Isobaric specific heat capacity of air
ΔE_{rel}	J s^{-1}	Net energy release rate due to combustion inside the compartment
ΔH_c	J kg^{-1}	Heat of combustion of fuel
g	m s^{-2}	Acceleration due to gravity
H	m	Height of compartment
h	m	Height of vent or smoke layer
L	m	Length of compartment
l_{ef}	m	Length of external flame
m	kg s^{-1}	Mass flow rate
Q_{net}	J s^{-1}	Net. power generation inside compartment
$Q_{rad\ wall}$	J s^{-1}	Radiative power transfer to wall or ceiling of compartment
$Q_{rad\ floor}$	J s^{-1}	Radiative power transfer from the bottom of the smoke layer towards the floor
$Q_{rad\ vent}$	J s^{-1}	Radiative power transfer through the vent
$Q_{mass\ flow}$	J s^{-1}	Net. power transfer carried by mass flow through vent
r	dimensionless	Stoichiometric air/fuel mass ratio for complete combustion
T	K	Temperature
W	m	Width of compartment or vent
z	m	Distance above floor level
ε	dimensionless	Emissivity
ϕ	dimensionless	Stoichiometry or equivalence ratio
ρ	kg m^{-3}	Density
σ	J s^{-1} m^{-2} K^{-4}	Stefan-Boltzmann constant (5.67×10^{-8})

Subscript	*Description*
a	Ambient air
ef	External flame
f	Fuel
n	Neutral plane
s	Smoke/gas layer
v, $v1$, $v2$	Single vent, lower vent, upper vent
$wall$	Wall or ceiling