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A STUDY OF THE EFFECT OF THE TANK DIAMETER ON THE THERMAL STRATIFICATION IN LPG TANKS SUBJECTED TO FIRE ENGULFMENT

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ABSTRACT

This paper describes a mathematical model developed to study the behaviour of LPG tanks when subjected to fire conditions. The model consists of a number of field and zone sub-models which are used to simulate the various physical phenomena taking place during the tank engulfment period. To verify the model, predicted results are compared with full scale experimental data obtained by the Health and Safety Executive, U.K. (ref.1). The comparisons indicate that the model can accurately predict the tank pressure and time to first valve opening. The model is used to investigate the effect of the tank diameter on the thermal stratification in the liquid region.

NOMENCLATURE

a	surface absorptivity;
A	area of radiation surface zone;
c	specific heat capacity;
E	emissive power;
F	geometric configuration factor;
h	convective heat transfer coefficient;
J	Jacobian of coordinate transformation; radiosity;
k	thermal conductivity
K	flame absorption coefficient;
P	non-dimensional pressure;
q	heat flux;
Q	heat transfer rate;
r,θ	cylindrical coordinates;
R	radius;
S	source term;
t	time and non-dimensional time
T	temperature and non-dimensional temperature
u,v	non-dimensional velocity components in Cartesian coordinates;
U,V	contravariant velocity components;
x,y	Cartesian coordinates;

Greek letters

α	thermal diffusivity;
α,β,γ	metric coefficients;

β	coefficient of thermal expansion;
Γ	diffusion coefficient;
ε	emissivity of surface;
ν	kinematic viscosity;
θ	angle between the normal vector at a surface element i and the radius vector connecting surfaces i and e;
ξ,η	boundary fitted coordinates;
ρ	density; reflectivity;
σ	Stephan-Boltzman constant;
τ	transmissivity;
Φ	general variable;

Subscripts

b	blackbody;
conv	convection;
e	fire boundary;
f	fire;
g	gas;
i	tank exterior surface zone;
m	weighted mean;
rad	radiation;
t	turbulent;
v	vapour;
w	wall

Dimensionless Numbers

Nu	Nusselt number;
Pr	Prandtl number;
Ra	Rayleigh number;
Re	Reynolds number.

INTRODUCTION

Computer models capable of predicting the conditions in Liquefied Petroleum Gas (LPG) tanks engulfed by fire are a necessary tool to enable an understanding of the physical phenomena taking place in the tank during such incidents. This knowledge can be used to design and develop safer procedures for the storage and transportation of liquefied gases. The processes during the fire engulfment period are complicated and interactive. Some of these processes, such as radiative and convective heat transfer from the fire to the tank, radiation from the vapour-wetted wall to the liquid interface and mass discharge rate can be accurately determined using zone models. The free convective flows and heat transfer in the liquid and the vapour region during the heat up period, (up to the first valve opening), however, cannot be simulated accurately by this modelling approach. The main reason is that these processes are not fully understood and the various zones and their interactions cannot be accurately defined, especially when dealing with partial fire engulfment. Despite these limitations a number of zone models, which rely on information extracted from a limited number of small and full scale experiments have been developed, (ref. 2-6). As the case simulated by a zone model deviates from the experimental scenario, this information may not be valid and its use may result in incorrect predictions.

Free convection can be predicted using field models, which solve the fundamental equations for conservation of continuity, momentum and energy to predict the convective flows and heat transfer in the tank. Field models, however have their own limitations as they require extensive computer time and they also rely on empirical information, especially in the application of the thermal and hydrodynamic boundary conditions at the inner tank surface. This is because the heat transfer process from the liquid wetted wall to the liquid changes with time and location.

During the initial period of fire engulfment, heat is transferred by natural convection, but as the applied heat flux increases, sub-cooled boiling takes place which alters both the thermal and the hydrodynamic boundary conditions. For this reason it is not feasible, at present, to develop a field model which can simulate the total period of fire engulfment. Field models, however, can be used to predict the conditions in the tank during the early stages of fire engulfment and to provide information, which can be fed to zone models to continue the simulation for the remaining fire engulfment period.

THE MATHEMATICAL MODEL

Modelling approach and assumptions.

The developed numerical model makes use of field and zone modelling approaches to simulate the LPG tank response to fire engulfment. Zone modelling is used to determine the fire heat flux applied at the tank exterior surface and the radiation heat transfer from the vapour wetted wall to the liquid interface. The heat transfer through the tank wall and the free convection in the vapour and the liquid regions are simulated using the field modelling approach.

Due to the complexity of the problem the following simplifying assumptions have been made:

1. The problem is considered to be two dimensional.
2. The Boussinesq approximation for the free convection governing equations is assumed to be valid.
3. Constant effective viscosity is assumed throughout the solution domain and the turbulent Prandtl number is taken as unity.
4. Boiling at the tank walls is not considered.
5. The interface is assumed to be waveless and static.
6. Fire size and fire properties are uniform.

Due to these assumptions the model can only be used to predict the tank response during the initial period of fire engulfment, up to the first valve opening since, during this period, boiling is somewhat suppressed by pressurization.

The model consists of four sub-models: the fire model, the wall conduction model, the vapour space radiation model and the free convection model. These sub-models are discussed in the subsequent sections.

Fire model

(i) Radiation heat transfer. Radiation heat transfer from the flames to the tank wall is determined following the approach used by Birk and Oosthuizen (ref. 7), in which the fire is assumed to occupy a rectangular space surrounding the cylindrical tank. The tank exterior surface is divided into a number of surface zones while the imaginary walls enclosing the fire are considered as a single surface zone. The radiative properties of the fire are assumed to be uniform, and its size does not change with time. The configuration factors, F_{ie} between each surface zone on the tank exterior surface, A_i, and the walls enclosing the fire, A_e, are computed by numerically integrating the relation,

$$F_{ie} = \int_{A_i} \int_{A_e} \frac{\cos\theta_i \cos\theta_e \, e^{-\int K dR}}{\pi R^2} dA_e \, dA_i \qquad (1)$$

where R = radius vector joining the centers of the surface zones i and e; K = mean absorption coefficient of the flames, and q = angle between the normal vector at a surface element and R.

Using the configuration factors and realizing that the zones on the cylinder can only "see" part of the enclosure wall and part of the surrounding gas, the energy balance equations are written as:

$$Q_i = A_i \varepsilon_i E_i - \alpha_i (F_{ei} E_e + F_{gi} E_g) \qquad (2)$$

where Q_i= heat transfer rate; E_i = emissive power; α_i = surface absorptivity, and ε_i = surface emissivity. The subscripts g and e stand for gas and fire enclosure respectively.

Equation (2) is solved directly to give the radiation heat flux at the various surface zones on the tank surface.

(ii) Convection heat transfer. The convection heat transfer from the hot combustion gases to the tank wall is given by

$$q_{conv} = h (T_w - T_f) \qquad (3)$$

where q_{conv} = convection heat transfer; h = heat transfer coefficient; T_w = temperature at the exterior surface of the tank; and T_f = fire temperature.

The convective heat transfer coefficient, h, is calculated based on heat transfer measurements for forced convection to cylinders in cross flow of air, (ref. 8). The results for Reynolds numbers less than 70800 are given in the form of the empirical relation:

$$Nu = c \, Re^m \, Pr^n \qquad (4)$$

where Nu = Nusselt number; Re = Reynolds number; Pr = Prandtl number; and c = 0.26; m = 0.6; and n = 0.36.

For Reynolds numbers greater than 70800, the heat transfer coefficient is given as a function of circumferential location. The Reynolds number is calculated based on the tank exterior diameter and a gas velocity determined from measured fuel evaporation rates for large diameter pool fires (ref. 9). This velocity is calculated immediately above the fuel surface.

Wall conduction model

Heat transfer by conduction through the tank wall is calculated by numerically solving the unsteady two dimensional conduction equation in cylindrical coordinates,

$$\rho c\frac{\partial T}{\partial t} = \frac{1}{r}\frac{\partial}{\partial r}\left(rk\frac{\partial T}{\partial r}\right) + \frac{1}{r}\frac{\partial}{\partial \theta}\left(\frac{k}{r}\frac{\partial T}{\partial \theta}\right) \tag{5}$$

where ρ = density of shell material; c = specific heat of shell; k = thermal conductivity of shell; r,θ = cylindrical coordinates; t = time; and T = temperature.

The solution of equation (5) requires the specification of boundary conditions at the exterior and interior tank surface. At the exterior surface, the total fire heat flux, as calculated by the fire model, is used and at the interior surface, the heat leaving the wall and entering the fluid is used. This is calculated by conducting a heat balance at the inside tank surface using the fluid and wall temperature distribution from the previous time step .

Vapour space radiation model

The vapour space radiation model computes the radiation heat transfer from the hot dry wall to the liquid interface. The net radiation method for an enclosure filled with an isothermal gas is used (ref. 10), with the gas temperature taken as the weighted mean temperature of the vapour. The liquid interface and the dry wall interior surface are divided into a number of surface zones each having a uniform temperature. Configuration factors are calculated for each surface zone and the energy balance equations, written for each zone, are solved yielding the radiation heat flux along the enclosure surface, ie; vapour wetted wall and liquid interface. An energy balance on the entire enclosure (vapour space) is also made which yields the heat absorbed by the vapour.

Free convection model

The flow and temperature fields in the tank (liquid and vapour space) are obtained from the solution of the equations governing free convection, continuity, momentum and energy. These equations are simplified with the assumptions stated in the introduction and normalized using the liquid properties and tank diameter as reference parameters. They are then transformed from the (x,y) coordinate system to a boundary fitted (ξ,η) coordinate system following the method of Maliska and Raithby (ref. 11). The resulting equations are:

$$\frac{\partial U}{\partial \xi} + \frac{\partial V}{\partial \eta} = 0 \tag{6}$$

$$J\frac{\partial\Phi}{\partial t}+\frac{\partial(U\Phi)}{\partial\xi}+\frac{\partial(V\Phi)}{\partial\eta} =$$

$$\Gamma\left[\frac{\partial}{\partial\xi}\left(\frac{\dot{\alpha}}{J}\frac{\partial\Phi}{\partial\xi}-\frac{\dot{\beta}}{J}\frac{\partial\Phi}{\partial\eta}\right)+\frac{\partial}{\partial\eta}\left(\frac{\dot{\gamma}}{J}\frac{\partial\Phi}{\partial\eta}-\frac{\beta}{J}\frac{\partial\Phi}{\partial\xi}\right)\right]+S(\xi,\eta) \qquad (7)$$

where $\dot{\alpha} = x_\eta^2 + y_\eta^2$; $\dot{\beta} = x_\xi x_\eta + y_\xi y_\eta$; $\dot{\gamma} = x_\xi^2 + y_\xi^2$; $J = x_\xi y_\eta - x_\eta y_\xi$; $U = uy_\eta - vx_\eta$;

$V = vx_\xi - uy_\xi$; Φ; general variable (see Table 1); Γ = diffusion coefficient (see Table 1); $S(\xi,\eta)$ = source term (see Table 1); u,v = cartesian veloctity components; ν = kinematic viscosity; α = thermal diffusivity.

The turbulent viscosity, ν_t, and thermal diffussivity, α_t are obtained using the prescribed eddy viscosity model developed by Thompson et al. (ref. 12).

TABLE 1
Symbols used in governing equations.

Equation	Φ	Γ	$S(\xi,\eta)$
ξ-momentum	u	$Pr\nu_t/\nu$	$-y_\eta\frac{\partial P}{\partial\xi}+y_\xi\frac{\partial P}{\partial\eta}$
η-momentum	v	$Pr\nu_t/\nu$	$-x_\xi\frac{\partial P}{\partial\eta}+x_\eta\frac{\partial P}{\partial\xi}+JRa\,(T-T_m)$
energy	T	α_t/α	$Q_{rad}(\xi,\eta)$

The governing equations are discretized using the control volume approach and solved following the SIMPLEC procedure (ref. 13). Details of the solution method are contained in (ref. 14).

Major steps of the solution method

The four sub-models discussed in this section are combined to form the complete numerical model. The solution of each of the sub-models depends on the solution of the other sub-models. This dependency is introduced either in the form of boundary conditions or as a heat source in the governing equations. The major steps of the solution procedure are:

1. Read and initialize all variables.
2. Calculate the heat flux distribution at the tank exterior surface due to the fire using the fire model.
3. Solve the conduction equation for the tank wall to obtain the temperature distribution of the shell.
4. Calculate the heat flux from the wall to the tank contents by convection and radiation.
5. Employ the SIMPLEC method to solve free convection in the vapour and liquid regions.
6. Calculate the tank pressure which is the saturation pressure corresponding to the mean interface temperature.

7. Proceed to the next time step, return to 2, and continue until the tank pressure has reached the set pressure for the valve to open.

RESULTS AND DISCUSSION

Model testing.

Each of the sub-models, and in particular the free convection model, has been tested extensively by comparing predictions with experimental data or numerical benchmark solutions found in the literature. The results of these validation experiments are found in (ref. 14).

The ability of the full model to predict the behaviour of LPG tanks under fire conditions has been examined by employing the model to simulate a test performed by the Health and Safety Executive (HSE) of U.K. (ref. 1). Other experimental data are also available (ref. 15,16) however, the HSE test have been selected as they are the most complete in terms of the recorded information. The tank considered has a diameter of 1.694 m and is 80% filled with liquid propane. The fire temperature is extracted from the experimental data; however an accurate representation cannot be easily made as the temperature varies with location and time. Variations of up to 200 K can be seen in the data. From the video record of the test it is apparent that a strong wind was present during the test period, which allowed the top of the tank to be seen through the flames. For this simulation, the fire shape is assumed to occupy the space between a wall which surrounds the tank; thus, during the numerical simulation, the fire is considered to fully engulf the tank.

The tank pressure predicted by the model was compared with the pressure recorded during the experiment, (Figure 1), and they are in excellent agreement. The numerical model

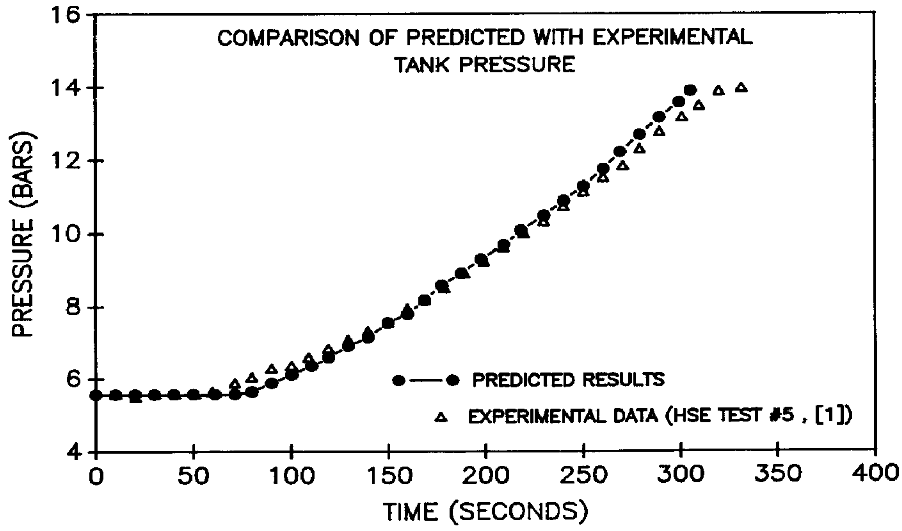
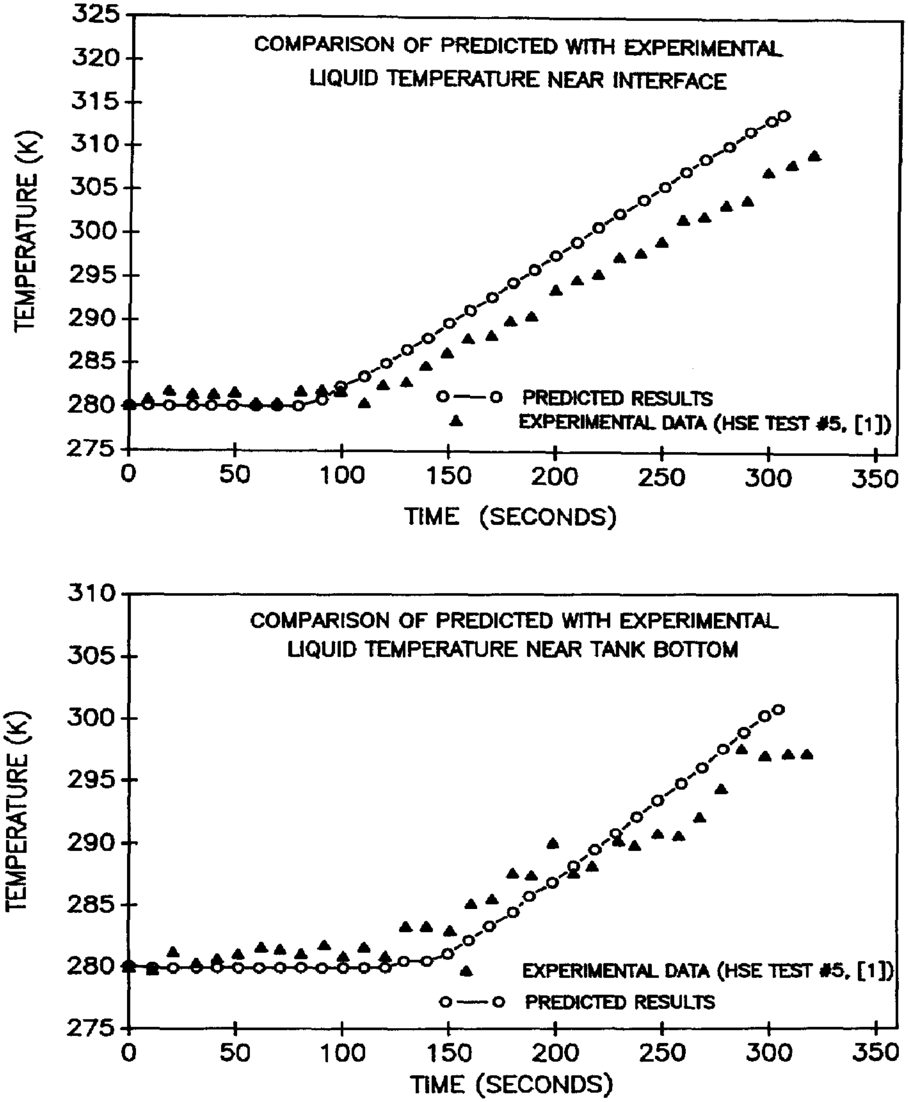


Fig. 1. Comparison of experimental and predicted tank pressure.



(a)

(b)

Fig. 2. Comparison of experimental tank temperature with predicted solution; (a) top of liquid, (b) bottom of tank.

predicts that the pressure relief valve opens in 305 seconds. During the test, the valve opened at 312 seconds.

The predicted liquid temperature is also in good agreement with the experimental data, especially during the initial 180 seconds of the test, (Figure 2). Figure 2a compares the temperature at the top of the liquid and Figure 2b at the bottom. Towards the end of the simulation, the model seems to overpredict the liquid temperature at the top of the liquid region.

The predicted temperatures of the vapour and vapour wetted wall were also found to be higher than the experimental ones. The main reason for this may be due to the fact that during the experimental test, the vapour-wetted wall was not engulfed completely by the flames due to wind effects. Hence, the temperature of this segment remained low due to radiation heat transfer to the ambient air. Another factor influencing the vapour and liquid temperature variations between the experimental and predicted data is the quantity of impurities in the experimental fluid, which is not accounted for in the numerical model.

Effect of tank diameter on tank response.

Simulations have been performed using the numerical model in order to study the effect of the tank size on the tank response when subjected to fire engulfment. For all simulations, the filling level is 80% and the maximum fire temperature 1100 K, which is reached within the first 10 seconds of the simulation. The tank diameter varies in a typical range of values from 0.6 m to 2.6 m.

Since the flow field in the tank, for all simulations, follows a similar pattern, results are presented only for the 1.0 m tank. Figure 3, which depicts the flow streamlines in the tank at 50 and 100 seconds, indicates that two counter rotating eddies occur in the liquid region. The warmer liquid adjacent to the tank wall creates a boundary layer flow moving upward towards the interface and occupies the upper liquid region forcing the colder fluid to move downward and diffuse into the bulk liquid. Two recirculating eddies are also generated in the vapour space. This fluid motion causes some mixing to occur, however, as shown in Figure 4, which depicts the isothermal lines in the tank, the fluid is thermally stratified. The temperature difference

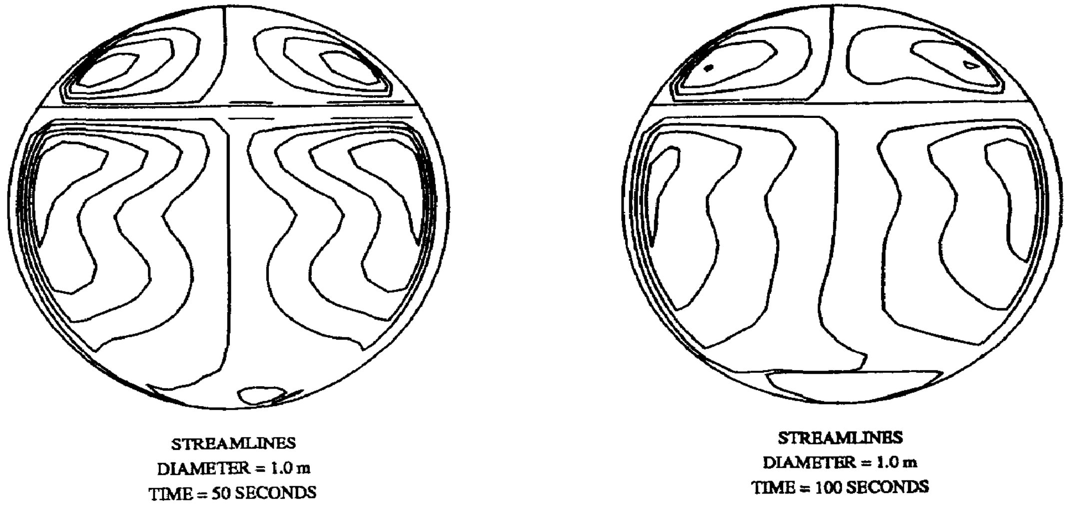


Fig. 3. Streamlines for the 1.0 m diameter tank at 50 and 100 seconds.

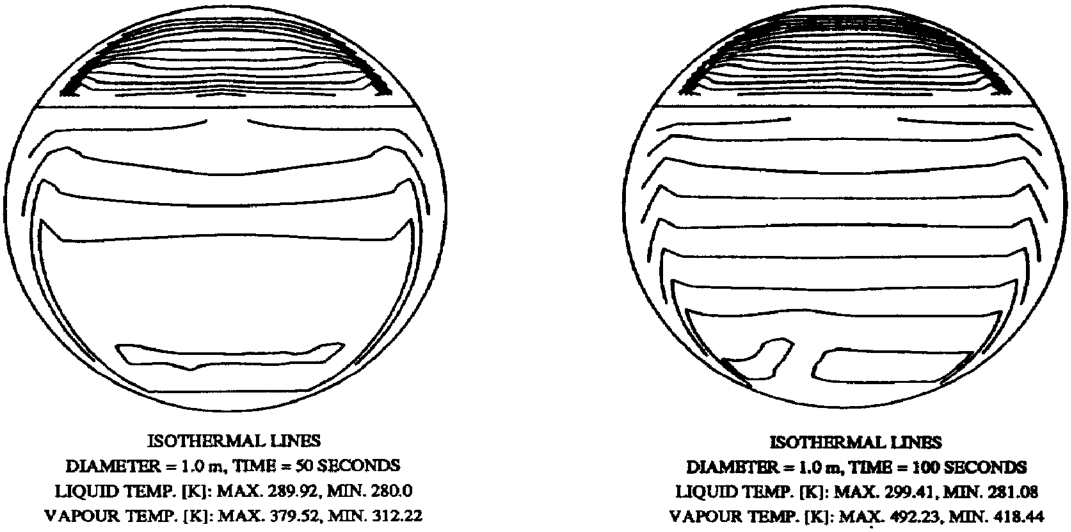


Fig. 4. Isothermal lines for the 1.0 m diameter tank at 50 and 100 seconds.

between isotherms in the liquid region is 1.0 K, while in the vapour space the difference is 10 K. The fluid temperature is uniform in the horizontal direction and decreases from top to bottom with the coldest region found just above the tank bottom. The liquid is warmer near the bottom of the tank due to direct heat input from below.

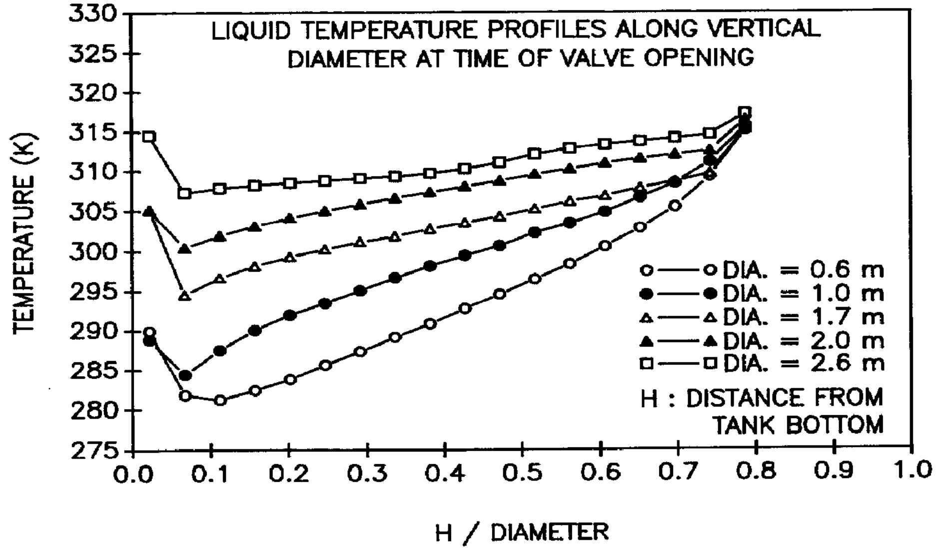


Fig. 5. Temperature profiles in the liquid region along the vertical diameter for the tank sizes simulated.

The temperature profiles along the vertical diameter in the liquid region, at the time of valve opening, are illustrated in Figure 5 for all cases simulated. As shown the temperature gradient in the vertical direction, which indicates the strength of the stratification, decreases as the tank size increases. This decrease can be attributed to a better mixing of the liquid when the tank diameter increases.

The tank pressure history for the tank sizes considered is shown in Figure 6. The pressure in the smaller tank rises rapidly and the valve opens in 95 seconds. As the tank diameter increases the rate of pressure rise decreases causing an increase in the time to valve opening. The valve opens at 180, 310, 378, and 464 seconds for the 1.0, 1.7, 2.0, and 2.6 m tanks respectively.

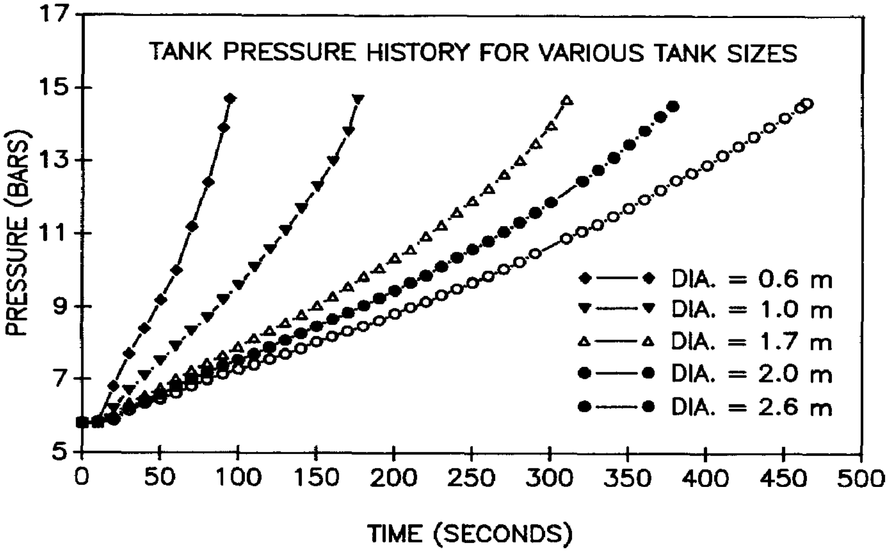


Fig. 6. Tank pressure history for the cases simulated.

Another important parameter, which affects the response of the tank to heating conditions, is the tank wall temperature. As the wall temperature increases, the strength of the wall decreases. This might lead to tank eruption and explosion due to the rising tank pressure. Figure 7, which shows the maximum wall temperature for the cases simulated, indicates that the rate of temperature rise decreases with an increase in the tank diameter. The difference in this rate, however, is negligible for the 1.7, 2.0 and 2.6 m tanks. At the time of valve opening, the maximum tank wall temperature is greatest for the 2.6 m tank and decreases as the tank size decreases; a result expected due the increase in the time required for the valve to open.

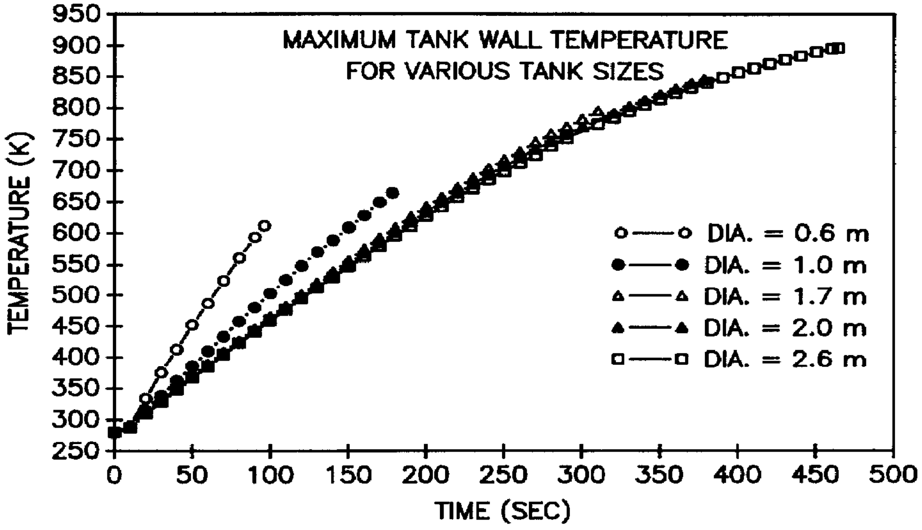


Fig. 7. Maximum tank wall temperature for the cases simulated.

CONCLUDING REMARKS

A numerical model has been developed to study the response of LPG tanks to fire conditions. The model has been used to simulate an experimental test in order to compare the predictions with the test data. The comparisons indicated that despite the difficulties in accurately modelling the engulfing fire, the model can accurately predict the time to valve opening. The model, however, overpredicts the vapour and liquid temperatures, which may be a result of variations in the fire exposure during the test due to wind effects.

The model has been used to investigate the effect of the tank diameter on the tank response and on the thermal stratification in the liquid region. The predicted results lead to the following conclusions:

(i) the time to first valve opening increases as the tank diameter increases;
(ii) thermal stratification in the liquid region decreases as the tank size increases;
(iii) the maximum tank wall temperature at the time of valve opening increases as the diameter increases.

The cases simulated are for a symmetric fire heat flux and the results presented cannot be assumed to represent scenarios where the fire heat flux is not symmetric. Heat input unsymmetries will cause completely different free convective flows in the liquid region and hence a different temperature distribution, which will affect the pressure rise. These conditions are now being investigated and the results will be presented in future publications.

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