

THERMAL RESPONSE ANALYSIS OF LPG TANKS EXPOSED TO FIRE

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ABSTRACT

The present paper reports on the development of a numerical model for LPG tanks engulfed in flames. A detailed description of the required phenomenological relations and assumptions is also given. The model uses a lumped approach for the lading with two major control volumes linked by the evaporation of a stratified interface. Comparison between the model's predictions and field test data show close agreement.

INTRODUCTION

The potential destructive power of uncontrolled releases of LPG can lead to substantial damage and fatalities (refs. 1,2). Typical situations of concern are LPG vessels which are engulfed in fire or exposed to solar heating for extended periods of time and not properly protected with adequate pressure relief.

In the present study, only the scenario involving a horizontal cylindrical vessel engulfed in flames will be considered. The origin of the fire may be due to a collision of a transport vessel resulting in a leakage followed by spilling and ignition of the LPG. Due to the intense radiation load of the inevitably resulting flames, the pressure inside the vessel will increase rapidly and the relief valve will open when the corresponding set pressure is reached. Material discharged through the valve may be either a single- or a two-phase fluid, depending on the liquid level inside the tank at the time of venting. This exiting material may also ignite and form a torch which may radiate to the

tank. The valve may close and open several times (cycle) and eventually remain open if the heat input to the tank is sufficient to maintain the level of vapourization necessary. If the fire is not extinguished and/or the vessel is not properly protected, the shell temperatures, especially those in the vapour space, will increase to levels which will weaken the steel. A rupture may then take place and a BLEVE (Boiling Liquid Expanding Vapour Explosion) may result during which fragments of the vessel can be propelled large distances.

Numerous studies of the accident scenario described above have been reported. These can be categorized as either a) experimental (refs. 1-6) or b) numerical and analytical (refs. 7-10). Among the experimental studies, refs. 3, 4 and 5 have been chosen to compare the results of the present model. Each of these represent field tests on different tank scales, filling levels, and protection.

A computer model, validated against all available and reliable test data, is very desirable due to prohibitive costs of full scale tests. The generation of a code for use in design or incident evaluation is the objective of this work. In developing such a code, one must appreciate the exact thermohydraulics of the lading. Any other approach would be heavily dependent on scale and would probably only give dependable results under very specific circumstances.

The present approach is based upon visual observations of the contents of a laboratory vessel (ref. 6) which permits a clear view of the total flow field. In addition, selected field test data are also utilized. From this data, it is apparent that the liquid is almost always thermally

stratified. The present Pressure Liquefied Gas Simulator (PLGS-I) code is unique in this sense because it is the only program that considers this phenomenon in both the liquid and the vapour. Other codes dealing with the same problem have been developed but they do not take thermal stratification into account (refs. 7,8).

PHYSICAL PHENOMENA

When a tank containing any pressure liquefied gas is subjected to external heating, the initial mode of heat transfer is pure conduction. It only takes, however, a few seconds for the fluid particles near the wall to heat and begin rising along the walls of the vessel. Upon reaching the liquid surface, these particles spread towards the centre and "fall" back into the cooler regions of the fluid body as they lose "heat"; they then re-enter the boundary layer to repeat the above process. This recirculating natural convective flow eventually establishes a vertical temperature gradient inside the tank in both the liquid and vapour. The fluid is said to be thermally stratified. The magnitude of the temperature gradient is a function of several parameters but for a given fluid and heat flux, experience indicates that the depth of lading is of primary importance; the lower the filling level, the less severe the stratification.

Another interesting observation from the laboratory vessel is the existence of vertically-upwards fluid motion near the bottom of the cylinder. In this region, fluid particles rise directly upwards and mix with the bulk of the fluid (ref. 9) instead of attaching to the wall to form a boundary layer. This mechanism is important since it provides direct heating

to the central core regions and also retards the initiation of the boundary layer (Fig. 1).

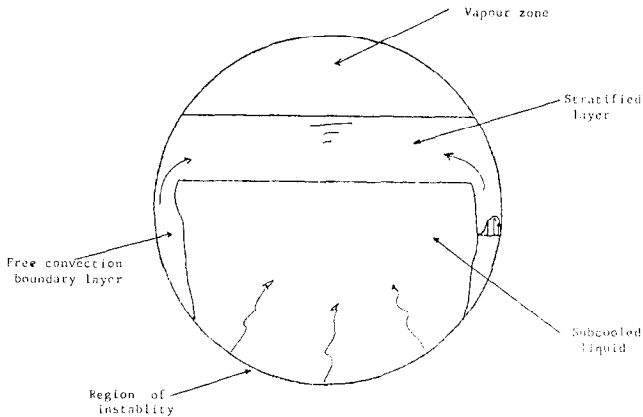


Fig. 1. Physical model for thermohydraulic behavior.

As heating progresses, the liquid layers adjacent to the walls attain and exceed the saturation temperature corresponding to the system pressure. This leads to heterogeneous vapour nucleation at the wall. These bubbles either follow the wall, forming a "bubble boundary layer" or collapse at the outer edges of the boundary layer where the amount of liquid superheat is not sufficient to sustain bubble growth. The latter mechanism is another means of transferring heat to the sub-cooled bulk fluid.

Continuing heat addition to the vessel eventually causes the pressure relief valve to open and the discharge of single- or two-phase material depending upon the fill level and valve orientation. From the tests of ref. 5, the opening of the relief valve clearly demonstrates those parts of the liquid that are superheated or subcooled by virtue of the amount of local void generation. In the upper fluid regions (stratified

liquid layers), boundary layer along the walls, and also a restricted zone near the bottom of the tank large void fractions develop with depressurization. The central fluid body has a void fraction very close to zero because for some time it is subcooled even if the valve is open and the pressure reducing. Naturally, as time progresses, continuous mixing of the fluid results in more uniform temperatures throughout the liquid; eventual valve openings take place in the presence of bulk or homogeneous boiling. This is usually the stage where the valve is most likely to stay open if the external heat input is sufficient.

The gas in the vapour space is superheated and severely stratified. If the valve is open and there is depressurization, the amount of superheat decreases and saturation conditions are approached. Higher levels of superheat are reached when the valve remains open and pressurization takes place. This is a critical stage since wall temperatures escalate and rupture is likely to occur due to weakening of the steel shell.

In what follows, idealizations of the above observations will be presented. These form the basis of the computer code used to generate the simulations presented.

MATHEMATICAL IDEALIZATIONS

The main assumptions of the present model are: a) a two-dimensional analysis and b) symmetry with respect to a vertical plane passing through the centre of the vessel.

Clearly, these assumptions are valid as long as the external fire is uniform and the tank is not tilted or rolled. With these assumptions in mind, the modeling equations are

summarized. For further details, the reader is referred to ref. 10.

Fire Environment

The fire is assumed to be fully engulfing and uniform. The mean fire temperature develops in an exponential manner, assuming an asymptotic value as evidenced in all the field tests. The time constant of this exponential variation and the asymptotic value of the developed fire are inputs to the program.

The heat transfer from the fire to the tank takes place in two modes:

Radiation. The tank and the fire are modeled as two gray bodies for which the equation of heat transfer is:

$$q_R = \sigma(T_f^4 - T_s^4) / [1/\epsilon_f + 1/\epsilon_s - 1]$$

where it has been assumed that 1) the flame is opaque and 2)

$$F_{f-s} = 1 \text{ and } A_f = A_s$$

Convection. This is computed based on the work of ref. 11:

$$q_C = h_C(T_f - T_s)$$

where $h_C = a_1(a_2/D)^{a_3}$,

where a_1 , a_2 , and a_3 are correlating coefficients.

The total heat transfer from the flames to the vessel is therefore

$$q = q_R + q_C$$

Tank insulation and shell

A detailed analysis of insulation and shell temperatures is necessary since the heat input into the tank is determined by the temperature gradient in the radial direction. The layers of insulation and the shell may, therefore, be

subdivided into as many as 24 circumferential sections containing up to eight radial elements. The governing heat diffusion equation has been discretized using central differences for the space derivatives with an explicit marching scheme. This method, although being very restrictive in terms of the allowable time step, is amenable to a Runge-Kutta algorithm.

Vapour space

The two essential modes of heat transfer in the vapour space are convection and radiation. The convection component is based on Newton's cooling law ($q = h_v(T_w - T_v)$) where h_v is a heat transfer coefficient based upon the following correlations (ref. 12) for a heated horizontal plate :

$$h_v = 0.13[\beta k_v^2 \rho_v^2 C_p g (T_w - T_v) / \mu_v]^{1/3}$$

for the case where the vapour is not condensing due to depressurization, and

$$h_v = 0.555\{g \rho_l (\rho_l - \rho_v) k_l^3 h_{fg} / [2 \mu_l (T_w - T_v) L]\}^{1/4}$$

for the case where vapour condenses on the wall due to pressurization. The radiation heat transfer is computed with the assumption that the properties of the gas and the liquid interface are uniform. The model is:

$$\text{Heat lost by the wall} = E_{bw} A_w - G_w A_w$$

$$\text{Heat gained by liquid interface} = G_l A_l - E_{bl} A_l$$

where

$$G_w = A_g F_{g-w} \epsilon_g (T_g) E_{bg} + A_l F_{l-w} \tau_g (T_l) E_{bl}$$

$$G_l = A_g F_{g-l} \epsilon_g (T_g) E_{bg} + A_w F_{w-l} \tau_g (T_w) E_{bw}$$

$$\text{and } F_{w-1} = d / (\Phi R), \quad F_{w-w} = 1 - F_{w-1}$$

Since $A_1 F_{1-w} = A_w F_{w-1}$ we have

$$F_{w-1} = A_1/A_w \cdot F_{1-w} = d/(R\phi). \quad F_{1-w} = d/(R\phi); \quad (F_{1-w} = 1)$$

Therefore $F_{w-w} = 1 - d/(R\phi)$ where $d = 2R\sin(\phi/2)$

$$= 1 - 2\sin(\phi/2)/\phi$$

We also have,

Heat gained by vapour = heat lost by the wall-heat gained
by the liquid interface.

Liquid space

The liquid space is subdivided into four different zones based on the experimental observations described earlier. These zones (Fig. 1) are a) stratified liquid, b) boundary layer region, c) bulk liquid and d) bottom regions of the tank.

The characteristics of the flows and the heat transfer depend very much on the temperature differences between the tank walls and the adjacent liquid. The two major modes of heat transfer in the liquid space are natural convection and nucleate boiling. Film boiling is a phenomenon which may occur due to sudden wetting of dry tank walls during venting and depressurization. It is unlikely to occur at a location which is already wetted by the liquid in this situation since the critical heat flux for propane exceeds 400 kw/m^2 .

Heat transfer in liquid space

Natural convection. Depending on whether the surface can be considered vertical or horizontal, the following correlations are used (ref. 13).:

- a) Vertical flat plate: $Nu=0.59(Ra)^{0.25}$ (laminar)
 $Nu=0.10(Ra)^{0.33}$ (turbulent)
- b) Horizontal flat plate facing upwards:
 $Nu=0.54(Ra)^{0.25}$ (laminar)
 $Nu=0.15(Ra)^{0.33}$ (turbulent)
- c) Horizontal flat plate facing downwards:
 $Nu=0.27(Ra)^{0.25}$ (laminar)

where

$$Nu = h_{nc} L_C / k, \quad Ra = g L_C^3 \rho \beta (T_w - T_l) / (\mu \alpha)$$

L_C , ratio of area to perimeter
and h_{nc} is the natural convective
heat transfer coefficient.

Nucleate boiling. The onset of nucleation is determined by monitoring the criterion:

$$T_r - T_{sat} = 2v_{gsat} T_{sat} \sigma_l / (h_{fg} r)$$

where

r is the radius of an active nucleation site and

$$T_r = (T_w - T_\infty) [1 - r h_{nc} / k_l] + T_\infty$$

Once nucleate boiling commences, the heat transfer coefficient is calculated from a correlation due to Rohsenow (ref. 14):

$$h_{NB} = [(\Delta T - T_{sat} + T_l) C_p / C_{sf}]^3 \mu_l / (\Delta T h_{fg}^2)^* \\
* ((\rho_l - \rho_v) / \sigma_l)^{1/2} / Pr^{5.4}$$

where $\Delta T = T_w - T_l$ and C_{sf} is given by:

$$C_{sf} = [\theta_{tr}^3 / h_{nc} \Delta T_{tr} C_{pl}^3 \mu_l / h_{fg}^2 ((\rho_l - \rho_v) / \sigma_l)^{1/2} / Pr^{5.4}]^{1/3}$$

with

$$\Delta T_{tr} = \theta_{tr} + T_{sat} - T_l$$

Transition to film boiling. The criterion for transition is met when the local heat flux reaches the value of critical heat flux which is given by:

$$q_{CHF} = (3.14/24)h_{fg}(\rho_l\rho_v)^{1/2}[\sigma_l g(\rho_l - \rho_v)/(\rho_l + \rho_v)]^{1/4}$$

During transition, the heat transfer coefficient, h_{tr} , is computed from

$$h_{tr} = [q_{CHF}(\Delta T_{LP} - \Delta T) + q_{LP}(\Delta T - \Delta T_{CHF})]/[\Delta T(\Delta T_{LP} - \Delta T_{CHF})]$$

where

$$\Delta T_{CHF} = \Theta_{CHF} + T_{sat} - T_{\infty}$$

with

$$\Theta_{CHF} = [q_{CHF} C_{sf}^3 h_{fg}^2 / (C_{pl}^3 \mu_l) \{\sigma_l / (\rho_l - \rho_v)\}]^{1/2} Pr^{5.4}]^{1/3}$$

and ΔT_{LP} is the Leidenfrost point.

Film boiling. When the temperature differential exceeds ΔT_{LP} , film boiling begins. A correlation due to Ellison-Bromley (refs. 15,16) is used to compute the heat transfer coefficient in this region:

$$h_{fb} = 0.714 \{k_v^3 g \rho_v \rho_l (h_v - h_l) / [L_c \mu_v (T_w - T_{sat})]\}^{1/4}$$

where L_c is the characteristic length.

Fluid flow

Boundary layer flows. The boundary layer flows inside the vessel are computed using the closed form analytical expressions given by ref. 9. The rate of flow into the stratified layer is:

$$B_{St} = 0.558 \Phi_2^{8/7} (\sin \Phi_2)^{2/7} \mu_l (Pr_l / Gr_R^*)^{-2/7} \quad (\text{turbulent})$$

$$B_{St} = 3.22 \Phi_2^{4/5} (\sin \Phi_2)^{1/5} \mu_l [(1/Pr_l^2 + 1.25/Pr_l) / Gr_R^*]^{-1/5} / Pr_l$$

(laminar)

$$\text{where } Gr_R^* = \rho_l^2 g \beta q R^4 / (k_l \mu_l^2)$$

and q is the value of the heat flux at the wall.

Since in the bottom regions of the tank the flow is not of the boundary layer type, the angle Φ does not necessarily start from zero but from a finite value, Φ_0 . This was determined in ref. 9 through a simple boundary layer stability analysis and is given by:

$$\begin{aligned} \Phi_0 \tan \Phi_0 &\geq (1 + 1.25 \text{Pr}) / 221 && \text{for laminar flow and} \\ \Phi_0 \tan \Phi_0 &\geq 2.22 && \text{for turbulent flow.} \end{aligned}$$

Therefore in the above equations for B_{st} , the term Φ is replaced by $(\Phi - \Phi_0)$ to account for the actual leading edge of the boundary layer.

Boiling boundary layers. When nucleate boiling commences, the characteristic liquid zones discussed earlier remain unchanged; however, additional mechanisms of heat transfer must be considered for a realistic model. One such mechanism is the mixing of the vapour generated at the walls in the boundary layer with the subcooled liquid due to collapsing/condensing of the bubbles at the outer edge of the layer. The extent of this mixing is, however, still unknown and the values employed in the present model are based upon very small scale experiments (ref. 6). This mixing phenomenon is modeled as:

$$\begin{aligned} B_{in} &= B_v + B_1 \\ B_v + B_1 &= B_{st} + B_{out} \\ B_{st} &= B_v f_1 + B_1 f_2 \\ B_{out} &= B_v (1 - f_1) + B_1 (1 - f_2) \end{aligned}$$

where

B_v = amount of vapour generation

B_{in} = amount of subcooled liquid entering the boundary layer

B_{out} = mass diffusion to the bulk region

B_1 = amount of subcooled liquid heated but not converted
to vapour

For the present code we have chosen $f_1 = 0.3$ and $f_2 = 1.0$.

During pressure relief, with pressure increasing, the generated bubbles are further assumed to collapse and release their energy into the surrounding liquid. With pressure decreasing, the bubbles are either allowed to move up into the stratified layer or collapse in the subcooled regions of the bulk fluid as determined by the coefficient f_1 .

Relief valve

When the pressure inside the vessel reaches the set pressure, the valve opens and discharge commences. The discharge rate is

$$\dot{G} = C_D (\rho AV)_{throat}$$

Since in most cases the value of set pressure exceeds that necessary for choked flow, the throat conditions for superheated vapour flow can be related to the entrance conditions using isentropic flow relations. Assuming further that the perfect gas relationship is valid, the maximum flow rate through the valve may be given (ref. 10):

$$\dot{G} = C_D A_p \rho_v [2/(1+\gamma)]^{1/(\gamma-1)} [g(\bar{R}/M) T_v 2/(1+\gamma)]^{1/2}$$

When the vapour is saturated or when all the vapour is discharged and the valve "sees" a two-phase mixture, one can no longer use these approximations. Temperature-enthalpy diagrams must now be used to evaluate the enthalpy drop between the inlet and throat of the valve to calculate the

flow velocity at the throat. The temperature of the throat is known from the inlet conditions if one assumes isentropic flow. The basic equation of mass flux is then:

$$\dot{G} = A_{\text{throat}} [2g_c(h-h_{\text{throat}})/v^2]^{1/2}$$

Closed form expressions relating mass flow rate quality and pressure were obtained using a curve fitting procedure from estimates produced in ref. 17 for propane. Two curves for qualities $X > 0.4$ and $X < 0.4$ were obtained and incorporated in the present program. These are:

$X > 0.4$

$$\dot{G} = A_c [436.73 - 484.17X + P(4.44 - 0.24X)] C_D$$

$X < 0.4$

$$\dot{G} = A_c [802.1 - 1397.89X + P(4.55 - 0.15X)] C_D$$

The PLGS-I program utilizes the wall heat fluxes based on the conservation equations of mass and energy to determine the state of the system and then proceeds to the next time step. The solution procedure is shown in Fig. 2.

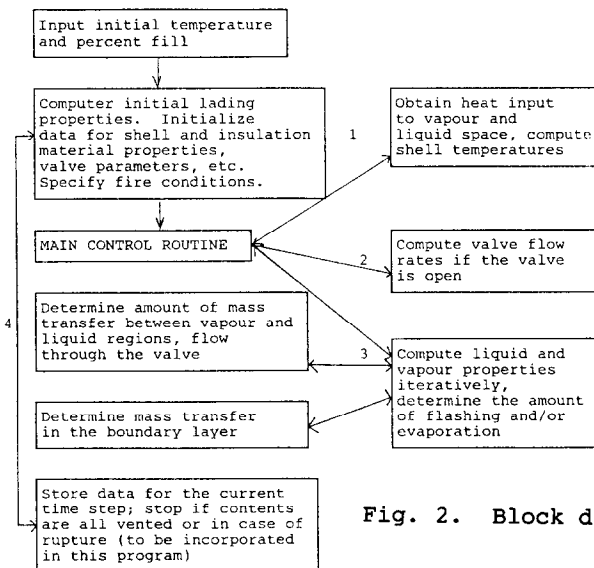


Fig. 2. Block diagram for PLGS-I.

The program is written in HP BASIC 5.0 and run on an HP98220 computer. FORTRAN PC and main frame versions are available.

TRIAL RUNS AND COMPARISONS

Three different PLGS-I computer simulations were conducted in order to consider the field tests of the Association of American Railroads (AAR) of the United States, the Bundesanstalt Fur Materialprufung (BAM) of West Germany, and the Health and Safety Executive (HSE) of the United Kingdom. Table 1 summarizes the test conditions, dimensions, and other important parameters utilized for these tests.

Table 1. Field test parameters and initial conditions.

	AAR (ref.4)	BAM (ref.3)	HSE (ref.5)
Tank diameter (m)	3.05	1.25	1.69
Tank length (m)	18.3	4.3	4.06
Tank capacity (litres)	127,000	4,850	10,090
Wall thickness (mm)	15.9	5.9	11.9
Valve set pressure (barg)	18.6	15.6	14.3
Nominal valve diameter (mm)	80.3	25.4	33.6
Initial temperature (°C)	21.1	10.0	5.7
Percentage fill	96	50	75
Fire temperature (°C) (max-min)	1100-550	900-420	1000-600
Mean fire temperature (°C)	871	843	754

Comparisons between the computer simulations and the corresponding field tests show good agreement. The valve discharge coefficient utilized for the BAM and HSE tests were fixed at 0.7 and 0.85, respectively. In the AAR test, however, the valve discharges two-phase liquid due to the high filling level. With the changing quality of the discharge, the discharge coefficient changes. The value of C_D chosen decreased from 0.85 (when the initial discharge was superheated vapour) to 0.25 after which it increased again due

to the falling liquid level.

The initial opening time of the relief valve is closely predicted in all the simulation runs. One reason for this is the use of a proper model for fire development. Supported by data from several field tests, the fire was assumed to develop exponentially as a function of three variables: time, maximum fire temperature and a time constant. The time constants for the above tests were chosen from the data presented in refs. 3, 4 and 5. The close agreement of the slope of pressure-time curves (Figs. 3,4,5) verifies the validity of the fire development model as well as the pre-vent modeling.

The significance of thermal stratification of the liquid layers is evident from Figs. 6 and 8. In ref. 3, only the liquid temperature data was provided and from Fig. 7, it can be seen that it is very likely that this reported data is representative of the bulk temperatures.

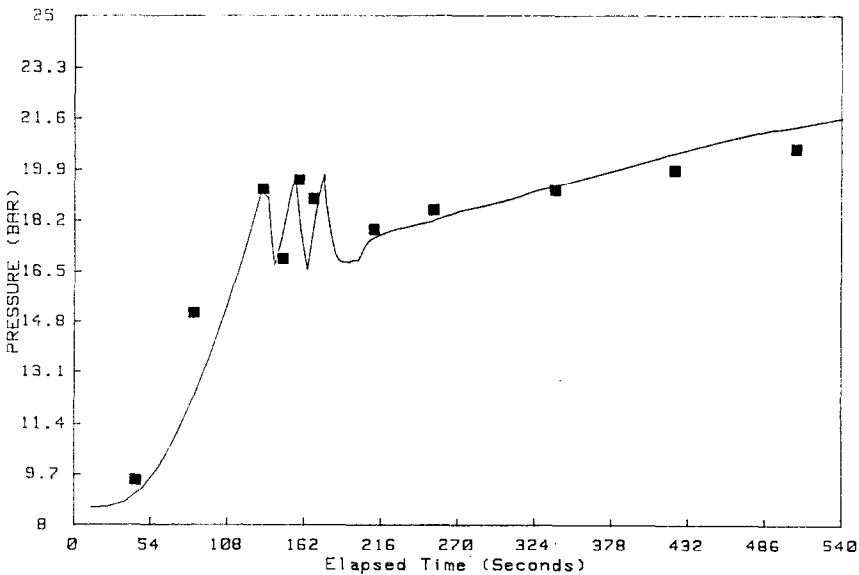


Fig. 3. Pressure response, AAR, 96% propane fill, ■ (ref. 4), — PLGS-1 simulation.

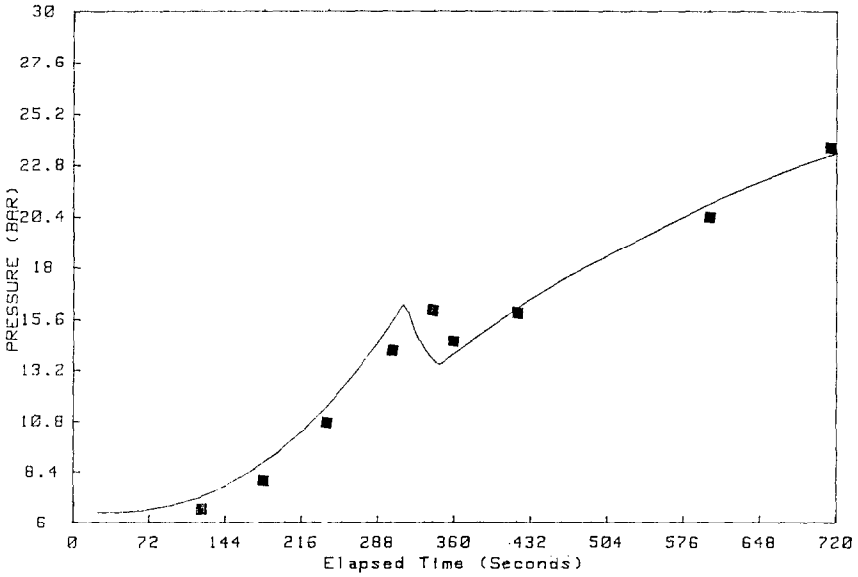


Fig. 4. Pressure response, BAM, 50% propane fill, ■ (ref. 3), — PLGS-1 simulation.

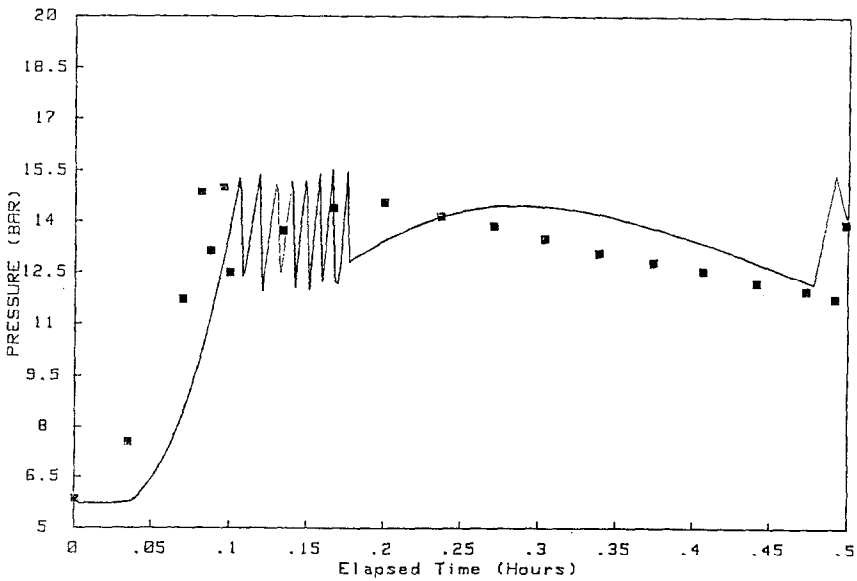


Fig. 5. Pressure response, HSE, 75% propane fill, ■ (ref. 5), — PLGS-1 simulation.

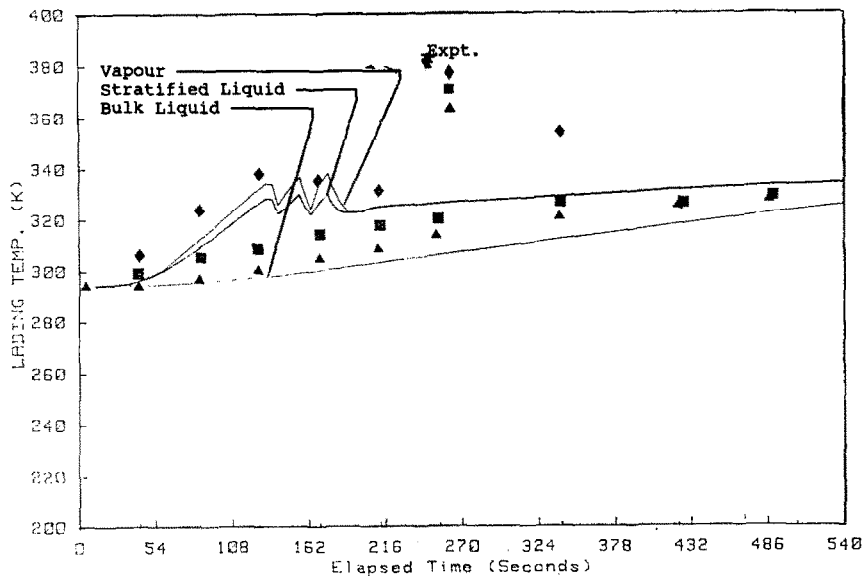


Fig. 6. Lading temperatures, AAR, 96% propane fill (ref. 4).

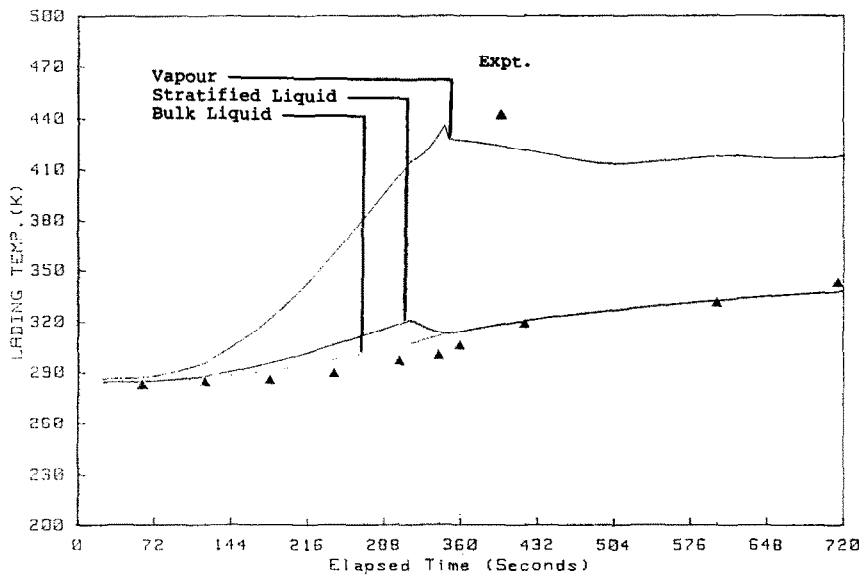


Fig. 7. Lading temperatures, BAM, 50% propane fill (ref. 3).

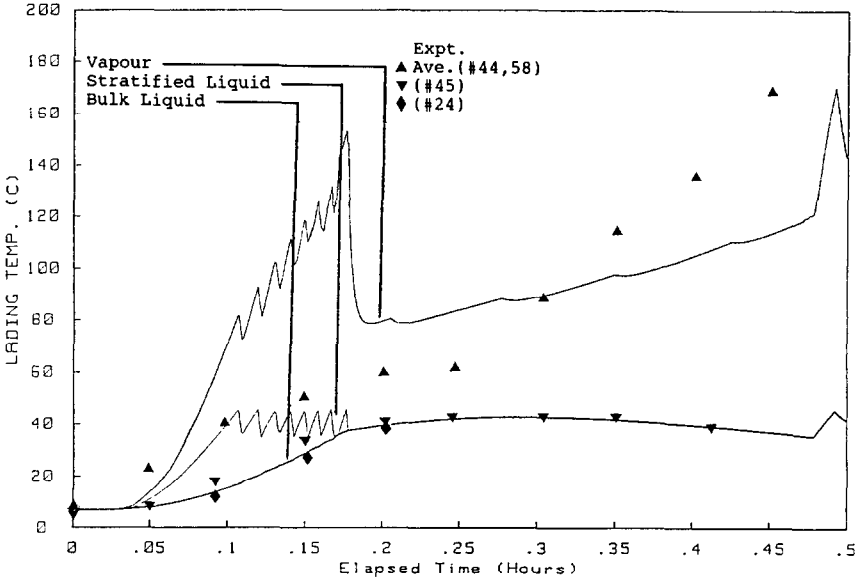


Fig. 8. Lading temperatures, HSE, 75% propane fill (ref. 5), (# : thermocouple number).

The tank shell temperatures are shown in Figs. 9, 10 and 11 along with the computed ones. Because ref. 3 has reported on the values of the shell temperatures at the time of the rupture, only two data points are available on Fig. 10. In Fig. 9, the change of slope in the computed vapour space shell temperatures is due to the presence of a two-phase fluid which totally fills the vapour space and acts as a cooling mechanism on the walls. However, it should be noted that the extent of cooling is not as effective as it would be if the fluid was single-phase. The computer program uses the mean properties of this mixture in calculating the heat transfer from the wall and the slight overprediction of shell temperatures may be interpreted as the inadequacy of the calculation procedure in this region.

Figures 12 and 13 display the predicted mass and percent fill responses for the test of ref 5; agreement is satisfactory.

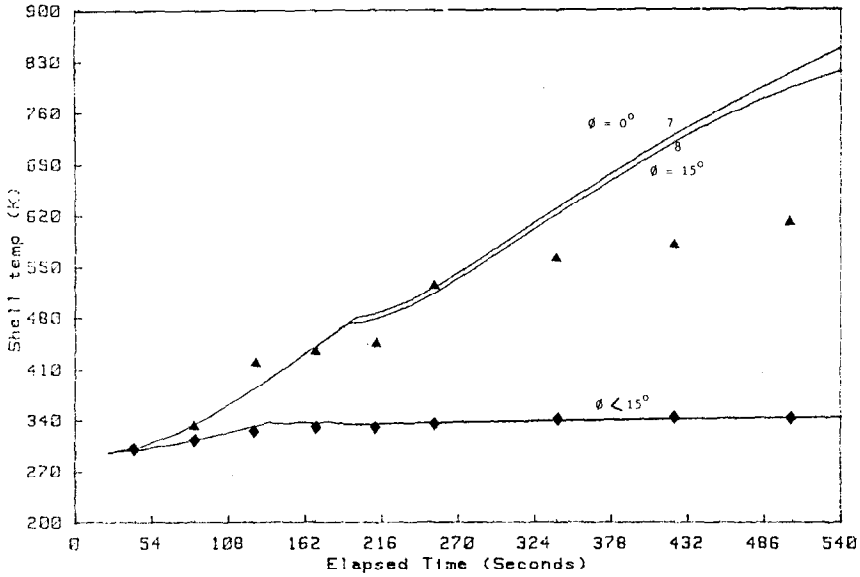


Fig. 9. Wall temperatures, AAR, 96% propane fill, \blacktriangle top, \blacklozenge bottom (ref. 4), — PLGS-1 simulation.

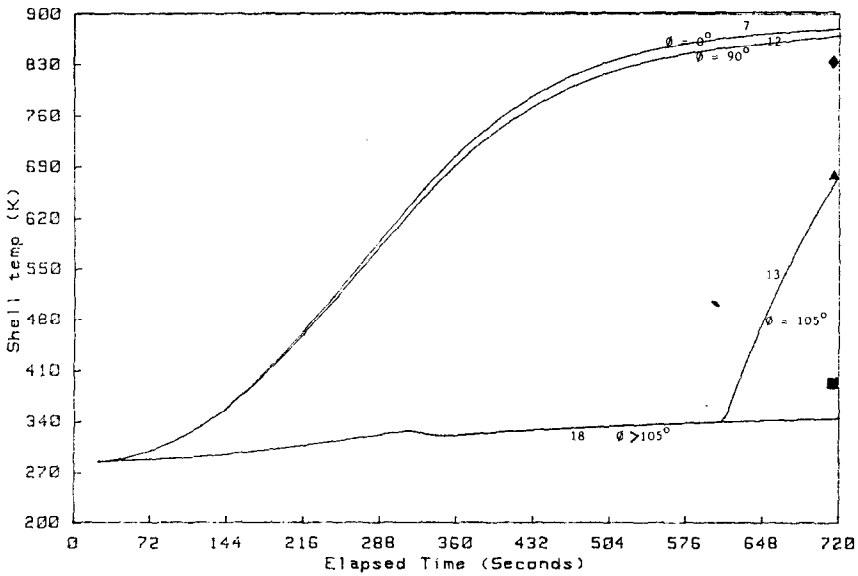


Fig. 10. Wall temperatures, BAM, 50% propane fill, \blacklozenge top, \blacktriangle mid section, \blacksquare bottom (ref. 3), — PLGS-1.

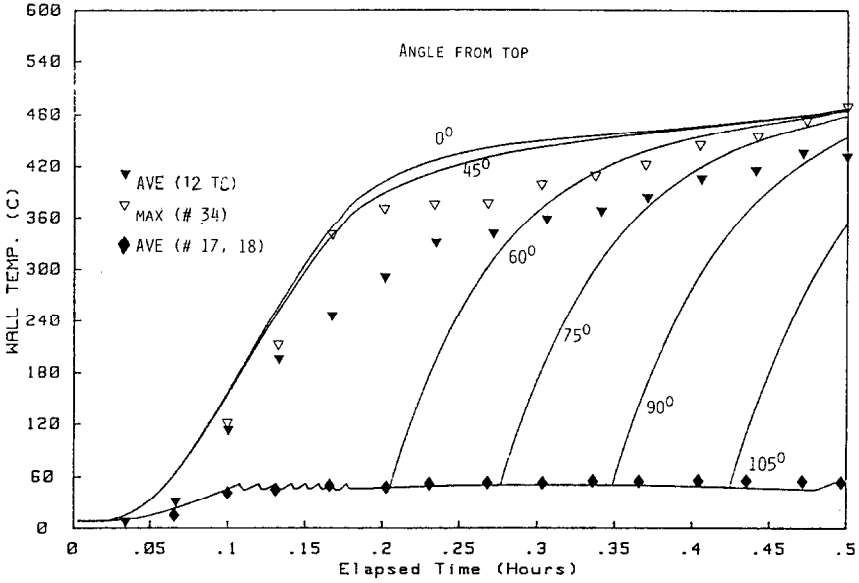


Fig. 11. Wall temperatures, HSE, 75% propane fill (# : thermocouple number, ref. 5), — PLGS-1.

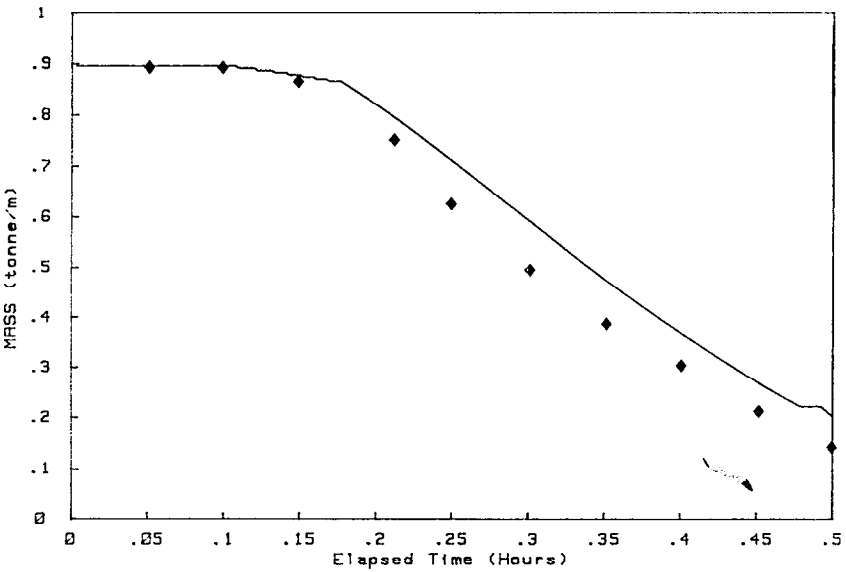


Fig. 12. Mass, HSE, 75% propane fill, ◆ data ref. 5, — PLGS-1.

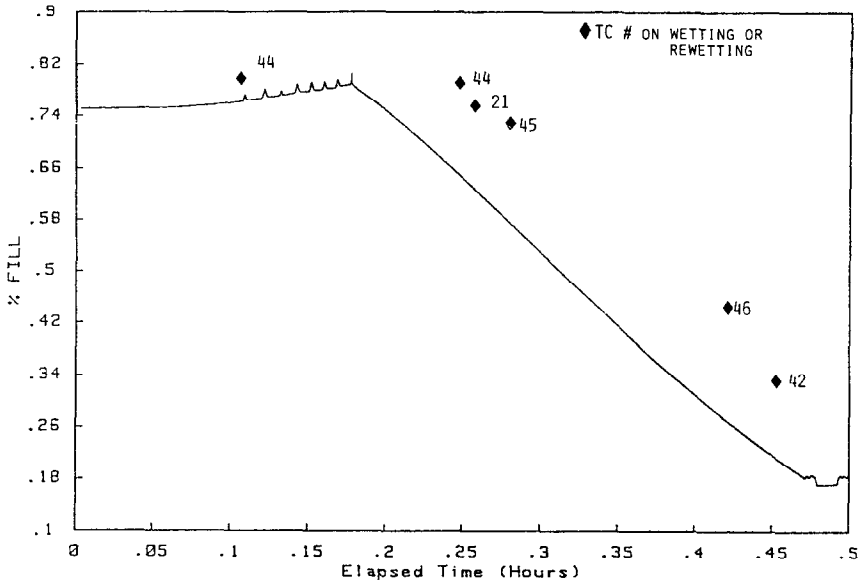


Fig. 13. Percent fill, HSE, 75% propane fill (ref. 5).

CONCLUSIONS AND RECOMMENDATIONS

The comparison of computer predictions with data from several field tests clearly shows that the present code is capable of simulating the response of a cylindrical tank loaded with LPG and exposed to fire.

It is shown that thermal stratification is an inevitable consequence of external heating and that it can be predicted using the present approach. The importance of the discharge coefficient is stressed, especially for those cases where the fill level is high. Detailed analysis is necessary in order to determine the relationship between C_D and other parameters such as void fraction and overpressure.

coefficient is stressed, especially for those cases where the fill level is high. Detailed analysis is necessary in order to determine the relationship between C_D and other parameters such as void fraction and overpressure.

Another point that deserves attention is the method of calculating the heat transfer when the tank walls are exposed to a two-phase fluid. Due to the complex geometry of the enclosure and the presence of other parameters such as finite flow velocities and void fraction, transients induced by pressurization (or depressurization) pose a difficult task. The approach taken in the present code is probably the simplest one where mean fluid properties are used in the correlations intended for single-phase phenomenon.

Because the present code is one of the few which can simulate valve cycling and predict thermal stratification, the present results reflect adequate validations of the physical models chosen.

ACKNOWLEDGEMENT

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NOMENCLATURE

A	area
A_C	area of valve
B_{St}	mass flow rate
C_D	discharge coefficient
C_p	specific heat at constant pressure, _l liquid, _v vapour
D	tank diameter
E	emissive power
F	radiation shape factor
g	gravity constant
g_c	conversion factor
G	incident radiation
G	mass flow rate
h	heat transfer coefficient

h_{fg}	enthalpy of vapourization
k	thermal conductivity
L	characteristic length
M	molecular weight
P	pressure
Pr	Prandtl number
q	heat flux, c convective, R radiative
R	tank radius
R	universal gas constant
Ra	Rayleigh number
T	temperature, sat saturation, w wall
v	specific volume
V	flow velocity
X	quality

Greek Letters

α	thermal diffusivity
β	coefficient of thermal expansion
γ	isentropic coefficient
ϵ	emissivity
θ_{tr}	temperature excess, $(T_w - T_{sat})$
μ	dynamic viscosity
ρ	density
σ	Stefan Boltzmann constant
σ_l	liquid surface tension
τ	transmissivity
Φ	arclength of wall-vapour interface
Φ_2	angle from tank bottom to commencement of boundary layer

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