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Exposure of a liquefied gas container to an external fire

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Abstract

In liquefied gas, bulk-storage facilities and plants, the separation distances between storage tanks and between a tank and a line of adjoining property that can be built are governed by local regulations and/or codes (e.g. National Fire Protection Association (NFPA) 58, 2004). Separation distance requirements have been in the NFPA 58 Code for over 60 years; however, no scientific foundations (either theoretical or experimental) are available for the specified distances. Even though the liquefied petroleum gas (LPG) industry has operated safely over the years, there is a question as to whether the code-specified distances provide sufficient safety to LPG-storage tanks, when they are exposed to large external fires.

A radiation heat-transfer-based model is presented in this paper. The temporal variation of the vapor-wetted tank-wall temperature is calculated when exposed to thermal radiation from an external, non-impinging, large, 30.5 m (100 ft) diameter, highly radiative, hydrocarbon fuel (pool) fire located at a specified distance. Structural steel wall of a pressurized, liquefied gas container (such as the ASME LP-Gas tank) begins to lose its strength, when the wall temperature approaches a critical temperature, 810 K (1000 °F). LP-Gas tank walls reaching close to this temperature will be a cause for major concern because of increased potential for tank failure, which could result in catastrophic consequences.

Results from the model for exposure of different size ASME (LP-Gas) containers to a hydrocarbon pool fire of 30.5 m (100 ft) in diameter, located with its base edge at the separation distances specified by NFPA 58 [NFPA 58, Liquefied Petroleum Gas Code, Table 6.3.1, 2004 ed., National Fire Protection Association, Quincy, MA, 2004] indicate that the vapor-wetted wall temperature of the containers never reach the critical temperature under common wind conditions (0, 5 and 10 m/s), with the flame tilting towards the tank. This indicates that the separation distances specified in the code are adequate for non-impingement type of fires. The model can be used to test the efficacy of other similar codes and regulations for other materials.

Keywords: Flammable liquid fire; LPG tank; Pool fire; Separation distance; Tank-wall temperature; Thermal radiation; View factor

1. Introduction

Many pressurized, liquefied gas tanks containing ambient temperature liquids are located in urban areas zoned for industrial activities. Many of these storage facilities abut other storage facilities storing hydrocarbon fuels such as gasoline, diesel and jet fuel. One of the safety concerns to the liquefied gas tanks is the detrimental effect of an external, non-impinging, hydrocarbon liquid pool fire on the tanks. The size of the pool fires could be large in comparison to the size of the liquefied gas tanks. Thermal radiation from the fire will heat the steel wall of the tanks. The wall in contact with vapor heats up faster than the wall in contact with the liquid due to the lower heat-transfer coefficient between the steel wall and vapor. The liquid-wetted tank wall will remain essentially at the liquid temperature because of high (boiling) heat-transfer rates between the wall and the liquid. Liquid temperature will increase (but not by a very large value) due to internal boiloff, the consequent increase in pressure of vapor inside the tank and the fact that, in general, the liquid and vapor are in saturation equilibrium.

Codes and regulations governing the location of pressurized liquefied gas in storage facilities and bulk plants have recognized the potential adverse impact of exposure of tanks to external fires. Many of the codes/regulations have strict

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Nomenclature

A_e	area of the elemental surface on the tank (m^2)
A_{VWF}	area of the vapor-wetted wall exposed to the fire (m^2)
A_{VWT}	total surface area of the vapor-wetted tank wall (m^2)
b	thickness of steel wall of the tank (m)
c_p	specific heat of the steel of which the wall is made (J/(kg K))
D_F	diameter of the base of the hydrocarbon liquid fuel fire (m)
D_T	outer diameter of the tank shell (m)
L_F	length of (or the height for a non-bent) visible fire (m)
L_T	axial length of the tank shell (m)
E_F	mean radiance ("emissive power") of the fire (W/m^2)
f	curve-fit constant to equivalent radiative heat-transfer coefficient (Eq. (7))
F	geometric view factor between the elemental surface on the tank and the radiating part of the fire
H_T	height of pedestal holding the tank (m)
H_F	height of the base of the fire above ground (m)
h_c	convective heat-transfer coefficient ($W/(m^2 K)$)
h_{eff}	effective (convective and radiative) heat-transfer coefficient ($W/(m^2 K)$)
h_R	equivalent radiative heat-transfer coefficient ($W/(m^2 K)$)
k	fraction of tank volume occupied by liquid
m	total mass of the VWW (kg)
m_e	mass of the elemental area of the tank (kg)
$\dot{q}''$	radiant heat flux from fire absorbed by the wall, $\alpha_S \dot{q}''_F$ (W/m^2)
$\dot{q}''_F$	radiant heat flux from fire incident on the exposed area (W/m^2)
$\bar{\dot{q}}''_F$	mean fire radiation heat flux incident on fire-side VWW surface (W/m^2)
Q	$\frac{A_{VWF}\alpha_S \bar{\dot{q}}''_F}{h_c T_a A_{VWT}} = \frac{\text{incident fire radiative heat input rate}}{\text{convective cooling rate at a temperature difference } T_a}$
R_T	radius of the tank shell (m)
S_D	separation distance between line of adjoining property that can (m) be built upon and the tank shell surface nearest to boundary line
t_{ch}	characteristic cooling time (s) = $(\rho_s b c_p T_a)/(h_c T_a)$ = [time to reduce the VWW sensible heat from a temperature of T_a to zero with a constant convective cooling with a temperature difference of T_a]
T	temperature of the elemental wall surface (K)
T_a	ambient air temperature (K)
$\bar{T}$	average temperature of the VWW (K)
α_s	absorptivity of the tank surface element for fire radiation
β	$\frac{\varepsilon_S f T_a}{h_c} = \frac{\text{effective radiative heat-transfer coefficient at a tempeature difference of } T_a}{\text{convective heat-transfer coefficient}}$
ε_F	emissivity of the fire (set to 1 for optically thick fires)
ε_S	emissivity of the (painted) VWW surface
ϕ	angle with X-axis subtended by the radiation receiving elemental area on the tank-wall surface (rad)
ϕ_{Liq}	angle (w.r.t. horizontal axis) subtended by the liquid surface at the tank center (rad)
ν_1	angle w.r.t. to the base plane of the unit hemisphere made by the line connecting the elemental area on the tank wall and the center of fire base (rad)
ν_2	angle w.r.t. to the base plane of the unit hemisphere made by the line connecting the elemental area on the tank wall and the point on the fire axis at the top of visible fire (rad)
Θ	dimensionless temperature $= \frac{\bar{T}-T_a}{T_a}$
θ_F	angle the axis of the fire makes with the vertical due to wind bending of the fire (positive if bending away from tank) (rad)
ρ_s	density of steel (constituting the tank wall) (kg/m^3)
σ	Stefan–Boltzmann constant (5.6697×10^{-8}) ($W/(m^2 K^4)$)
τ	dimensionless time for heating the VWW surface = t/t_{ch}
τ_{Atm}	transmissivity of the atmosphere to thermal radiation
ω_1	half angle of the tangent to the fire base from the elemental area measured in the plane containing the lines from the elemental area to the center of fire base and the tangent at fire base (rad)
ω_2	half angle of the tangent to the fire top from the elemental area measured in the plane containing the lines from the elemental area to the fire axis at the tops and the tangent to fire top (rad)
ψ	$\frac{A_{VWF}}{A_{VWT}} = \frac{\text{area of VWW over which radiant heat is incident}}{\text{total surface area of VWW}}$

requirements for minimum inter tank distances and between the tank that is nearest to the edge of the plant property and the line of adjoining property that can be built upon. One such code is the National Fire Protection Association (NFPA) 58 Liquefied Petroleum Gas (LPG) Code [1], which requires a minimum distance of 7.6 m (25 ft) for containers of water capacity 500–2000 gal, 15.2 m (50 ft) for tank sizes between 2001 and 30,000 gal and 22.9 m (75 ft) for tanks of 30,001 to

70,000 gal, etc. [1] (NFPA 58, 2004;) from a tank closest to the plant property boundary to the nearest property line that can be built upon. These required separation distances have been in the NFPA 58 Code for over 60 years. A review of the archived minutes of the NFPA's LPG Committee for the past several decades throws no light on the scientific foundation (theoretical or experimental) from which the above minimum distances were set.

One of the concerns is whether the code-specified distances provide sufficient safety to LPG-storage tanks, when they are exposed to external fires. The structural steel from which the tank walls are fabricated begins to lose its yield strength with increase in temperature. As the temperature reaches a critical value of about 810 K (1000 °F), the yield strength decreases by a factor of 2 (SFPE, [2]). While the LPG tanks are constructed to strict ASME design standards, with a factor of safety of about 4 on the allowable stress level, the potential for steel strength decreasing to one half of its normal value is disconcerting. The consequences of a LPG tank-wall failure due to wall failure could be catastrophic (leading to a BLEVE). Therefore, it is very important to ensure that the specified separation distances do provide acceptable levels of safety. Generally, a large fire is responded to by emergency response services (fire departments, etc.) within a very short time of the order of 10 min or less, especially, in an urban setting. In this paper, a maximum fire burning time of 30 min is used for conservative calculations.

This paper addresses the safety of exposure of a LPG container in a storage facility to an external fire. It is not the intent of the paper to determine safety distances for exposure of LPG tanks to external fires or the exposure of (external) structures on property line that can be built upon from fires on LPG tanks. The objective of the paper is strictly to evaluate whether the safety distances enshrined in the LPG Code for over 60 years (and which seem to have provided de facto safety, historically) could be provided a theoretical basis and ascertain their adequacy under normal operating conditions that occur (wind, tank paint conditions, etc).

The model presented is general and is applicable to other facilities and structures exposed to external, non-engulfing fires. The model assumes a large, highly radiative, non-impinging hydrocarbon fuel pool fire located at a specified distance from a container nearest to the property line that can be built upon. The radiant heat incident on the tank elevates the temperature of the tank steel wall in contact with the vapor inside the tank; the time history of the vapor-wetted tank wall is calculated. It is assumed that the exposure to fire is safe, when the tank wall in contact with the vapor does not attain the critical temperature.

Results for exposure of different size ASME containers to large, hydrocarbon pool fires of 30.5 m (100 ft) diameter with the ground level flame surface located at the separation distances specified by NFPA 58 are obtained. Very conservative values are used for the fire parameters to evaluate whether, even under severe conditions, the separation distances are sufficient. The effect of fire plume tilt towards the tank caused by common wind velocities (0, 5 and 10 m/s) has also been evaluated.

2. Analysis

The scenario of the exposure of a LPG tank to a (non-engulfing) fire is illustrated, schematically, in Fig. 1. A "large" hydrocarbon fire of diameter D_F ($D_F \geq 100$ ft) is located such that the flame sheet at the firebase nearest to the tank is just beyond the adjacent boundary. "S_D" represents the distance between the adjacent boundary and the part of the tank nearest to the boundary. The definitions of other parameters used in the model are indicated in Nomenclature.

3. Assumptions in formulating the mathematical model

In developing the model described below, the following assumptions are made:

(1) The fire is located such that the vertical plane containing the vertical axis of the fire, and the line (perpendicular to the long axis of the tank) joining the tank and the fire center bifurcates the tank into two equal parts.
(2) The fire is considered to be optically thick at all locations along the visible height of the fire. The implication of this is that fire can be considered to be a "black body" emitter (radiating the maximum energy flux at a given flame temperature).
(3) The fire is considered to be tilted circular cylinder of length equal to thrice the base diameter and emitting radiation from its entire surface.
(4) The emissive power (i.e., the energy radiated per unit nominal area of the visible fire plume, with radiation along the normal to the this nominal surface of the fire) is constant at all points on the visible plume of the fire. This assumption the same as assuming that the fire has uniform temperature over the entire visible flame height.
(5) Absorption of thermal radiation by the intervening atmosphere is very small for the relatively short distances between the fire and the location of the tank.
(6) The exterior of the tank wall wetted by vapor has a uniform absorption coefficient for radiant heat absorption and a constant emissivity for re-radiation of heat.
(7) The conduction of heat from the hot vapor-wetted wall (VWW) to the relatively cooler liquid-wetted wall is neglected.
(8) Convective and radiative cooling of the vapor-wetted wall is considered.
(9) Calculations are performed to determine the temperature rise of the mid part (along the length) of the tank wall.

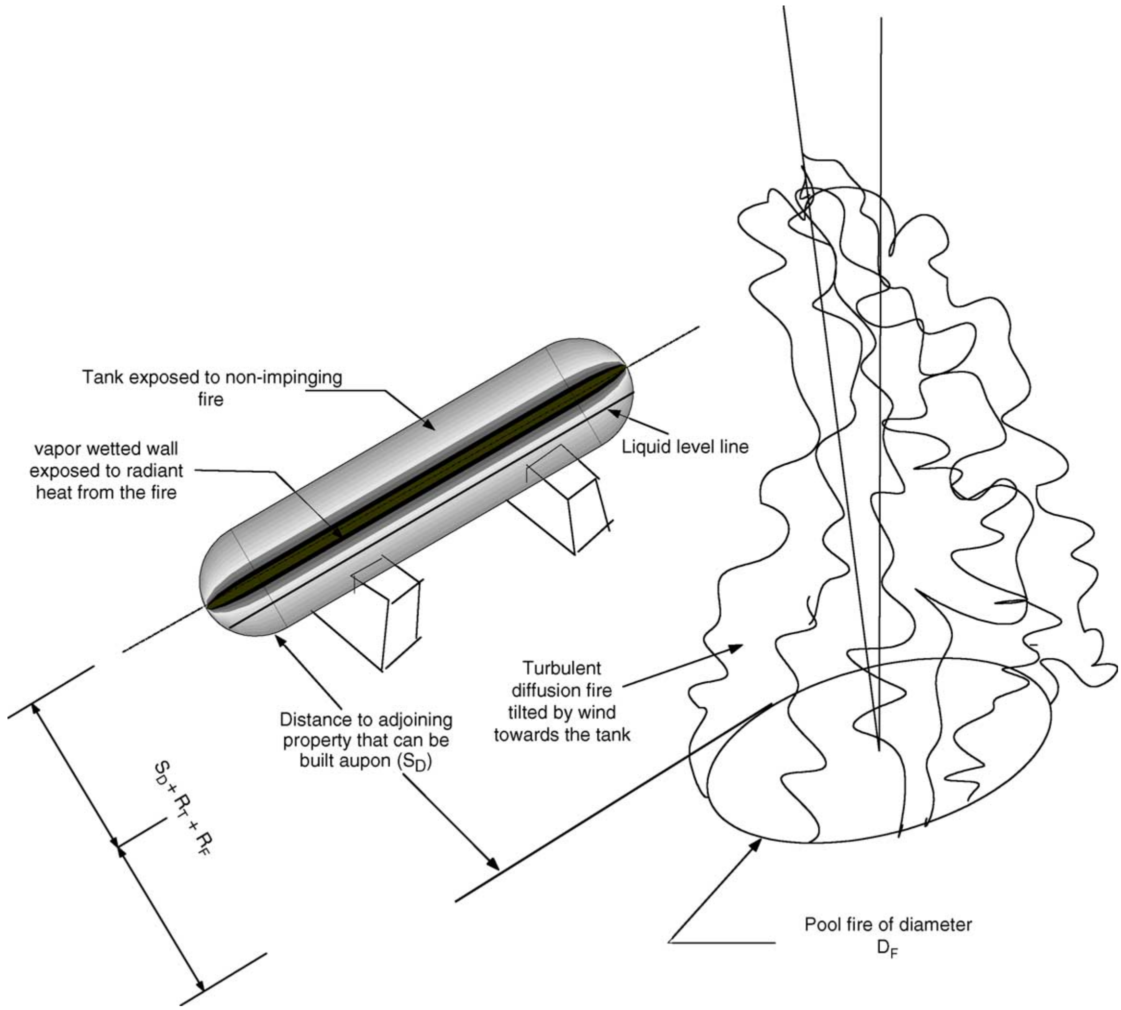

Fig. 1. Schematic representation of the exposure of a LP-Gas tank to an external fire.

4. Heat-transfer equations

The heat flux received by an elemental surface on the tank wall due to radiation from the fire is represented by[1]:

$$\dot{q}'' = \alpha_S \varepsilon_F \tau_{Atm} F\, E_F \quad (1)$$

where $\dot{q}''$ is the heat flux received by the tank wall (W/m^2), α_S the absorptivity of the tank surface element for fire radiation, ε_F the emissivity of the fire (set to 1 for optically thick fires), τ_{Atm} the atmospheric transmissivity for fire thermal radiation, F the geometric view factor between the elemental surface on the tank and the radiating part of the fire and E_F is the mean radiance ("emissive power") of the fire (W/m^2).

The value of the view factor (F) depends on the relative sizes, geometrical locations and orientations of the fire and the heat-receiving element on the tank wall. It is a number, which is always less than unity.

The time-wise variation of the temperature of an elemental area on the tank wall due to heat input from the fire, heat loss by convection and re-radiation to the atmosphere is given by the following equation, with the assumption that that heat loss due to conduction through the metal to other parts of the wall is small:

$$m_e c_p \frac{dT}{dt} = \left[\dot{q}'' - h_{eff}(T - T_a)\right] A_e \quad (2)$$

where m_e is the mass of the elemental area of the tank (kg), c_p the specific heat of the steel of which the wall is made (J/(kg K)), T the temperature of the elemental wall surface (K), T_a the ambient air temperature (K), $\dot{q}''$ the radiant heat flux from fire absorbed by the exposed area (W/m^2), h_{eff} the effective (convective and radiative) heat-transfer coefficient (W/(m^2 K)) and A_e is the area of the elemental surface on the tank (m^2).

Eq. (2) describes the temperature variation with time of any elemental surface area of the tank exposed to the thermal radiation from the fire. The tank wall in contact with the liquid will be essentially at the temperature of the liquid because of high heat-transfer rate between the inner tank wall and liquid (because of nucleate boiling of the liquid).

[1] See Nomenclature for the description of the symbols.

Our interest in this paper is the temperature of the vapor-wetted wall of the tank. This part of the tank wall does not transfer heat efficiently to the vapor and hence increases in temperature with time. We can assume, without loss of generality, that the spatial variation in the vapor-wetted tank-wall temperature is small because of good heat conduction through the thickness of the metal compared to the low heat transfer to the vapor. That is, we can assume a mean temperature for the VWW (for both parts of the tank wall facing the fire and away from the fire). The temporal variation of this mean temperature of the VWW is then given by:

$$mc_{\mathrm{p}}\frac{\mathrm{d}\bar{T}}{\mathrm{d}t} = \bar{\dot{q}}'' A_{\mathrm{VWF}} - h_{\mathrm{eff}} A_{\mathrm{VWT}}(\bar{T} - T_{\mathrm{a}}) \quad (3)$$

where m is the total mass of the VWW (kg), $\bar{\dot{q}}''$ the mean fire radiation heat flux absorbed by the VWW surface facing the fire (W/m^2), $\bar{T}$ the average temperature of the VWW (K), A_{VWF} the area of the vapor-wetted wall exposed to the fire (m^2) and A_{VWT} is the total surface area of the vapor-wetted tank wall (m^2).

The mass of the VWW surface is given by:

$$m = \rho_{\mathrm{s}} b A_{\mathrm{VWT}} \quad (4)$$

where ρ_{s} is the density of steel (constituting the tank wall) (kg/m^3), and b is the thickness of steel wall of the tank (m).

Eq. (3) can be solved for $\bar{T}$ as a function of time for specified value of $\bar{\dot{q}}''$, the fire heat flux received averaged over the VWW surface facing the fire. In our calculations, we assume that the heat flux from the fire does not vary with time but is dependent on the fire size, fire characteristics, and its distance from the tank wall. The details of calculation of $\bar{\dot{q}}''$ are described in a later section.

5. Maximum vapor-wetted wall temperature

For a given tank (of a known volume) containing a specified volume of liquid the total surface area of the tank wall wetted by the vapor can be determined. Depending upon the relative geometrical location of the fire, its size and tilt by any wind that may be present, the vapor-wetted wall surface area exposed to the radiant heat flux from the fire will vary. Fig. 2 shows the relationship between the vapor-wetted area and the fraction of the tank volume filled with liquid for a cylindrical tank (neglecting the dished ends).

The ratio of the vapor-wetted wall area to the area over which heat radiation is incident on the vapor-wetted wall is represented as $\psi(k)$ where k is the fractional liquid volume in the tank and,

$$\psi(k) = \frac{A_{\mathrm{VWF}}}{A_{\mathrm{VWT}}} \quad (5)$$

Substituting Eq. (5) in Eq. (3), it can be shown that the maximum VWW temperature is given by,

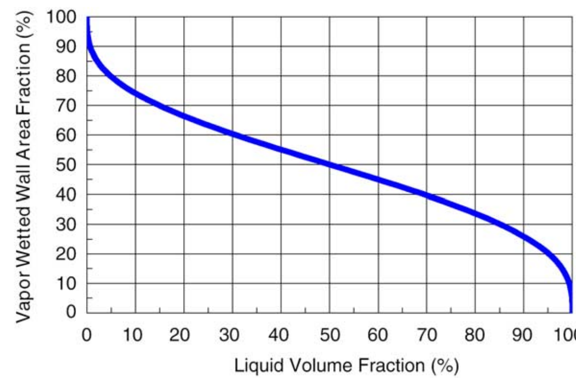

Fig. 2. Fraction of the total tank surface area wetted by vapor vs. fraction of total tank volume occupied by liquid in an ASME tank.

$$\bar{T}_{\max} - T_{\mathrm{a}} = \psi(k)\frac{\bar{\dot{q}}''}{h_{\mathrm{eff}}} \quad (6)$$

6. Effective heat-transfer coefficient

The VWW of the tank surface is cooled both by the convective heat transfer by the movement of air (thermally induced air circulation or due to wind) around and over the tank. In addition, cooling also occurs due to thermal radiation emanating from the VWW surface. For relatively small temperature difference between the VWW surface and air, the cooling rate by thermal radiation can be represented by a linear dependence on the temperature difference. Based on this assumption, the effective hear transfer coefficient (h_{eff}) for VWW surface cooling an be represented by,

$$h_{\mathrm{eff}} = h_{\mathrm{c}} + \varepsilon_{\mathrm{s}} h_{\mathrm{R}} \quad (7)$$

and,

$$h_{\mathrm{R}} = \frac{\sigma(\bar{T}^4 - T_{\mathrm{a}}^4)}{\bar{T} - T_{\mathrm{a}}} \quad (8)$$

where h_{c} is the convective heat-transfer coefficient, h_{R} the equivalent radiative heat-transfer coefficient, σ the Stefan–Boltzmann constant and ε_{S} is the emissivity of the VWW surface.

For specified environmental conditions (air temperature, wind speed, etc.) and dimensions of the tank, the mean convective heat-transfer coefficient h_{c} can be calculated using the correlations provided by Rohsenow and Choi [3].

A correlation for the radiative heat-transfer coefficient (h_{R}) in the form:

$$h_{\mathrm{R}} = f(\bar{T} - T_{\mathrm{a}}) \quad (9)$$

where "f" is a curve-fit constant can be developed using Eq. (8) for a specified range of temperature. Fig. 3 shows the variation of h_{R} with tank surface temperature for the temperature range from 300 K (80 °F) to 800 K (980 °F). Also, shown in this figure is the least square fit for the radiative heat-transfer coefficient using the empirical correlation in Eq. (9).

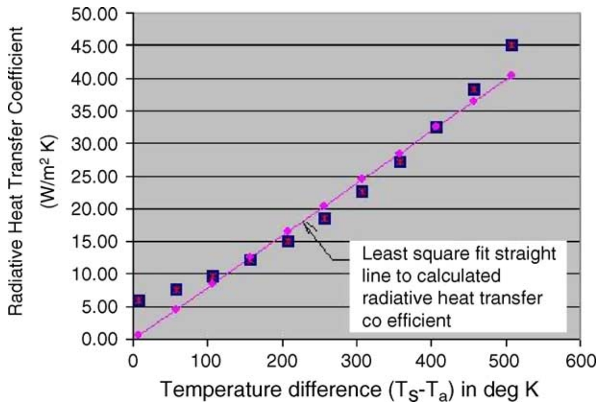

Fig. 3. Radiative heat-transfer coefficient and its correlation with temperature difference between radiator and air.

The value of the factor "*f*" is 0.079657. Because of the dependence of the effective radiative heat-transfer coefficient on the temperature difference, the radiative heat flux is assumed to be modeled with dependence on the square of the temperature difference rather than a linear dependence.

7. Calculation of the mean radiative heat flux absorbed by the wall ($\bar{\dot{q}}''$)

In general, an average value can be used for the emissive power (E_F) of a turbulent diffusion fire of specified diameter over a specified hydrocarbon liquid pool. The absorption of thermal radiation in the intervening atmosphere is strongly dependent on the distance between the fire and the object as well as on the humidity in the atmosphere. The radiant thermal absorptivity (α_S) of the surface of a material depends on the material properties, surface conditions and the spectral distribution of heat radiation. In general, LP-gas tanks are painted white and reflect most of the incident visible and IR radiation. Thermal radiative absorptivity of white paint is about 0.3 (Baumeister and Marks [4]).

In the problem of interest to this paper, the distances between the fire and the exposed tank surface are of the orders of several meters to about 30 m. Over this distance, the atmospheric transmissivity ranges between 0.9 and 0.75 [5]. However, for a conservative calculation in hazard assessment, in this paper, the transmissivity (τ in Eq. (1)) of the atmosphere is considered to be 1. Also, hydrocarbon fuel fires of the size of interest to this paper ($D_F > 30$ m) are optically thick; that is, the fire emissivity is unity ($\varepsilon_F = 1$).

Using the above assumptions, Eq. (1) can be re-written as follows:

$$\bar{\dot{q}}''_F = \bar{F}_{F\to S}\, E_F \tag{10a}$$

$$\bar{\dot{q}}'' = \alpha_S \bar{\dot{q}}''_F \tag{10b}$$

where $\bar{F}_{F\to S}$ is the mean view factor between the fire and the VWW is the surface facing the fire.

Table 1
Values of parameters used in calculations

LPG tank nominal w.c. volume (gal)	Tank overall dimensions				Shell surface area (m^2)	
	Diameter (m)	Length (m)	Tank pedastal height above ground (m)	Shell thickness (mm)	Total surface	Vapor-wetted wall surface area @ 25% full liquid level
1000	1.041	4.88	0.5	8.15	16.0	10.1
2000	1.168	7.32	0.5	8.56	26.9	17.0
4000	2.134	5.18	0.75	11.20	34.7	22.0
12000	2.134	13.72	0.75	11.20	91.9	58.2
18000	2.743	12.5	1.0	14.96	107.7	68.2
30000	2.743	20.12	1.0	18.06	173.4	109.7
60000	3.327	27.43	1.0	18.06	286.8	181.5

Other parameter values

Parameter	Symbol	Value
Tank steel		
Tank steel density (kg/m^3)		8030
Tank steel specific heat (J/(kg K))	C_S	502.3
Tank steel paint emissivity @ air temperature	ε_S	0.85
Tank steel paint radiation absorptivity	α_S	0.3
Fire		
Hydrocarbon fuel fire diameter (m)	D_F	30.5
Visible fire height (m)	H_F	91.4
Fire duration (min)	t_F	30
Fire radiative emissive power (kW/m^2)	E_F	100
Ambient air		
Temperature (K)	T_a	294.4
Overall convective heat-transfer coefficient (W/(m^2 K))	h_c	10
Tank cooling by re-radiation: effective radiant heat-transfer coefficient (W/(m^2 K))	h_R	20

Therefore, for a given fire characteristics, the calculation of $\bar{\bar{q}}''$ is equivalent to calculating $\bar{F}_{\mathrm{F}\to\mathrm{S}}$ (which is represented by F for convenience).

8. Calculation of the mean view factor ($\bar{F}$) between the fire and VWW

To calculate $\bar{F}$ we assume that a tilted cylinder represents the radiating part of the fire plume. Also assumed is that fire axis is tilted and is located on the perpendicular bisector plane of the longitudinal axis of the tank (Fig. 1). In addition, the circular periphery on land of the firebase is assumed to be touching the boundary line of the adjoining property that can be built upon (see Fig. 1). The view factor to each elemental tank VWW surface facing the fire is calculated and overall mean value of the view factor is obtained from these results. The details of the calculation of the view factor are discussed in Appendix A.

9. Solution to the equation on VWW surface-temperature variation with time

Substituting the results of Eqs. (4), (5), (7) and (9) in Eq. (3) and re-arranging, we get the following equation:

$$\rho_{\mathrm{s}} b c_{\mathrm{p}} \frac{\mathrm{d}(\bar{T}-T_{\mathrm{a}})}{\mathrm{d}t} = \psi \alpha_{\mathrm{S}} \bar{\bar{q}}''_{\mathrm{F}} - [\varepsilon_{\mathrm{s}} f(\bar{T}-T_{\mathrm{a}}) + h_{\mathrm{c}}](\bar{T}-T_{\mathrm{a}}) \tag{11}$$

The above equation is written in dimensionless form as follows:

$$\frac{\mathrm{d}\Theta}{\mathrm{d}\tau} + \Theta + \beta\Theta^2 = Q \tag{12}$$

with the condition,

$$\Theta \equiv \Theta(\tau); \qquad \Theta(0) = 0 \tag{13}$$

The various dimensional and dimensionless parameters are defined as follows:

t_{ch} = characteristic cooling time = $(\rho_{\mathrm{s}} b c_{\mathrm{p}} T_{\mathrm{a}})/(h_{\mathrm{c}} T_{\mathrm{a}})$ = time to reduce the VWW sensible heat at T_{a} to zero with a constant convective cooling with a temperature difference of T_{a}

τ = dimensionless (VWW heating) time = t/t_{ch}

$$\Theta = \text{dimensionless temperature} = \frac{\bar{T}-T_{\mathrm{a}}}{T_{\mathrm{a}}} \tag{14}$$

$$\beta = \frac{\varepsilon_{\mathrm{S}} f T_{\mathrm{a}}}{h_{\mathrm{c}}} = \frac{\text{effective radiative heat-transfer coefficient at a tempeature difference of } T_{\mathrm{a}}}{\text{convective heat-transfer coefficient}}$$

$$Q = \frac{A_{\mathrm{VWF}}\, \alpha_{\mathrm{S}}\, \bar{\bar{q}}''_{\mathrm{F}}}{h_{\mathrm{c}}\, T_{\mathrm{a}}\, A_{\mathrm{VWT}}} = \frac{\text{incident fire radiative heat-input rate}}{\text{convective cooling rate at a temperature difference } T_{\mathrm{a}}}$$

It can be shown that the solution to Eq. (12) with the condition in Eq. (13) is:

$$\tau = \frac{1}{\sqrt{1+4\beta Q}} \ln\left[\frac{\sqrt{1+4\beta Q}-1}{\sqrt{1+4\beta Q}+1}\,\frac{\sqrt{1+4\beta Q}+(1+2\beta\Theta)}{\sqrt{1+4\beta Q}-(1+2\beta\Theta)}\right] \tag{15}$$

and

$$\Theta_{\max} = \frac{\sqrt{1+4\beta Q}-1}{2\beta} \tag{16}$$

10. Results

The temperature variation with time of the vapor-wetted wall of different size ASME containers (horizontal axis, cylindrical) when exposed to an external, non-impinging hydrocarbon pool fire is calculated using Eqs. (15) and (16). In this paper, several sizes of LPG containers are considered and the temperatures of the respective VWWs when exposed to a 30.5 m diameter hydrocarbon pool fire are calculated. It is assumed in these calculations that the edge of the fire is located at the safe separation distance specified in NFPA 58 for the nearest property line that can be built upon (this distance is 25 ft for containers of less than or equal to 2000 gal capacity, 50 ft for all containers with volumes in the 2001 to 30,000 gal range and 75 ft for a 60,000-gal container).

The values of parameters used in the example calculations are indicated in Table 1. The variations of VWW temperature at different wind speeds have also been obtained. Wind has two effects, namely, it bends the fire plume and second, it increases the forced convective cooling of the tank surface. In calculating the results presented in this paper, it is assumed that the wind is normal to the axis of the tank and is blowing in such a way as to bend the fire towards the tank; such an assumption makes the evaluation conditions most severe because the wind makes the fire plume closer to the tank than in calm conditions. The fire duration is arbitrarily assumed to be 30 min. This time represents the maximum duration within which emergency response will be initiated for a LPG tank exposed to a fire.

The calculated results (including the values for the various non-dimensional characteristic parameters) are presented in Table 2. The results in Table 2 are also shown in Fig. 4A and B. Fig. 4A shows the VWW temperature attained in 30 min

Table 2
Temperature of vapor-wetted wall of LPG tanks exposed to thermal radiation from 30.5 m (100 ft) diameter external, hydrocarbon fuel fire

Tank capacity (w.c.), V (gal)	Tank shell diameter, D_T (m)	Tank shell thickness, B (mm)	Wind velocity, U_W (m/s)	Distance from tank to nearest edge of fire plume, S_D (m)	Effective convective HTC, h_{eff} (W/(m^2 K))	Other parameters			View factors to the VWW		Temperature attained by vapor-wetted wall	
						β	Q	t_{ch} (s)	Average over the all surfaces receiving thermal radiation, F_{avg}	Maximum view factor which occurs on one VWW element, F_{max}	During a 30 min duration exposure, T_{VWW} (K)	Maximum Temp attained for ver y long exposure, T_{max} (K)
1000	1.041	8.15	0	7.62	4.5	4.426	7.839	7300	0.346	0.352	622.0	654.3
			5		13.4	1.470	1.671	2425	0.222	0.400	494.0	523.7
			10		20.4	0.970	1.355	1600	0.273	0.449	504.0	522.2
2000	1.168	8.56	0	7.62	4.6	4.585	8.124	7941	0.347	0.353	615.8	655.5
			5		12.8	1.555	1.773	2694	0.223	0.402	495.4	528.1
			10		19.4	1.026	1.437	1777	0.274	0.451	504.4	527.7
4000	2.134	11.20	0	15.24	3.7	5.330	7.535	12134	0.277	0.284	537.0	617.9
			5		10.1	1.979	1.855	4506	0.183	0.339	450.7	514.6
			10		15.3	1.306	1.554	2973	0.233	0.392	472.0	522.1
12000	2.134	11.20	0	15.24	3.7	5.330	7.535	12134	0.277	0.284	537.0	617.9
			5		10.1	1.979	1.855	4506	0.183	0.339	450.7	514.6
			10		15.3	1.306	1.554	2973	0.233	0.392	472.0	522.1
18000	2.743	14.96	0	15.24	3.5	5.676	7.956	17181	0.274	0.284	498.3	617.9
			5		9.1	2.188	2.054	6625	0.184	0.341	425.3	520.2
			10		13.8	1.444	1.721	4371	0.233	0.393	449.7	529.7
30000	2.743	18.06	0	15.24	3.5	5.676	7.956	20741	0.274	0.284	469.1	618.0
			5		9.1	2.188	2.054	7998	0.184	0.341	409.5	520.2
			10		13.8	1.444	1.721	5276	0.233	0.393	430.9	529.7
60000	3.327	18.06	0	22.86	3.347	5.956	6.925	21767	0.227	0.236	444.2	588.1
			5		8.432	2.364	1.868	8640	0.155	0.294	391.6	501.1
			10		12.78	1.560	1.629	5700	0.204	0.353	415.9	515.4

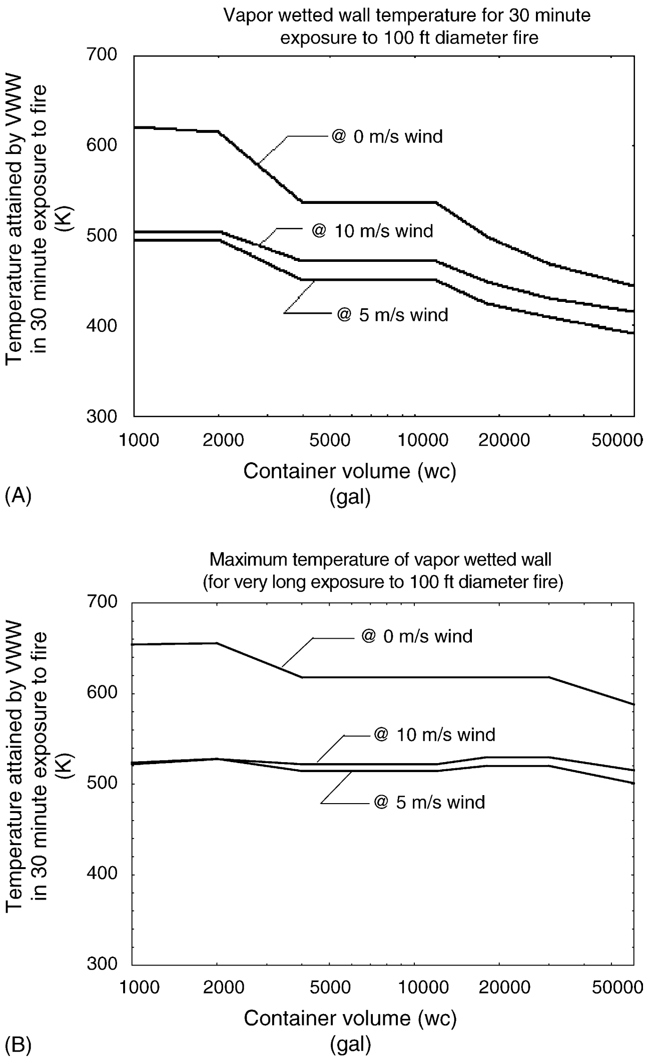

Fig. 4. (A) VWW temperature in different volume containers attained in 30-min exposure to the fire. (B) Maximum VWW temperature attained in containers of different volumes for very long exposure to the fire.

exposure for different tank sizes. Fig. 4B shows the maximum temperature attainable consistent with the wind, fire and tank characteristics. Fig. 5A shows the both the maximum temperature that could be attained by the VWW and the time-wise history of VWW temperature, for different wind speeds fir a 1000-gal container. Fig. 5B and C provide similar calculated results for 4000-gal container and a 30,000-gal container, respectively.

11. Discussions

It is seen from the results shown in Fig. 4A that irrespective of the wind speed the VWW temperature of the 1000-gal tank is higher than that of either the 4000-gal tank or the 30,000-gal tank for the 30-min exposure to the fire. That is, a smaller size container attains a higher temperature compared to that of a larger size container, for any specified time. Also, the maximum attainable temperature (Fig. 4B) is also higher for a smaller container. There are several phenomena that are contributing to this result. First, the location of the fire to a smaller container (because the fire edge was located at the

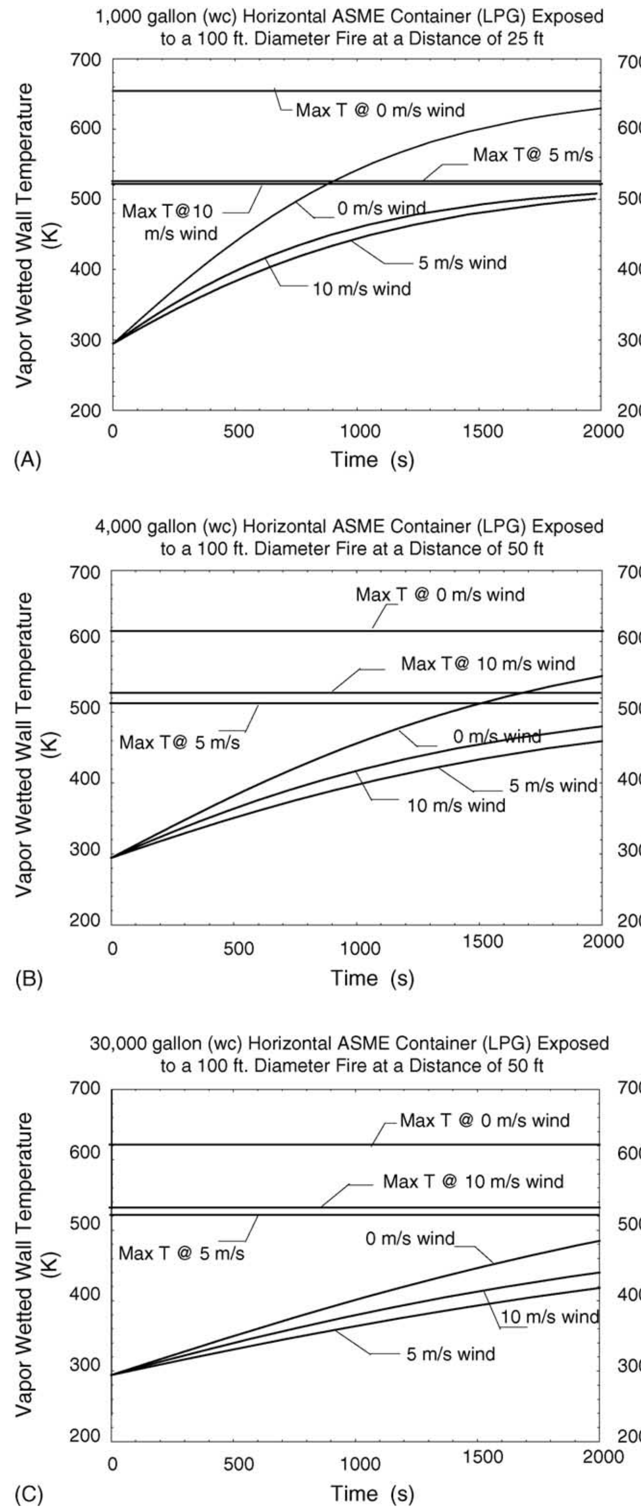

Fig. 5. (A) VWW temperature variation with time for a 1000-gal container. (B) VWW temperature variation with time for a 4000-gal container. (C) VWW temperature variation with time for a 30,000-gal container.

minimum distance specified in NFPA 58 [1]) is closer than to larger container. Second, because the fire bends in a wind and the fire plume length (assumed 3* fire base diameter) is large closer in fires “see” a larger VWW area than in the case of a larger tank. Third, smaller tanks have thinner walls contributing to the more rapid rise of the tank-wall temperature for the same heat input. However, for a given wind speed the convective heat-transfer coefficient is larger for a smaller tank diameter than for a tank of larger diameter; but this effect is small.

It is also noticed from Fig. 4A and B that temperature reached is highest (for each tank size) when the wind has zero velocity (“calm condition”). This is a result of the fact that at low wind the convective heat transfer is primarily by natural convection and this is not a very efficient cooling mechanism. Even a slight wind will increase the cooling of the surface by forced convection. The results indicate that the forced convective heat-transfer coefficient is generally a factor of two to three higher than the natural convection heat-transfer coefficient. In the analysis both natural convection and forced convection heat-transfer coefficients were calculated, at each wind speed, and the higher of the two values was used in subsequent calculations.

The results shown in Fig. 5A–C show the variation of the VWW with time for each of the tank sizes considered. Also shown are the maximum temperatures attainable for very long exposures. The time axis is terminated at about 2000 s because of the assumption of fire life of 30 min. It is seen that the 1000-gal tank has higher temperature at any instant of time compared to that of either 4000- or 30,000-gal tank due to the effects discussed above. The second important information that can be seen is that there is not much difference in the VWW temperatures at the 5 m/s wind and 10 m/s wind; in fact, the temperature in the 5 m/s wind is somewhat lower than that in the higher wind. There are a number of non-linearly interacting physical parameters that result in such an apparent anomaly. These include the degree of bending of the fire, the extent of VWW area “seen” by the fire, and the cooling effect of wind. However, it is seen that the differences in the temperature between 5 and 10 m/s wind is minimal. Again this is due to the fact that the parameters have about the same values (with slight differences) compared to the values at 0 m/s wind.

One of the most interesting result from this study is that the VWW temperature, either in the 30 min exposure to the fire or the maximum attainable temperature for very long exposure does not exceed the critical “softening” temperature of 810 K (1000 °F) (SFPE [2]) for any of the tank sizes considered in this analysis. Considering that ASME tanks are built with a factor of safety of 3–4 and that the worst VWW temperature is, at least, 160 K below the critical limit, it can be argued that the tanks are relatively safe even under the very severe exposure conditions considered in this assessment.

In developing the results presented in this paper, very conservative assumptions were made. The significant ones relate to the characteristics of a hydrocarbon fire. The locations of LPG facilities will be at best in industrial urban areas, where there may exist next to the facility oil tanks, gasoline tanks and other flammable chemical storages. However, it is known that all pool fires of these hydrocarbon fuels burn with copious production of black soot, which reduces the effective emissive power of even visible flames. The experimentally measured emissive powers for visible flames are in the 30–50 kW/m^2 range. In these calculations a value of 100 kW/m^2 has been assumed. Also, the visible length of the flame plume has been set to three times the diameter. Generally, the visible plume length for a 30 m diameter fire with very little smoke production is about 1.5–2 times the diameter. These very conservative assumptions have been made here to calculate the worst-case VWW temperature increase. As seen by the results presented, even under these conditions the maximum temperature reached is well below the critical temperature for structural steel.

The temperatures shown in Table 2 are calculated with the absorptivity (α_S) of 0.3 for the white paint coated steel. White paint reflects most of the thermal radiation received. The calculated maximum temperature is, however, sensitive to the value used for this parameter. If this factor is 0.6 instead of 0.3, then the calculated maximum temperature for a 30,000-gal tank will increase from 483 K (410 °F) to 584 K (592 °F). Therefore, from the point of view of tank protection, it is imperative to maintain the LP-gas tank surface clean and white to reflect most of the incident heat radiation.

12. Conclusions

The analysis presented in this paper together with the calculated results for the temperature attained by LPG tanks of various volume capacities lead us to conclude the following:

(1) The safe separation distance requirements in NFPA 58 Standard for locating ASME LPG containers from the line of property that can be built upon are adequate to protect the tank wall (VWW) from overheating when exposed to hydrocarbon fires located at the property line provided that the tank surface is maintained to be white to reflect much of the incident heat radiation.
(2) The model developed provides a scientific basis on which the effectiveness of protections against exposure of LPG or other tanks to external fires can be evaluated.

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Appendix A

A.1. Calculation of the view factor between a horizontal axis cylindrical tank and a bent-over pool fire

The procedure for obtaining the radiation view factor between an elemental surface on the tank wall of a horizontal axis, cylindrical tank and an external pool fire (not impinging on the tank) is discussed in this appendix. The "unit hemisphere method" by Hottel and Sarofim [6] is used to obtain conservative estimates of the view factors. This method has the advantage that it is simple and provides analytical solutions without losing the accuracy, significantly. Exact equations for calculating the view factor, if used, will require a numerical solution; hence, that procedure is not used here.

Fig. A.1 shows, schematically, the relative geometrical positions and orientations of the pool fire and the cylindrical tank being irradiated. The parameter of interest to the study in this paper is the radiative heat transfer to the wall of the tank wetted by the vapor within it. The definitions of the parameters used are indicated at the end of appendix.

The analysis indicated below:

(1) Calculates the maximum view factor between the elemental area and the fire. This is because the axis of the unit hemisphere is oriented to be normal to the 'plane' of the elemental area.
(2) Takes into account, automatically, only the portions of the fire, which are 'visible' from the position of the elemental area.
(3) Obtains a conservative estimate for the view factor (and not its exact value).

Referring to Fig. A.1, the horizontal distance between the tank center and the center of the firebase is given by:

$$S = R_T + S_D + \frac{1}{2}D_F \qquad \text{(A.1)}$$

Assume a Cartesian coordinate system with the origin on the ground directly under the center of the tank. Let the coordinates of any point in space be represented by *x*–*z*, where *x* is the horizontal coordinate, *y* the coordinate along the axis of the tank (right-hand screw rule) and *z* is the vertical coordinate.

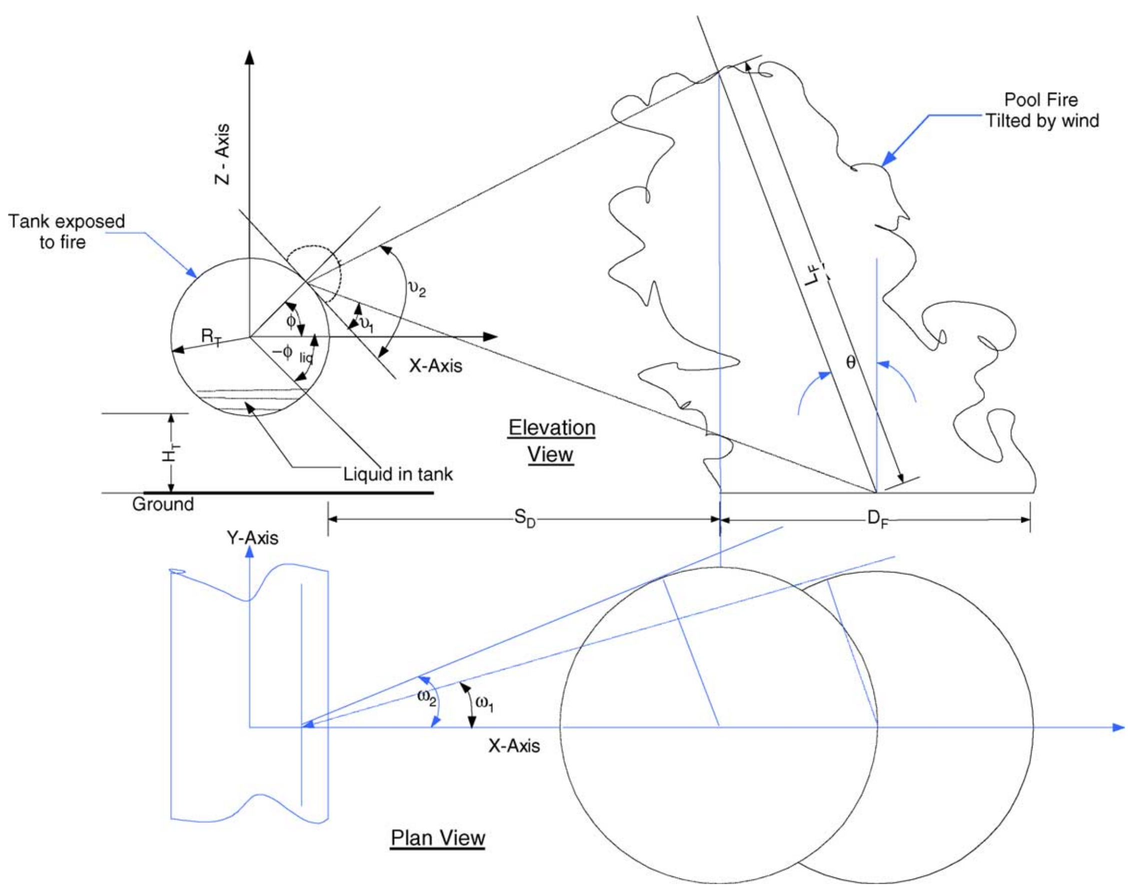

Fig. A.1. Sketch illustrating the relative positions of the fire, tank and the radiation receiving element on the tank surface and the various angles.

With the above convention, the elemental area at P has the following coordinates:

$$P(x, y, z) = P[R_T \cos(\phi), 0, \{H_T + R_T + R_T \sin(\phi)\}] \tag{A.2}$$

The coordinate of the center of fire on the base is:

$$M(x, y, z) = M[S, 0, H_F] \tag{A.3}$$

The coordinate of the top center of fire on the axis is:

$$N(x, y, z) = N[\{S + L_F \sin(\theta_F)\}, 0, \{H_F + L_F \cos(\theta_F)\}] \tag{A.4}$$

Consider a hemisphere of unit radius drawn at the location of the elemental surface on the wall of the tank. The hemisphere is aligned such that its base plane forms the tangential plane to the tank surface at the location of the elemental area of interest.

The distance PM from the center of the unit hemisphere on the wall element and the fire axis at the pool (firebase) is given by:

$$\begin{aligned}\mathrm{PM} = \mathrm{sqrt}[&\{S - R_T \cos(\phi)\}^2 \\ &+\{H_F - (H_T + R_T + R_T \sin(\phi))\}^2]\end{aligned} \tag{A.5}$$

The distance PN to the top of the fire is given by,

$$\begin{aligned}\mathrm{PN} = \mathrm{sqrt}[&\{S + L_F \sin(\theta_F) - R_T \cos(\phi)\}^2 \\ &+\{(H_F + L_F \cos(\theta_F)) - (H_T + R_T + R_T \sin(\phi))\}^2]\end{aligned} \tag{A.6}$$

Therefore, the angles subtended by the tangent to the fire base circle and the fire top circle at the center of the unit hemisphere are given by,

$$\omega_1 = \tan^{-1}\left(\frac{R_F}{\mathrm{PM}}\right) \tag{A.7}$$

and

$$\omega_2 = \tan^{-1}\left(\frac{R_F}{\mathrm{PN}}\right) \tag{A.8}$$

It can also be shown that:

$$\cos(\nu_2 - \nu_1) = \frac{\mathrm{PN}^2 + \mathrm{PM}^2 - L_F^2}{2 \times \mathrm{PM} \times \mathrm{PN}} \tag{A.9}$$

and if:

$$S + L_F \sin(\theta_F) - R_T \cos(\phi) > 0, \text{ then}$$

(A.10a) $\nu_2 = \left[\frac{\pi}{2} - \phi\right] + \sin^{-1}\left[\frac{\{H_F+L_F \cos(\theta_F)\}-\{H_T+R_T+R_T \sin(\phi)\}}{\mathrm{PN}}\right]$

or if $\{S + L_F \sin(\theta_F) - R_T \cos(\phi)\} < 0$, then[2]:

$$\nu_2 = \left[\frac{\pi}{2} - \phi\right] + \pi - \sin^{-1}\left[\frac{\{H_F+L_F \cos(\theta_F)\}-\{H_T+R_T+R_T \sin(\phi)\}}{\mathrm{PN}}\right] \tag{A.10b}$$

Note that if,

$$\nu_1 < 0, \quad \text{then}, \quad \text{we set } \nu_1 = 0 \tag{A.11a}$$

$$\nu_2 > \pi, \quad \text{then}, \quad \text{we set } \nu^2 = \pi \tag{A.11b}$$

The widths of the lower end and upper end, respectively, of the fire shadow on the surface of the unit hemisphere projected on to the base plane of the unit hemisphere are,

$$W_1 = 2\sin(\omega_1) \text{ and } W_2 = 2\sin(\omega_2) \tag{A.12}$$

The extent of the shadow along the elemental plane is:

$$Q_1 - Q_2 = \cos(\nu_1) - \cos(\nu_2) \tag{A.13}$$

The overall view factor between the element and the fire is, therefore,

$$F_{\mathrm{dA}\to\mathrm{Fire}} = \frac{1}{\pi}\left[\frac{W_1 + W_2}{2}\right][Q_1 - Q_2] \tag{A.14}$$

i.e.,

$$F_{\mathrm{dA}\to\mathrm{Fire}} = \frac{1}{\pi}[\sin(\omega_1) + \sin(\omega_2)][\cos(\nu_1) - \cos(\nu_2)] \tag{A.15}$$

In making the above calculation, it is assumed that the shape of the fire shadow projection on the base plane of the hemisphere is that of a trapezium with sharp straight sides. This is certainly not true for all conditions. It is more than likely that the shape will resemble an ellipse or some such rounded figure. However, by making the assumption of a trapezoidal shape, we are overestimating the area of the shadow and hence, the view factor value. The maximum value of the over estimate is by about 27%.

References

[1] NFPA 58, Liquefied Petroleum Gas Code, Table 6.3.1, 2004 ed., National Fire Protection Association, Quincy, MA, 2004.

[2] SFPE, The SFPE Handbook of Fire Protection Engineering, Tables 1–10.2, The National Fire Protection Association, Quincy, MA, CY, 2002 (Chapter 1).

[3] W.M. Rohsenow, H.Y. Choi, Heat, Mass and Momentum Transfer, Prentice Hall Inc., Englewood Cliffs, NJ, 1961.

[4] T. Baumeister, L.S. Marks (Eds.), Standard Handbook for Mechanical Engineers, seventh ed., McGraw Hill Book Company, NY, 1958.

[2] This is because the inverse sine gives results in $-\pi/2 < \sin^{-1}(x) < \pi/2$. In the physical case being analyzed here, the requirement for ν_2 is $0 < \nu_2 < \pi$. Hence, the modification to the equation.

[5] P.K. Raj, Calculations of Thermal Radiation Hazards from LNG Fires: A Review of the State of the Art, Paper No. 2, Session 18, AGA Transmission Conference, St. Louis, MO, May 1977.
[6] H.C. Hottel, A.F. Sarofim, Radiative Heat Transfer, McGraw-Hill Book Co., NY, 1967.

Glossary

BLEVE: boiling liquid expanding vapor explosion
VWW: vapor-wetted (container) steel wall