Emergency Relief Vent Sizing For Fire Emergencies Involving Liquid-Filled Atmospheric Storage Vessels

Hans K. Fauske, Michael Epstein, Michael A. Grolmes and Joseph C. Leung
Fauske & Associates, Inc., 16W070 West 83rd Street, Burr Ridge, Illinois 60521

For large atmospheric vessels the potential occurrence of sufficient liquid swell resulting in two-phase flow is of special importance. Based upon an extension of analytical work it would appear justifiable to ignore two-phase flow effects for non-foamy systems.

INTRODUCTION

Current methods and regulations for sizing emergency relief vents for storage vessels subjected to an external fire are based on all vapor flow [*1*]. However, some recent analytical considerations have raised the issue of two-phase flow [2, 3]. For large atmospheric vessels the potential occurrence of sufficient liquid swell resulting in two-phase flow is of special importance. Since little or no overpressures (< 0.1 psi) can be accommodated in many cases of interest the vent area augmentation due to two-phase flow is to the first order proportional to $(\rho_l/\rho_g)^{1/2}$ where ρ_l and ρ_g are the liquid and vapor densities, respectively. For typical systems an increase in the vent area requirement of about ten to thirty is therefore suggested. Fortunately, as discussed below, based upon an extension of the analytical work described in Reference [2], it would appear justifiable to ignore two-phase flow effects for non-foamy systems. Experiments are summarized which support this conclusion.

ANALYTICAL CONSIDERATIONS

The first attempt to mechanistically describe the void distribution and liquid swell in case of an external fire was presented in Reference [2]. The analysis included a description of the boiling two-phase boundary layer and the resulting strong recirculating flow pattern. However, only hydrodynamic considerations were included in the treatment. As such, when the recirculating flow velocity (U_c) was calculated to exceed the terminal bubble rise velocity (U_∞), vapor bubble carry-under was assumed to take place (see Figure 1). The above assumption led to significant vapor carry-under and hence significant liquid swell relative to the no vapor carry-under case for large storage vessels of interest. The two extreme cases are illustrated in Figure 2.

For example, consider a 2 ft. diameter right circular cylindrical vessel with a liquid height of 3 ft. (H/D ~ 1.5) and fire heat flux of ~ 40 kW/m². The two-phase boiling boundary layer in the absence of vapor carry-under results in an average void fraction of only 0.035 (~ 1.3 inch swell) while with the vapor carry-under assumption the average void fraction becomes 0.13 (~ 5.4 inch swell).*

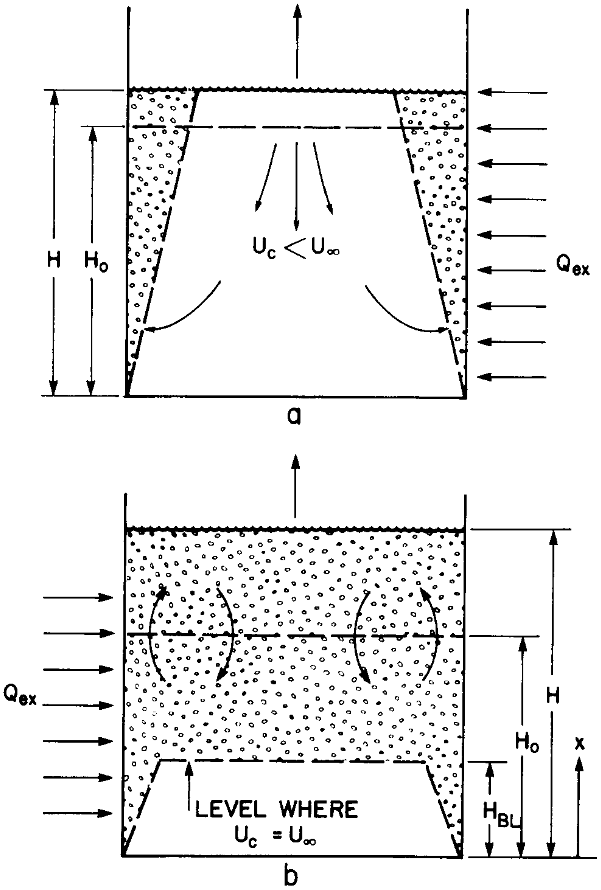

Figure 1. Illustration of the Grolmes-Epstein boiling boundary layer and vapor carry-under model.

The vapor carry-under assumption of Reference [2] has been utilized in Reference [3] to formulate simplified guidelines for sizing relief systems based on two-phase flow.

While the occurrence of vapor carry-under is possible from a hydrodynamic point of view, considerations of static head effects, i.e. the increasing subcooling of the liquid must be considered as the vapor bubbles are dragged under by the recirculating flow. The liquid

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*The average void fraction for the carry-under case is given by [2] $\bar{\alpha} = \hat{\alpha}_{BL} H_{BL}/H_o + \hat{\alpha}_o (1 - H_{BL}/H_o)$
where $\hat{\alpha}_{BL}$ is the average void fraction over the region H_{BL} (the boiling boundary layer height) and $\hat{\alpha}_o$ is the average void fraction over the region $(H_o - H_{BL})$ (the carry-under region). For the above example $H_{BL}/H_o \sim 0.58$.

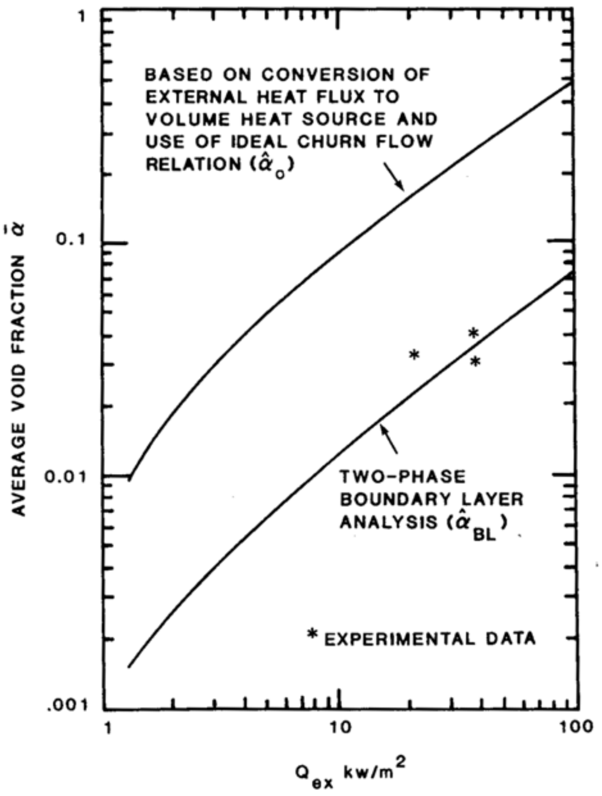

Figure 2. Comparison of average void fraction in vessel for a given external heat flux Q_{ex} depending upon whether: (1) vapor generation is assumed to occur uniformly or (2) vapor generation and motion is confined to a wall boundary layer region H/D = 1.5.

depth, H, resulting in collapse of carry-under vapor bubbles is

$$H \sim \frac{\lambda^2 \rho_g^2 \alpha}{cT\, \rho_l^2\, (1-\alpha)(1-\bar{\alpha})\, g} \tag{1}$$

λ - latent heat of vaporization
c - liquid specific heat
T - saturation temperature
ρ_l - liquid density
ρ_g - vapor density
α - is the void fraction at the top of the liquid pool
$\bar{\alpha}$ - average void fraction
g - gravitational constant

Assuming $\alpha \sim 0.30$ and $\bar{\alpha} \sim 0.15$, H is less than 0.1 m (water as well as organic systems), i.e. the major bulk of the liquid will remain bubble free as a result of the static head effect (see Figure 3).

The above considerations suggest that the liquid swell to the first order is determined by the boiling two-phase boundary layer in absence of vapor carry-under. In the case of vertical tanks on legs, (heating to the bottom as well as the verrtical sides) this suggestion would remain essentially unaltered. The large subcooling present at the bottom of the vessel would result in subcooled nucleate boiling, i.e. on a thin horizontal boiling layer would exist with the heat from the fire being removed largely as sensible heat to the vertical walls by the recirculating flow. Similar arguments can be put forth for horizontal vessels. Experiments supporting this conclusion are summarized below.

EXPERIMENTAL CONSIDERATIONS

A schematic of the experimental configurations tested are shown in Figure 4 and test conditions and results are summarized in Table 1. Noted liquid swells and overpressures in Table 1 were measured at a time when all thermocouples (for locations see Figure 4) read ~ 100°C. Water was used as the test fluid and with the exception of Test Run No. 11, which represented typical non-foamy systems.

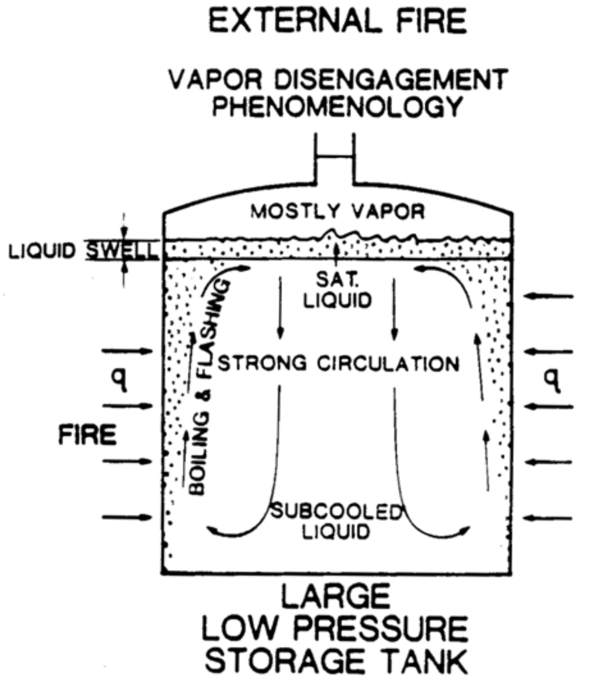

Figure 3. Illustration of expected flow regime pattern for non-reactive systems with external fire.

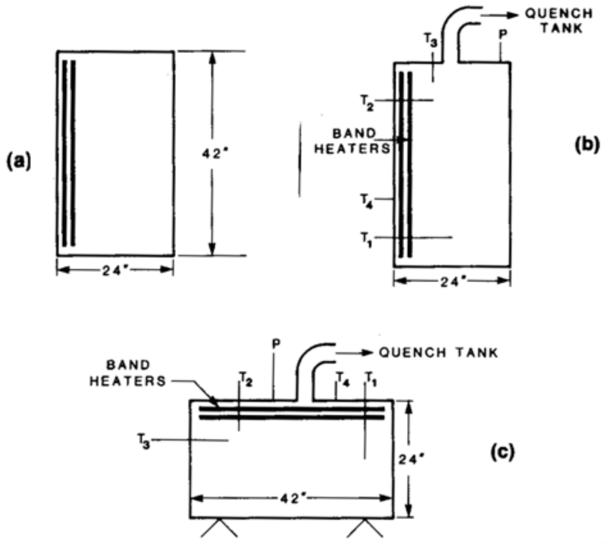

Figure 4. Illustration of vertical and horizontal vessel configurations used for measuring liquid swell and overpressures for fire exposure simulations using electrical band heaters.

Figure 4(a). Completely open vessel used for measuring liquid swell as a function of power level.

Figure 4(b). Restricted opening (see Table 1 for dimensions)—used for measuring overpressure starting out with completely filled vessels.

Figure 4(c). Restricted opening and horizontal vessel configurations used for measuring liquid swell and overpressures for fire exposure simulations using electrical band heaters.

TABLE 1. SUMMARY OF TEST RESULTS

Test No.	Date	Orientation	Vent Configuration	Power Level	Observation
1	(2/27/86)	Vertical*	Open-top	36.5 kW/m^2 (50 heaters)	$\sim$ 1" swell above collapsed height
2	(2/28/86)	Vertical*	Open-top	36.5 kW/m^2	1.5" swell
3	(2/28/86)	Vertical*	Open-top	16.6 kW/m^2 (25 heaters)	$\sim$ 1" swell
4	(3/11/86)	Horizontal**	2" dia. 35 L/D 2 std. elbow $(4\,f\,L/D)_{tot} \approx 2$	35 kW/m^2 (48 heaters)	0.2 psi overpressure
5	(3/12/86)	Horizontal**	Same as Test 4	35 kW/m^2	0.2 psi overpressure
6	(3/13/86)	Horizontal**	1" dia. 70 L/D 1 std. elbow $(4\,f\,L/D)_{tot} \approx 2.2$	35 kW/m^2	2.4 psi overpressure
7	(3/14/86)	Horizontal**	Same as Test 6	35 kW/m^2	1.9 psi overpressure with no water placed in quench tank
8	(3/17/86)	Horizontal**	1" dia. 4" length	35 kW/m^2	0.7 psi overpressure
9	(3/18/86)	Horizontal**	2" dia. 5" length	35 kW/m^2	No overpressure
10	(3/19/86)	Vertical**	2" dia. 5" length	35 kW/m^2	No overpressure
11	(3/20/86)	Vertical**	2" dia. 5" length	35 kW/m^2	0.4 psi overpressure with 1000 ppm detergent in water

*Collapsed liquid height $\sim$ 36 inches.
**Vessel completely filled with water (no free board).

The three first tests in Table 1 were specifically carried out to measure the liquid swell. For this purpose a vertical vessel configuration with open-top was used (see Figure 5) which provided a direct observation of the swell height. In the absence of carry-under the predicted liquid swell is $\sim$ 1.3 inch (see Analytical Considerations) for a power level of $\sim$ 40 kW/m^2. This is in excellent comparison with measured liquid swells of 1 and 1.5 inches (see also Figure 2). Based on carry-under the predicted swell is $\sim$ 5.4 inches. For the $\sim$ 20 kW/m^2 heat flux the boundary layer analysis predicts a liquid swell of $\sim$ 0.7 inch, again in good agreement with the measured value of $\sim$ 1 inch. Based upon these results, there would appear to be no need to alter the current vent sizing practice for atmospheric vertical storage vessels which is based upon all vapor flow as long as the systems can be characterized as non-foamy. The methods summarized in Reference [*1*] would therefore appear adequate for design purposes. This is further substantiated by Test No. 10, using a 2 inch diameter vent. No overpressure was noted with the vessel initially completely filled. Note that the superficial vapor velocity ($\sim$ 20 m/s) for this vent size is somewhat less than the characteristic entrainment velocity ($\sim$ 21 m/s) (see Concluding Remarks).

Five tests were carried out with a horizontal vessel configuration. Significant entrainment and corresponding overpressures were noted in tests using one inch diameter vent lines (Test Numbers 6, 7 and 8). For these tests the superficial velocity ($\sim$ 80 m/s) is well above the characteristic entrainment velocity ($\sim$ 21 m/s). Assuming a vent line vapor quality of 50% (note that this corresponds to a vent line void fraction of $\sim$ 1.0), the noted overpressure can readily be explained by accounting for momentum and frictional losses based upon homogeneous flow conditions.** Again, no overpressure is noted for the short 2 inch diameter vent line, where the superficial vapor vent line velocity is less than the critical entrainment velocity.

Finally, the vertical vessel configuration (Test No. 11) was tested with a foamy system (water with 1000 ppm detergent). In contrast to Test No. 10 (a comparable test with non-foamy system), a significant overpressure was measured and significant liquid entrainment was observed in the exiting vapor stream (approximately two-thirds of the water content was removed in this manner). This test reinforces the need to distinguish between nonfoamy and foamy systems.

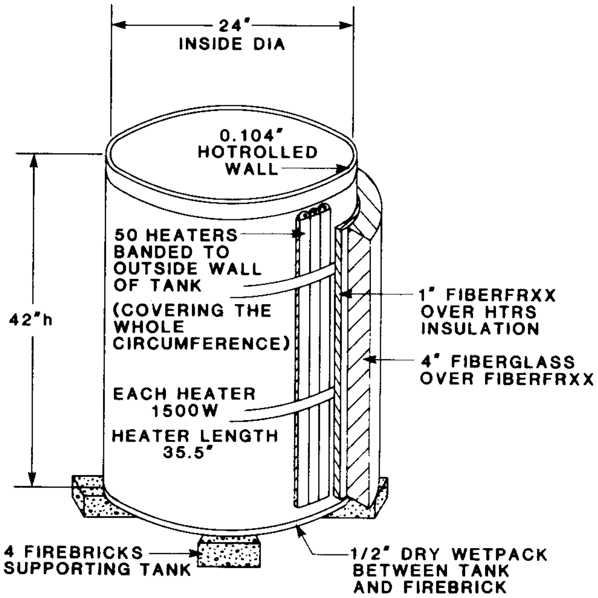

Figure 5. Schematic drawing of the vertical test vessel with open-top.

**A quality of 50% increases the mass flow rate by a factor of two and therefore results in an increase in pressure drop of a factor of four, i.e. the overpressure is rather sensitive to the degree of liquid entrainment although the vent line void fraction remains essentially 100%.

CONCLUDING REMARKS

For liquid-filled atmospheric storage vessels where little or no overpressures can be tolerated a vent size based upon all vapor flow would appear acceptable as long as the vapor velocity in the vent line is kept below the entrainment velocity given by (*4*)

$$U_a \sim 3.0 \left[\frac{\sigma\, g\, \rho_l}{\rho_g^2} \right]^{1/4} \qquad (2)$$

where σ is the liquid surface tension.
The required vent area is then given by

$$A = \frac{Q_T}{\lambda\, \rho_g\, U_a} \qquad (3)$$

where Q_T is the total energy release rate due to the fire.

The above guideline has been verified by experiments simulating fire exposure to vessels (vertical as well as horizontal configurations) completely filled with water (nonfoamy systems). For the water case, the guideline results in a vent area requirement about thirty times less than that suggested if two-phase flow is considered.

ACKNOWLEDGEMENT

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H. K. Fauske, a consultant to firms in the chemical and nuclear industries, earned his D.Sc. from the Norwegian Institute of Technology. The author of more than 100 technical articles, he is a member of the editorial board of the *International Journal of Multiphase Flow.* The first recipient of the Univ. of Chicago Award for Distinguished Performance at Argonne National Laboratory, in 1982 he was presented with the Tommy Thompson Award by the American Nuclear Society.