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ON THE DESIGN OF THERMAL BREATHING DEVICES FOR LIQUID STORAGE TANKS

P. SALATINO*, G. VOLPICELLI* and P. VOLPE**

**Dipartimento di Ingegneria Chimica, Università degli Studi di Napoli 'Federico II', Napoli, Italy*
***Kuwait Raffinazione e Chimica S.p.A., Napoli, Italy*

The dynamic behaviour of an atmospheric liquid storage tank upon sudden change of meteorological conditions is analysed with the aim of establishing criteria for the safe and effective design of thermal breathing devices. A simulation model is presented based on a lumped-parameter multiple-zone representation of the tank. Simplified design criteria based on reasonable and conservative approximations of the governing equations are derived. Application of the simulation procedure and of design criteria is exemplified. Results are discussed and critically compared with prescriptions of existing codes.

Keywords: storage tank; thermal breathing; dynamic simulation.

INTRODUCTION

Venting of atmospheric liquid storage tanks is normally required in order to maintain pressure inside the tank at the atmospheric level upon occurrence of one of the following events:

- feeding/withdrawal of liquid streams to/from the tank;
- changes in the density of gases/vapours in the tank associated with changes in meteorological conditions (either temperature or pressure);
- changes in density of gases/vapours in the tank due to overheating associated with exposure to accidental fires.

General design criteria for venting of storage tanks are provided by the ANSI/API Standard 2000-1992[1]. Prescriptions cover either minimum thermal venting capacities or emergency venting to be provided in case of accidental fire exposure. The prescribed thermal inbreathing capacity is evaluated (Table 2 in Reference 1) on the basis of a maximum estimated rate of heat loss (20 Btu per hour and per square foot of tank surface) upon sudden decrease of the external temperature (as might happen, for instance, after a summer rainstorm). Maximum thermal outbreathing capacity is rated at 60% of the maximum inbreathing capacity. Alternative design criteria for venting devices are provided by the Naumann Formulas, developed by ESSO, and by the German PTB-TRbF Formulas. Hoechst Formulas, reported by Sigel *et al.*[2], take into account the heat capacity of the tank metal enclosures when heat transfer rate between the tank enclosures and the environment is finite. Fullarton and co-workers[3,4] embodied consideration of vapour condensation in the assessment of venting requirements. A draft European normative has recently been developed[5].

All the cited methods are based on the assumption that the enclosures of the gas in the tank are at uniform temperature. In the present paper a design procedure for thermal venting devices for atmospheric liquid storage tanks is presented which takes into account temperature non-uniformity by the recourse to a multiple-zone description of the tank enclosures. The role of vapour condensation on the dynamic behaviour of the tank is also critically considered. A simplified design criterion is obtained on the basis of approximations of the governing equations. Application of the analysis is exemplified and results compared with prescriptions of design criteria provided by the API code[1], by the Naumann and PTB Formulas and by the draft EN standard[5]. Sensitivity of the procedure to selected variables is presented and discussed.

DEVELOPMENT OF A SIMULATION PROCEDURE

The Reference Scenario

A schematic representation of the tank is given in Figure 1. The temperature T_G of bulk gases in the tank has been assumed uniform throughout the vessel at any time. The temperature T_L of the liquid in the tank has been assumed constant over the time and averaged over the period considered.

The following scenario has been assumed as a reference:

Phase 1

The weather is initially hot and sunny. The tank is exposed to solar radiation and to average ambient temperature $T_{A,1}$.

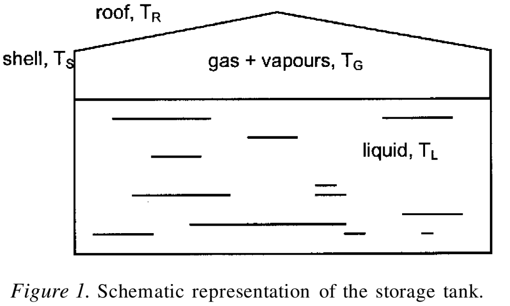

Figure 1. Schematic representation of the storage tank.

Phase 2

At time $t = 0$ weather conditions change. The combined effect of sudden drop of solar radiative flux incident on the tank roof, decrease of ambient temperature to $T_{A,2}$ and strong downpour are assumed to determine stepwise decrease of the enclosure's temperature. The temperature of gases and vapours in the tank starts decreasing, and so does the pressure. As pressure reaches a threshold vacuum level (that depends on inbreathing valves characteristic curve), inbreathing starts and ambient air is admitted into the tank.

Evaluation of Variables Before Change of Weather Conditions

The energy balance over a surface element of the tank enclosure (assuming that temperature is uniform across the metal sheet) yields:

$$\rho_e c_e s \frac{dT}{dt} = ks\nabla^2 T + q\varepsilon + h_A(T_A - T) + h_G(T_G - T) \quad (1)$$

In equation (1) T is the element temperature, $\nabla^2 T$ is its Laplacian, $\rho_e c_e$ and k the heat capacity per unit volume and the thermal conductivity of the metal sheet, respectively, s its thickness, q is the incident radiative flux, ε is the surface emissivity, h_A and h_G the heat transfer coefficients of the element with the ambient and the gases in the tank, respectively. The latter embody contributions from both radiation and convection (either free or forced).

Order of magnitude calculations based on typical parameters of storage tanks suggest that the first term at RHS of equation (1) is much smaller than the others, i.e., conduction along the metal enclosures can usually be neglected. This implies that spatial variation of T is related only to change of parameters $q\varepsilon$, h_A and h_G. Provided that these parameters can be assumed locally uniform, the enclosure can be lumped into multiple zones at piecewise uniform temperature. This approach was followed in the present study, and the enclosures lumped into the *roof* and the *shell*, each characterized by uniform values of parameters T_R, $q_R\varepsilon$, h_{RA}, h_{RG} and T_S, $q_S\varepsilon$, h_{SA}, h_{SG}, respectively. The liquid free-surface, at temperature T_L, completes the enclosure of the gas in the tank.

Analysis of equation (1) further indicates that the term at LHS is typically much smaller than the others as far as heating (or cooling) rates associated with daily excursion of ambient parameters are considered. This implies that, as far as the dynamic behaviour of the tank over the time-scale of a day is concerned, the pseudo-steady state approximation can be invoked.

In the light of the above approximations, the pseudo-steady state roof and shell temperatures can be obtained by solution of the following energy balances:

$$q_R\varepsilon = h_{RA}(T_{R,1} - T_{A,1}) + h_{RG}(T_{R,1} - T_{G,1}) \quad (2)$$

$$q_S\varepsilon = h_{SA}(T_{S,1} - T_{A,1}) + h_{SG}(T_{S,1} - T_{G,1}) \quad (3)$$

The last term in both equations (2) and (3), representing the heat flux from the roof/shell to the gas inside the tank, is generally negligible. Equations (2) and (3) in the unknowns $T_{R,1}$ and $T_{S,1}$ are solved once the radiative heat fluxes absorbed by the roof/shell, the heat transfer coefficients between the roof/shell and ambient air and the ambient temperature $T_{A,1}$ are known.

The energy balance on the gas within the tank, under pseudo-steady state conditions, reads:

$$0 = A_R h_{RG}(T_{G,1} - T_{R,1}) + A_S h_{SG}(T_{G,1} - T_{S,1}) + A_L h_{LG}(T_{G,1} - T_L) \quad (4)$$

where A_R, A_S and A_L are the areas of the surfaces enclosing the gas, i.e. the roof/shell internal surface and the gas-liquid interface. h_{LG} is the heat transfer coefficient between the gas and the bulk of the liquid.

Solution of equation (4) with respect to T_G yields:

$$T_G = \frac{\sum_i A_i h_{iG} T_i}{\sum_i A_i h_{iG}} \quad (5)$$

where summation is extended over all the surfaces enclosing the gas.

Evaluation of Variables After Change of Weather Conditions

At $t = 0$ it is assumed that sudden change of weather conditions associated with: (a) vanishing of radiative flux absorbed by the roof/shell; (b) decrease of ambient temperature and, possibly, (c) downpour, occurs. Under such conditions it is likely that roof/shell temperatures might approach, over a relatively limited time-scale, the wet bulb temperature corresponding to the new ambient values of temperature and humidity. Due to the difficulty of assessing the actual heat transfer rate between the gas enclosures and the environment, it is (conservatively) assumed that stepwise drop of roof/shell temperatures to the values $T_{R,2}$, $T_{S,2}$ occurs upon change of weather conditions. Correspondingly and in the absence of inbreathing the temperature of the gas/vapours in the tank decreases towards an ultimate value $T_{G,2}$ that can be calculated from equations (2), (3) and (5) referring to the new conditions.

In the absence of inbreathing and neglecting condensation of vapours at temperatures below the dew point, the maximum vacuum level in the tank would be:

$$|p_2 - p_1| = p_1\left(1 - \frac{T_{G,2}}{T_{G,1}}\right) \quad (6)$$

The actual vacuum level established in the tank under the combined effect of gas cooling and of inbreathing has the value given by equation (6) as the upper limit. The actual maximum vacuum level in the tank can be calculated by following the dynamic behaviour of the system.

Simulation of the Dynamics of the Tank upon Sudden Change of Boundary Conditions

The energy balance on the gas in the tank, under transient conditions, modifies into:

$$nc\frac{dT_G}{dt} = \sum_i A_i h_{iG}(T_G - T_{i,2}) + nc(T_{A,2} - T_G) + V\frac{dp}{dt} \quad (7)$$

where n is the number of moles of gas/vapour in the tank, V the volume occupied by gas/vapours, c the gas molar specific heat at constant pressure, p the absolute pressure in the tank. Summation in equation (7) is again extended over all the i-th surfaces enclosing the gas. The material balance on gas in the tank yields:

$$\frac{\mathrm{d}n}{\mathrm{d}t} = n \tag{8}$$

where n, the molar flow rate of gas entering the tank via the inbreathing valves, depends on the vacuum level inside the tank according to characteristic curves of the inbreathing valves:

$$n = n(p - p_A) \tag{9}$$

Pressure in the tank is related to the number of moles of gas by the ideal gas equation of state:

$$p = \frac{nRT}{V} \tag{10}$$

Simulation of the transient behaviour requires integration of the two ordinary differential equations (7) and (8), with initial conditions:

$$\text{for } t = 0, \quad T_G = T_{G,1} \quad \text{and} \quad n = \frac{p_1 V}{RT_{G,1}}$$

DEVELOPMENT OF A SIMPLIFIED DESIGN CRITERION

A useful approximation of equation (7) can be developed when considering that the inbreathing valves have to provide enough air for tank pressure not to change appreciably. If the pressure derivative related term is dropped in equation (7), and if the further simplification is made of neglecting the term associated with the enthalpy of incoming air, equation (7) becomes:

$$nc\frac{\mathrm{d}T_G}{\mathrm{d}t} = \sum A_i h_{iG}(T_G - T_{i,2}) = T_G \sum A_i h_{iG} - \sum A_i h_{iG} T_{i,2} \tag{11}$$

or:

$$\tau\frac{\mathrm{d}T_G}{\mathrm{d}t} = T_G - T_{G,2} \tag{12}$$

where $T_{G,2}$ has already been defined and τ is a time constant given by:

$$\tau = \frac{nc}{\sum A_i h_{iG}} = \frac{pV}{RT_G}\frac{c}{\sum A_i h_G} \tag{13}$$

The solution of equation (12) is straightforward and yields:

$$T_G = T_{G,1} - (T_{G,1} - T_{G,2})\left[1 - \exp\left(-\frac{t}{\tau}\right)\right] \tag{14}$$

From this equation the meaning of the characteristic time τ can be appreciated: it is the time-scale over which gas temperature decreases from the value $T_{G,1}$ to approach the ultimate (equilibrium) gas temperature $T_{G,2}$.

The initial rate of change of gas temperature is:

$$\frac{\mathrm{d}T_G}{\mathrm{d}t} = -\frac{1}{\tau}(T_{G,1} - T_{G,2}) \tag{15}$$

The rate of change of pressure in the tank is, from differentiation of the equation of state:

$$\frac{\mathrm{d}p}{\mathrm{d}t} = \frac{RT_G n}{V} + \frac{Rn}{V}\frac{\mathrm{d}T_G}{\mathrm{d}t} = \frac{pQ}{V} + \frac{p}{T_G}\frac{\mathrm{d}T_G}{\mathrm{d}t} \tag{16}$$

where Q is the volumetric flow rate of the inbreathed air. In order to prevent pressure from decreasing, the inbreathing valves must provide at least the air flow rate required to ensure that $\mathrm{d}p/\mathrm{d}t = 0$, that is:

$$Q \geq -\frac{V}{T_G}\frac{\mathrm{d}T_G}{\mathrm{d}t} \tag{17}$$

At the beginning of the cool-down process the inbreathing required is maximum. Particularization of equation (17) at $t = 0$ yields:

$$Q \geq \frac{V}{T_{G,1}}\frac{1}{\tau}(T_{G,1} - T_{G,2}) = \frac{\sum A_i h_{iG}}{c}\frac{R}{p}(T_{G,1} - T_{G,2}) \tag{18}$$

APPLICATION TO DESIGN/VERIFICATION: AN EXAMPLE

Application of the above procedure is exemplified in the following. A cone-roof tank characterized by the following geometrical parameters is considered: diameter: 70 m; height: 15 m; liquid storage capacity: 56,000 m^3; roof tangent: 0.167; total tank volume: 63,000 m^3. The tank is equipped with four breather valves whose characteristic curve is reported in Figure 2. It is assumed that at the time of the sudden change of weather conditions the tank is almost completely empty.

The evaluation of input parameters to the simulation procedure is detailed in the Appendix. It is: q_R = 800 $\mathrm{W m^{-2}}$; $\varepsilon = 0.35$; $T_{A,1}$ = 309 K; T_L = 298 K. Furthermore it is assumed that the radiative heat flux incident on the shell is negligible and that shell temperature can be taken equal to the ambient temperature ($T_{S,1} = T_{A,1}$ = 309 K). h_{RA} in equation (2) sums up contributions from convection, either free or forced, and radiation. The radiative contribution accounts for about 5 $\mathrm{W m^{-2} K^{-1}}$. Free convection regime is assumed and the convective contribution is 5 $\mathrm{W m^{-2} K^{-1}}$. Accordingly h_{RA} = 10 $\mathrm{W m^{-2} K^{-1}}$. Equation (2) yields the equilibrium roof temperature $T_{R,1}$ = 337 K (64°C) before change of weather conditions. Assuming: A_R = 3900 m^2; A_S = 3200 m^2; A_L = 3800 m^2;

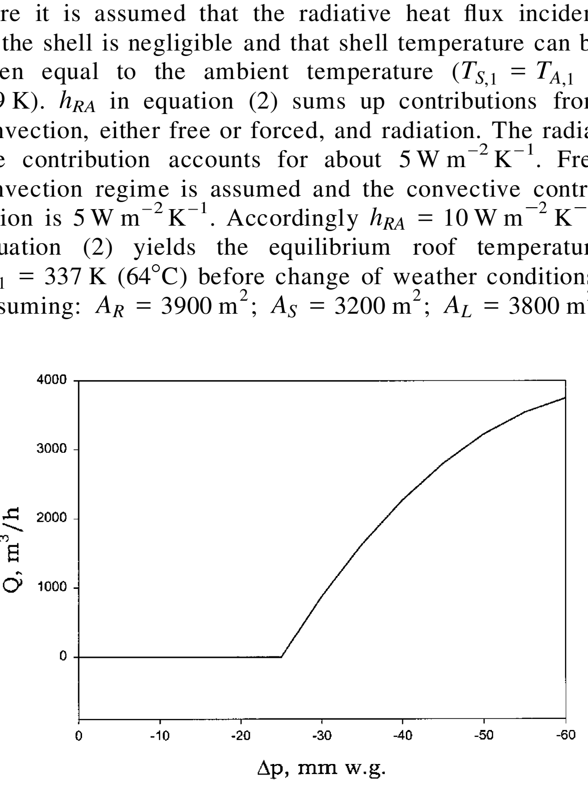

Figure 2. Characteristic curve of inbreathing valves.

$h_{RG} = h_{SG} = h_{LG} = 3\,\mathrm{W\,m^{-2}\,K^{-1}}$, equation (5) yields $T_{G,1} = 315$ K (42°C).

Immediately after the change of weather conditions both ambient and roof/shell temperatures change. It is assumed $T_{R,2} = T_{S,2} = T_{A,2} = 293$ K. From equation (5), and assuming $h_{RG} = h_{SG} = 5\,\mathrm{W\,m^{-2}\,K^{-1}}$; $h_{LG} = 3\,\mathrm{W\,m^{-2}\,K^{-1}}$, it is $T_{G,2} = 294$ K (21°C).

Dynamic Simulation

Results of dynamic simulations are presented in Figures 3 to 6. They report the pressure in the tank (Figure 3), the volumetric flow rate of inbreathed air (Figure 4), the temperature of gas in the tank (Figure 5) and the total heat flux from the roof/shell to the ambient (Figure 6) as functions of time. The equilibration of temperature in the tank with the new ambient temperature takes place over a time interval of about one hour. Table 1 (column 1) reports the main parameters of the calculations, namely the maximum inbreathing rate, the maximum vacuum level and the maximum heat flux from the tank to the atmosphere after change of weather conditions. A maximum vacuum level of about −40 mm w.g. is reached shortly after the beginning of cool-down. Correspondingly, maximum flow rate of inbreathed air is $9600\,\mathrm{m^3\,hr^{-1}}$. Maximum heat flux from the tank to the atmosphere is in the order of $110\,\mathrm{W\,m^{-2}}$. This figure, which corresponds to about 35 BTU/hr sqft, is far larger than the value (20 BTU/hr sqft) that provides the basis for the ANSI/API 2000 standard[1].

Computations were repeated considering the case in which the tank is half filled. Correspondingly, $A_S = 1600\,\mathrm{m^2}$ and the total volume occupied by gas/vapours would be $V = 35{,}000\,\mathrm{m^3}$. Results, summarized in Table 1 (column 2), indicate that maximum vacuum level and inbreathing rate would be about −37 mm w.g. and $8400\,\mathrm{m^3\,hr^{-1}}$, respectively. This situation turns out to be, therefore, less critical than that occurring in the case of an empty tank.

Results reported so far are relative to a computed initial gas temperature of $T_{G,1} = 42$°C, with gas temperature drop after change of weather conditions of 21°C. The API design directives [Appendix A of Reference 1] indicate that temperature drops of roof plates as large as 60°F (33°C) can be observed upon change of weather conditions. In order to check the sensitivity of the model to the gas temperature drop, further calculations have been carried out assuming $T_{G,1} = 50$°C, while leaving the other parameters unchanged. Results are compared with those of the base case in Table 1 (column 3). It can be noted that maximum underpressure rises to −48 mm w.g., while maximum load on inbreathing valves is in the order of $13{,}000\,\mathrm{m^3\,hr^{-1}}$.

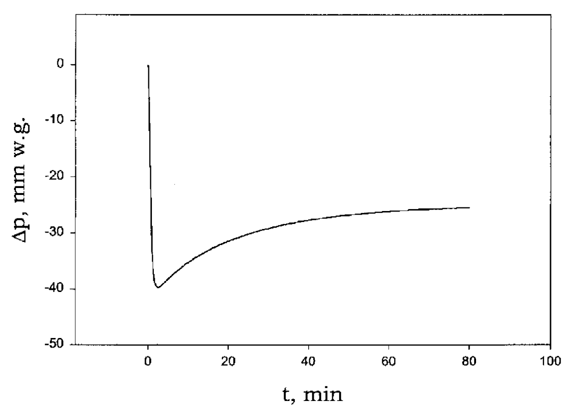

Figure 3. Results of numerical simulation: vacuum level vs time.

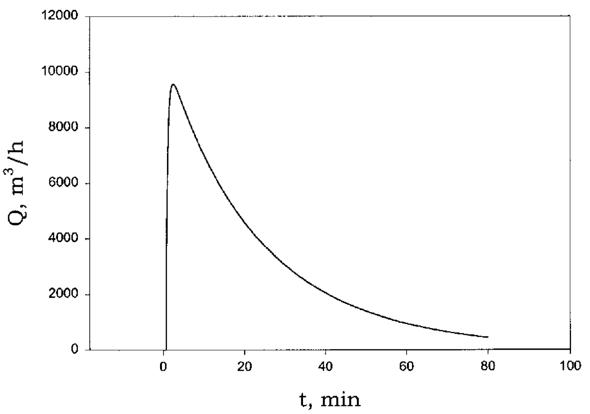

Figure 4. Results of numerical simulation: inbreathed flow rate vs time.

The analysis crucially relies on the knowledge of gas-roof/shell and of gas-liquid heat transfer coefficients, whose determination is at present the most critical aspect of the whole procedure. Values adopted in the calculations were determined on the basis of literature correlations for free-convective heat transfer from flat surfaces[6]. The validity of these equations, however, was extrapolated well beyond the ranges of Nu and Gr numbers that provide their experimental background, for lack of better suited correlations. In order to check the sensitivity of the simulation procedure to the values of the heat transfer coefficient h_{RG}, additional computations were carried out by taking twice the value used in the base case calculation. Results of this sensitivity analysis are reported in Table 1 (column 4). It is observed that maximum vacuum level increases to about −48 mm w.g.. Correspondingly inbreathing demand is raised to more than $13{,}000\,\mathrm{m^3\,hr^{-1}}$.

The Influence of Vapour Condensation

Computations have been developed so far neglecting vapour condensation as roof/shell temperatures are brought below the dew point of the gas/vapour mixture. Fullarton

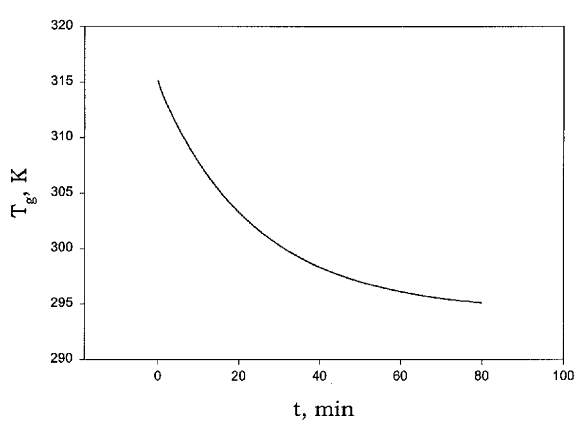

Figure 5. Results of numerical simulation: temperature of gas/vapours in the tank vs time.

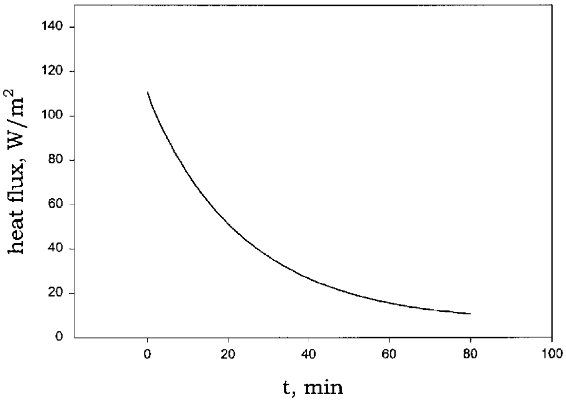

Figure 6. Results of numerical simulation: heat flux from the roof/shell to the atmosphere vs time.

and co-workers[3,4] addressed the influence of vapour condensation on the establishment of vacuum in storage tanks. They recognized the existence of two mutually opposing phenomena, namely the release of the heat of condensation at the wall, which decreases the wall cooling velocity, and the additional volume flow rate associated with the condensation flux. The first effect is ruled out in the present study as the metal enclosure temperature is assumed to change stepwise upon change of weather conditions. Assessment of the maximum extent of vapour condensation is based on the evaluation of the molar fraction y of condensables in the gas phase prior to change of weather conditions. It is worth noting that the value of y should be upperly limited by saturation at the *lowest* temperature of the enclosures (that would also be the upper limit of the gaseous mixture dew point) and not at the *gas* temperature T_G (this has been a common assumption in the literature[2–4]). The maximum amount of condensables present in the gas phase prior to change of weather conditions should be evaluated accordingly. With reference to the case in hand and assuming water vapour as the only condensable specie, the maximum dew point of the mixture would be $T_L = 25°C$ and vapour condensation as gas is cooled to $T_{G,2} = 21°C$ would account for no more than 10% of the total inbreathing demand. By no means should this finding be considered a general conclusion.

In addition to the effects discussed above, some enhancement of heat transfer between gas and the enclosures might be expected as a consequence of condensation: it has been observed[7] that heat transfer can be dramatically augmented by occurrence of phase change even in the presence of only small amounts of condensables. According to equation (18), this would lead to larger inbreathing requirements.

Simplified Design Criterion

The design criterion expressed by equation (18) has been applied to the example. Using temperatures and heat transfer coefficients reported above, the characteristic time for temperature decay is $\tau = 25$ min. Correspondingly the maximum volumetric flow rate of inbreathed air required is $Q = 10{,}000\ m^3\,hr^{-1}$ and is reported in Table 1 (column 1). The figure obtained with the approximated criterion equation (18) compares very satisfactorily (within a 4% error) with that obtained after numerical simulation of the tank dynamics, and it lies on the safe side. Similar computations have been carried out with reference to the other cases considered in Table 1. It is suggested that the design criterion expressed by equation (18), with τ given by equation (13) and T_{G1}, $T_{G,2}$ by equation (5), be used to evaluate the maximum load on inbreathing valves.

When developing a conservative design criterion the most critical case, i.e. that corresponding to the empty tank, is to be considered. Equation (18) can be further simplified if an average value is assumed for the heat transfer coefficients h_{iG}. Accordingly, equation (18) yields:

$$Q = K \cdot A \tag{19}$$

where the constant K takes the value 1 m hr^{-1} if it is assumed $h_{iG} \cong 4\ W\,m^{-2}\,K^{-1}$ and $T_{G1} - T_{G2} \cong 25°C$. The area A of surfaces enclosing gas/vapours can be simply expressed in this case in terms of the volume V of the tank (supposed cylindrical) and of the aspect ratio r between the tank height H and its diameter D. It is:

$$Q = K \cdot V^{2/3} \frac{1 + 2r}{\sqrt[3]{r}} \tag{20}$$

With the above assumptions K takes the value 2.6 m hr^{-1}. Application of equation (20) to the design/verification of inbreathing devices for the tank considered in the case study is exemplified in Table 2. Its results are compared with those obtained from the application of equation (18) as well as from direct numerical simulation. Results obtained from the application of the ANSI/API Code, of the PTB-TrbF and EN Formulas (both based on expressions of the type $Q \propto V^m$, m being 0.71 and 0.70, respectively) and of the Naumann Formulas (based on an expression of the type $Q \propto D^2\ (1 + 4r)$) are also reported for comparison. It can be noted that the ANSI/API prescriptions significantly underestimate the maximum inbreathing demand with respect to results of the present analysis. On the other hand, both the PTB-TrbF and the Naumann Formulas appear to give conservative prescriptions. Application of the draft EN code yields values of the inbreathing demand that can be either smaller or larger than those of the present analysis depending on the value of the constant C adopted in the computation.

Table 1. Results of numerical simulation for the case study.

	1 base case	2 half-filled tank	3 $T_{G,1} = 50°C$	4 $h_{RG} = 10\ W\,m^{-2}\,K^{-1}$
Maximum inbreathing flow rate[1], $m^3\,h^{-1}$	9600 (10,000)	8400 (8650)	13300 (13,800)	13100 (14,500)
Maximum vacuum level, mm w.g.	−40	−37	−48	−48
Maximum heat flux, $W\,m^{-2}$	111	116	150	172

[1] The inbreathing flow rate calculated according to equation (18) is given in parentheses.

Table 2. Maximum thermal inbreathing flow rate for the case study (empty tank) calculated according to different design criteria.

Design criterion	Thermal inbreathing flow rate, $m^3\ hr^{-1}$
ANSI/API Code[1]	4100
Naumann Formula	12,800
PTB-TRbF Formula	11,300
pr EN 265001[5]	6300–10,500
Present work—dynamic simulation	9600
Present work—equation (18)	10,000
Present work—equation (20)	9900

CONCLUSIONS

Detailed simulation of the dynamics of an atmospheric liquid storage tank upon sudden change of weather conditions has been carried out by means of a simple lumped-parameter model. Simplified criteria for the design of inbreathing valves have been developed, based on reasonable and conservative approximations.

The dynamics of a cone-roof liquid storage tank subject to a sudden change of weather conditions has been simulated. Maximum inbreathing capacities and vacuum levels calculated by direct numerical solution of the governing equations compare well with indications coming from the simplified design criteria. The simplified criterion provides a reasonably good, slightly conservative tool for the design of inbreathing devices, easier to use than the direct numerical simulation. Results of the analysis are extremely sensitive to the values of the heat transfer coefficients assumed in the computations. Their evaluation is at present the most critical point of the evaluation procedure.

Analysis of results relative to the case study indicates that rates of heat transfer from the tank to the environment well in excess of 20 BTU/sqft hr (i.e., the value assumed by the ANSI/API Standard[1]) might establish after sudden change in weather conditions. Inbreathing requirements calculated as per the Naumann and PTB Formulas are slightly larger than those predicted with the design criterion developed in this paper. The draft EN prescription[5] might not lie on the safe side. Application of the ANSI/API code appears to be not conservative enough, and consistent oversizing of the thermal breathing devices in respect to its prescriptions should be considered as a safer design measure.

APPENDIX

Evaluation of Heat Fluxes and Heat Transfer Coefficients

- Total radiant heat flux over a horizontal surface: Data published by C.N.R.[8]. The figure corresponds to 42° latitude N, summer, noon.
- Emissivity of the roof surface: 0.35 is emissivity to solar radiation, 0.9 the emissivity for heat flux leaving the surface at $T_R = 50°C$[9]. In either case the roof is assumed to be painted in white.
- Heat transfer coefficient, free convection, horizontal plate, hot-facing up or cold-facing down: $Nu = 0.14\ (Gr\,Pr)^{1/3}$ [6]
- Heat transfer coefficient, free convection, horizontal plate, cold-facing up or hot-facing down: 60% of the value calculated with the previous equation.
- Radiative heat transfer coefficient from the roof to the atmosphere: $h = 4\sigma\varepsilon T_R^3$.

NOMENCLATURE

A	area
c	molar specific heat at constant pressure
c_e	specific heat of wall material
D	tank diameter
h	heat transfer coefficient
H	tank height
k	thermal conductivity of wall material
K	proportionality constant
n	number of moles of gas in the tank
n	molar rate of air inbreathing
p	pressure
q	radiative flux
Q	volumetric flow rate of inbreathed air
r	tank aspect ratio (= H/D)
R	gas constant
s	tank wall thickness
t	time
T	temperature
V	gas volume in the tank
y	molar fraction of condensables in the gas
ε	emissivity
ρ_e	density of wall material
τ	characteristic time for gas temperature decay

Subscripts

A	ambient
G	gas
L	liquid
LG	between liquid and gas
R	roof
RA	between roof and ambient
RG	between roof and gas
S	shell
SA	between shell and ambient
SG	between shell and gas
1,2	before, after change of weather conditions

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ADDRESS

Correspondence concerning this paper should be addressed to Professor P. Salatino, Dipartimento di Ingegneria Chimica, Università degli Studi di Napoli ‘Federico II’, P. le Tecchio 80-80125 Napoli, Italy.

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