(Hughes & Deumaga, 1974)

Impact of weather conditions on the protection of low-pressure and atmospheric storage tanks

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PURPOSE OF THIS PROJECT

Problem Statement and Objectives

Low-pressure and atmospheric storage tanks are protected against atmospheric weather changes following the vent sizing procedures in ISO 28300, EN14015 and in the newest API 2000 7th Edition. These procedures for thermal breathing are identical and more conservative than those highlighted in API 2000 5th Edition, which has been integrated in the latest edition as Annex A.

This project aims to derive and publish a DIERS guideline for the calculation of the in- and out-breathing models for a tank storing non-volatile liquids and non-condensable gases. Special focus will be given to uninsulated storage tanks, which rely on API 2000 7th Ed. for thermal breathing. Tanks with volatile liquids or condensing vapors or cryogenic media are excluded from this study.

Justification and Approach

The major justification for this project is to have a reliable thermodynamic understanding of the phenomenon in order to provide guidance to the users.

A committee consisting of five core members will oversee this project; collect and evaluate available literature in English and foreign languages; collect and analyze the data, and publish a final report.

This report will provide some feedback about the existing sizing methods, described in both the above mentioned standards and in the literature. When literature in languages other than English is referred, the guideline should contain accurate translations.

The results from this research may interest and potentially remarkably affect the work of other committees like API 2000. The session chair will be informed of the contents and purpose of this project and, whenever possible, the results obtained during the development of this project should be presented at their meetings for feedback.

Due to the tight bond between this project and other committees, though this work is developed for and within DIERS, in order to collect and implement their corrections and feedback, the projected duration of this project is estimated at eighteen months.

Duties of all committee members include, but not limited to:

- Execute assigned tasks and gather information and results on time;
- Present the gathered data for group discussion;
- Evaluate each other's results in terms of physicality and application ranges, and
- Write the chapters in the Guideline of this supervision

Specific roles and responsibilities

Member	Roles	Responsibilities and contribution	Advisors
Davide Moncalvo	Project Manager	1. Accounts for project progress 2. Collects and compares existing scientific literature 3. Collects results and questions, and distribute tasks to each team member 4. Develops the final Project Guideline 5. Estimates tank breathing requirements for collapsed tanks using existing and proposed methods	Michael Davies
Georges Melhem	Vice Project Manager	1. Evaluates thermodynamic correctness of models 2. Supports the project manager at API meetings	Dona Chakra
Dilip Das	Thermodynamics expert	1. Provides guideline for key thermodynamic variables 2. Supports Vice Project Manager in the model evaluation	Georges Melhem
Robert Siml	Expert in field	1. Gather data on collapsed tanks and analyze the reason for tank failure	Scott Tipler Don Eure
Greg Hendrickson	Operating Committee Liaison	1. Reports project progress to DIERS and AIChE committees 2. Represents DIERS stake on the project 3. Contributes with his decision in case of project standstill (2 vs 2 vote)	

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TANK BREATHING MODELS IN THE INTERNATIONAL STANDARDS

The International standard commonly used to estimate the tank breathing requirements is (ISO-28300, 2008), which has the status of API 2000 6th Ed. ISO 28300 merges the experience in tank breathing from North America in API 2000 5th Ed. (API-2000-V, 1998) and from Europe in (EN-14015, 2005). Since a more recent API 2000 7th Ed. (API-2000-VII, 2014) has been published, this new edition will be referenced here throughout the text and further reference to ISO 28300, when not strictly necessary, will be omitted.

API 2000 7th Edition (2014)

The scope of API 2000 is to cover the normal and emergency venting requirements for aboveground liquid petroleum – crude or products – tanks designed to operate at a pressure below 1.034 bar-g or 15 psig. Sound engineering judgement shall be used to determine whenever this standard is applicable to tanks containing other liquids.

For non-refrigerated aboveground tanks, which this report is intended for, the standard provides detailed calculations for pressure and vacuum relief requirements for liquid movement, e.g. pumping of liquid into or from the tank, and for weather changes. Overpressure and vacuum, due to liquid movement for a tank filled with a non-volatile liquid, are respectively equal to the maximum volumetric filling rate and the maximum discharge rate. For flashing liquids, where the vapor pressure of the entering stream is greater than the operating pressure of the tank or liquids containing dissolved gases, an equilibrium flash calculation shall be performed. Overpressure or vacuum related to weather changes is the expansion or contraction in the gaseous space inside the tanks caused by for instance an increase in solar radiation or in wind velocity, or the insurgence of precipitation. However, weather changes may also result in liquid evaporation or vapor condensation, which may dramatically increase the breathing requirements for volatile liquids. However, breathing requirements, due to phase change inside the tanks, are not part of this report. Furthermore, vapor condensation is one of the reasons why API 2000 7th Ed. suggests to perform an engineering study for uninsulated hot tanks with vapor space temperatures exceeding 120°F or containing steam. The relief requirements, due to weather changes, are referred in API 2000 and throughout this document as thermal breathing. Relief requirements from liquid movement and weather changes are also referred as normal breathing to distinguish them from the emergency venting when the tank is exposed to fire or others. Emergency relief breathing requirements are not part of this study.

This report is focused on the relief requirements due to weather changes, which are referred in API 2000 and throughout this document as thermal breathing. There has been a substantial change in the determination of the thermal breathing requirements in the main body of API 2000 5th and 7th Edition, the former being included in the latter as Annex A. Both procedures are described below.

Table 1: Definition of normal and standard condition as per API 2000 7th Edition

	Temperature	Pressure
Normal condition	0°C	101.3 kPa
Standard condition	60°F	14.7 psi

Thermal breathing in the main body of API 2000 7th Edition (2014)

Thermal breathing is caused by atmospheric heating or cooling of the external surfaces of tank shell and roof. The thermal breathing requirements throughout API 2000 are expressed in normal cubic meters per hour of air in SI units or in standard cubic feet per hour of air in USCS units. The normal and standard conditions are specified in Table 1.

The thermal outbreathing as maximum thermal flow rate due to wall heating is expressed by Eq. (1)

$$\dot{V}_{OT} = Y \cdot {V_{tk}}^{0.9} \cdot R_i \quad (SI\ unit) \qquad \dot{V}_{OT} = 1.51\ \cdot Y \cdot {V_{tk}}^{0.9} \cdot R_i \quad (USCS\ unit) \qquad (1)$$

with V_{tk} being the tank volume in cubic meters (SI) or cubic feet (USCS), while Y and R_i are two dimensionless factors, respectively. the latitude factor and the reduction factor for insulation, identical in both equations. The latitude factor Y depends on latitude as given in Table 2.

The thermal inbreathing as maximum thermal flow rate due to wall heating is expressed by Eq. (2)

$$\dot{V}_{IT} = C \cdot {V_{tk}}^{0.7} \cdot R_i \quad (SI\ unit) \qquad \dot{V}_{IT} = 3.08\ \cdot C \cdot {V_{tk}}^{0.7} \cdot R_i \quad (USCS\ unit) \quad (2)$$

with V_{tk} being the tank volume in cubic meters (SI) or cubic feet (USCS), while C is a more complex factor, depending on latitude, average storage temperature, and vapor pressure, as given in Table 3. The insulation factor R_i is identical to that described for thermal outbreathing.

Table 2 Latitude factors Y

Latitude	Examples of locations	Latitude factor, Y
Below 42°	Most of US, most of Latin America and PRC, the whole Arabian Peninsula, Indian subcontinent and Africa	0.32
Between 42° and 58°	Most of Continental Europe, lower half of Canada	0.25
Above 58°	Norway, most of Russia, Alaska	0.2

Table 3 Thermal inbreathing factor C

Latitude	Thermal inbreathing Factor C			
	Vapor pressure similar to Hexane		Vapor pressure higher than Hexane or unknown	
	Average storage temperature			
	< 25°C	≥ 25°C	< 25°C	≥ 25°C
Below 42°	4	6.5	6.5	6.5
Between 42° and 58°	3	5	5	5
Above 58°	2.5	4	4	4

In case of uninsulated tanks exposed to a rapid temperature drop of 40°C (72°F) or more, the thermal inbreathing rates may be larger than those given here, and an individual case-by-case study may be required.

The insulation factor[1] is given by Eq. (3a) as a function of the insulation factor for fully insulated tanks R_{in} and the ratio of the insulated surface area A_{inp} over the total surface area A_{TTS}. Both surfaces are expressed in square meters in SI units and square feet in USCS units. It can easily be proven that R_i goes from a minimum value of R_{in} for fully insulated tanks to one for uninsulated tanks

$$R_i = R_{in} \cdot A_{inp}/A_{TTS} + 1 - A_{inp}/A_{TTS} \quad (3a)$$

The insulation factor for fully insulated tanks R_{in} is given in Eq. (3b) as a function of the inside heat transfer coefficient h, the insulation thickness l_{in} and, the thermal conductivity of the insulation λ_{in}. API 2000 states that an inside heat transfer coefficient of 4 W/m²K or 0.7 BTU/(h ft² °F) is commonly assumed for typical tanks. The insulation thickness l_{in} and the thermal conductivity λ_{in} are expressed in meters and W/(m K) in SI units or in feet and BTU/(h ft °F) in USCS units. From Eq. (3b) thermal breathing can be minimized by either increasing the thickness of the insulation or selecting an insulator with a very poor thermal conductivity.

$$R_{in} = 1/[1 + h \cdot l_{in}/\lambda_{in}] \quad (3b)$$

There is also a reduction factor also for the so-called “double wall tanks”, here meaning the same as a tank inside a containment tank. The formula in API 2000, reported in Eq. (3c), originated from (EN-14015, 2005). In that equation ,A_c is the shell and roof area placed outside the containment tank.

$$R_i = 0.25 + 0.75 \; A_c/A_{TTS} \quad (3c)$$

[1] In API 2000 7th Edition the insulation factor for partially insulated tanks is called R_{inp} and R_i is equal to R_{inp} for partially insulated tanks and to R_{in} for fully insulated tanks. Eq. (3a) means exactly the same.

Assumptions in API 2000 7th Edition Main Body method

The model presented in the main body of API 2000 7th Edition relies on the following assumptions and boundary conditions, explicitly enumerated in API 2000 Annex E and inherited from the method derived by Förster, et al. (Förster H., 1984), which is presented in more detail in a separate chapter of this report.

The tank is assumed fully filled with vapors or gases, meaning there is no liquid inside the tank. Concerning its shape and construction, the minimum wall thickness is as given in DIN 4119, and if it has a conical roof, the minimum roof angle is 15°. Concerning the thermodynamics, there is no heat flux to the tank bottom as well as no other heat capacity besides that of the walls. The emission ratio for the wall radiation is assumed as 0.6 on the basis of data for dirty bronze aluminum.

Additionally, the following assumptions apply to tank heating for thermal outbreathing. The tank product is air, and as an initial condition both the tank walls and the content are assumed at 15°C. Furthermore, free convection both inside and outside the tank has a heat transfer coefficient of 2 W/(m² K), and the sun radiation starts at the maximum level and remains constant. The ambient temperature remains constant until the maximum outbreathing flow rate is reached.

In addition, the following assumptions apply to tank cooling for thermal inbreathing. The tank is filled with air, and as an initial condition, the tank walls and the content are assumed at 55°C. Cooling starts instantly when the tank is soaked by a constant rain intensity of 225 kg/(m² h) intensity and a rain angle of 30°. Rain temperature is the same as ambient at 15°C, and the heat transfer coefficient between the rain film and ambient is 15 W/(m² K). The rain film on the outer surface of the walls has a heat transfer coefficient of 5000 W/(m² K), while free convection between the content and the inner side of the walls has a heat transfer coefficient of 5 W/(m² K). Air cooling, due to the inbreathing of colder ambient air into the tank, is neglected.

The calculation relies on the assumption that the temperature of the tank contents and the walls are assumed uniform and that the heat transfer coefficients are temperature independent. Atmospheric pressure fluctuations are neglected as well as the fact that vents start opening at a certain set point.

API 2000 7th Edition Annex A

This Annex contains a calculation method to determine the normal venting requirements, which include at least liquid movement and thermal effects for a tank satisfying the conditions below:

A. The volume is less than 180 000 bbl.
B. The maximum operating temperature of the vapor space is 120°F (48.9°C)
C. The tank is uninsulated
D. The temperatures of the tank and the feeding lines are below the boiling point at the maximum operating pressure of the tank.

Annex A states that if a tank does not satisfy all of the above mentioned requirements, the general sizing model in the Main Body should be applied. The choice to adopt Annex A for determining the breathing requirements for existing and new tanks is the responsibility of the user. Re-evaluation of existing tanks, even if changing service, may still use the method in Annex A, provided that all the above mentioned requirements are satisfied and some rationale for that is provided.

According to Annex A, a petroleum product with a flash point below 100°F is considered volatile. If the flash point is not available, a liquid is then considered volatile if its atmospheric boiling point is below 300°F. The in- and out-breathing rates due to liquid movement in Table 4 are expressed in an equivalent air flow in either Nm^3/h per m^3/h of pumped liquid (SI units) or in SCFH per bbl./h of pumped product (USCS units). In comparison, the outbreathing flow rate in the Main Body of API 2000 7th Ed. is the volumetric flow rate of vapors in the actual pressure and temperature of the tank, which is equal to the maximum volumetric filling rate for non-volatile liquids and twice as much for volatile liquids. The inbreathing flow rate in the Main Body is equal to the maximum rate of liquid discharge. Both the breathing flow rates and the maximum liquid rates in the Main Body are expressed in the SI units of m^3/h. If the pumping rates are expressed in gpm and the breathing flow rates in SCFH, then the breathing rates should be multiplied by a factor of 8.02. The user should also convert the outbreathing flow rate to an air equivalent flow rate when the actual temperature in the tank exceeds 120°F.

Table 4 In- and outbreathing in SI units (USCS units) given in Annex A

<table>
<tr><th></th><th colspan="2">Inbreathing</th><th colspan="2">Outbreathing</th></tr>
<tr><td>Flash point ≥ 100°F</td><td rowspan="4">0.94 (SI)</td><td rowspan="4">5.6 (USCS)</td><td rowspan="2">1.01 (SI)</td><td rowspan="2">6 (USCS)</td></tr>
<tr><td>Boiling point ≥ 300°F</td></tr>
<tr><td>Flash point < 100°F</td><td rowspan="2">2.02 (SI)</td><td rowspan="2">12 (USCS)</td></tr>
<tr><td>Boiling point < 300°F</td></tr>
</table>

Thermal breathing in API 2000 7th Edition Annex A (2014)

In Annex A, the rapid cooling in the tank vapor space caused by rapid weather changes is the controlling case for thermal breathing. The heat transfer rate is the highest for uninsulated hot tanks with little or no liquid inventory, which may be referenced in this report as “empty” tanks. For southwestern USA, Annex A reports that roof plates can be cooled by as much as 60°R, while shell plates by about 30°R.

For tanks with a volume of less or about 20 000 bbl., the inbreathing requirements due to thermal contraction are about 1 SCFH per barrel of empty tank volume under the assumption of a maximum cooling rate of 100°R/h in the vapor space and a maximum initial temperature of 120°F. For tanks with a volume greater than 20 000 but less than 180 000 bbl., the inbreathing requirements due to thermal contraction are about 2 SCFH each ft^2 of exposed surface area under the assumption of a maximum heat transfer rate of 20 BTU/(h ft^2). The thermal outbreathing requirement caused by vapor expansion is identical to the thermal inbreathing for volatile liquids and 60% of the thermal inbreathing for non-volatile liquids.

A theoretical or experimental validation for the approach in Annex A is neither mentioned in the standard nor could be found in the literature

Comparison of thermal breathing between API 2000 7th Ed. Main Body and Annex A

In this paragraph, the thermal in- and outbreathing requirements for storage tanks of increasing volume are calculated following the procedures in API 2000 7th Edition Main Body or Annex A. All requirements for the application of Annex A are satisfied, e. g. air filled tank at a temperature below 120°F. The thermal inbreathing in Fig (1) is calculated using the C factors for tanks stored in Continental Europe, and it shows that the procedure in Annex A always results in smaller requirements than following the method in the Main Body. For tanks of very small volume, e.g. less than 10 m^3, the difference between the two inbreathing requirements is almost an order of magnitude. However, whatever calculation is chosen, for these small tanks even at the minimum design vacuum of -1 in w.c. as per (API-650, 2014), a 2” manhole or vacuum breather is usually oversized.

The difference in the estimation of the thermal outbreathing requirements using the Main Body and Annex A in Fig. (2) is more modest: at first glance, it seems that the outbreathing requirement for volatile liquids in Annex A lies between those of the Main Body calculated using latitude factors of 0.25 and 0.32.

A comparison between the calculation procedures in the Main Body and Annex A clearly shows that inbreathing is where the largest discrepancies are observed, and it needs to be analyzed in more detail. Therefore, it was decided to study the existing scientific literature and determine which other approaches exist and how they have been validated in order to derive the basic requirements for this guideline.

Figure 1 Thermal Inbreathing in in Nm³/h$_{air}$ using API 2000 Annex A and Main Body for Continental Europe

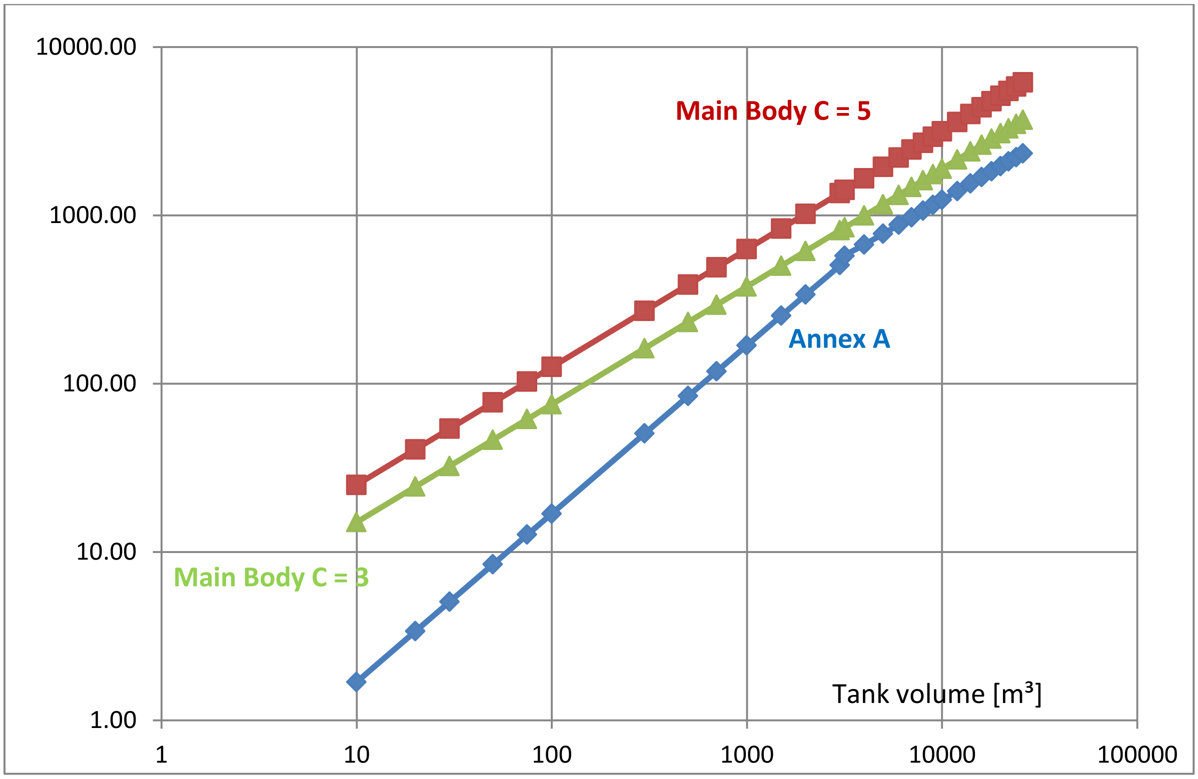
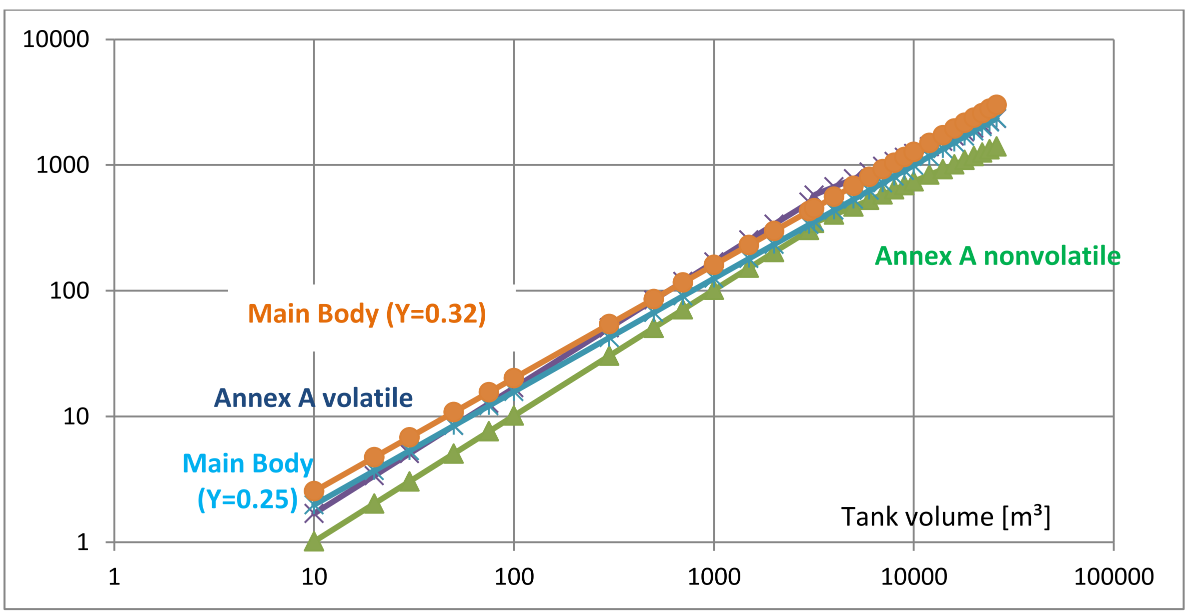


Figure 2 Thermal Outbreathing in Nm³/h$_{air}$ using API 2000 Main Body and Annex A



Tank breathing models in the state of the art scientific literature

It has already been referenced that the method on which the Main Body of API 2000 relies was proposed by (Förster H., 1984) as PTB Report W-22 in German, and it is commonly referenced as the PTB method.

Föster et al. (PTB METHOD)

The PTB Report W-22 investigates the thermal flow rates to prevent collapse of a tank, which is exposed to prolonged solar radiation or a sudden rainstorm.

Even under the assumption of steady thermal radiation on an exposed uninsulated tank, the temperature inside the tank is three dimensional and time dependent. In an empty, large and flat tank, exposed to a steady ambient condition, gas stratification may occur after a long period of time. Given the target of estimating the breathing flow rates, the authors chose to neglect any inhomogeneity of the product temperature T_B and describe only the changes over time. Similarly, their method neglects any heat transfer within the walls due to the thermal gradient between hotter and colder surfaces so that a uniform time dependent wall temperature T_E is assumed.

Furthermore, evaporation from the liquid surface, as well as condensation of the vapors, is ignored. Hence, the only effect of the temperature change in the gas volume is an expansion of the gas space if heated or compression if cooled. On behalf of the ideal gas law, this concept is well expressed in Eq. (6a). From that equation, the breathing flow rate $\dot{V}$ occurs at the point of maximum change of the volume occupied by the gaseous products, V, and it is proportional to it.

$$\dot{V} = dV/dt = V/T_B \cdot dT_B/dt \quad (6a)$$

Since the volume of the gaseous space V in a partially filled tank is always smaller or at most equal to the tank volume V_B , the maximum breathing is given by Eq. (6b). Both the maximum in- and outbreathing, called here $\dot{V}_{in}$ and $\dot{V}_{out}$, occur at the moment of maximum temperature change.

$$\max(\dot{V}) = V_B/T_B \cdot dT_B/dt \quad (6b)$$

The symbols in Table 5 are taken from the original publication and should be adopted only within this context.

Table 5 List of Symbols used in PTB W-22

Symbol	Meaning	Units
c_W	Specific heat of rain	J/(kg K)
$\tilde{c}_p$	Molar air specific heat	J/(kg K)
d_W	Average rain film thickness	m
y_m	Maximum geometrical factor	--
$\dot{m}_e$	Rain mass flux	kg/(m² s)
p_v	Vapor pressure	Pa
t	Time	s

Symbol	Meaning	Units
A	Radiative power of a black body per unit area at 300K	W/m^2
C_B	Tank product heat capacity	J/K
C_E	Tank walls heat capacity	J/K
D	Tank diameter	m
F	Exposed tank surface	m^2
H	Tank height	m
Q_1	Heat transferred via solar radiation	W
$\tilde{H}_v$	Molar latent heat of water (rain)	J/mol
Q_2	Heat reflected from tank surface	W
T_B	Temperature of tank products	K
T_E	Temperature of tank walls	K
T_U	Ambient temperature	K
T_W	Temperature of rain film	K
T_{Wa}	Temperature of rain drops	K
V	Volume of gaseous phase	m^3
V_B	Volume of tank	.m^3

Symbol	Meaning	Units
α_B	Internal heat transfer coefficient	$W/(m^2K)$
α_S	Radiative power of a black body per unit area and temperature	$W/(m^2K)$
α_U	External heat transfer coefficient	$W/(m^2K)$
α_W	Heat transfer coefficient between tank walls and rain film	$W/(m^2K)$
ε_1	Absorption coefficient of tank walls	--
ε_2	Emissivity coefficient of tank walls	--

Outbreathing requirements

From Eq. (6b), the outbreathing requirements are determined at the peak of temperature gradient inside the gaseous space in the tank due to heat transfer by convection from the walls, as shown in Eq. (7)

$$\alpha_B \, F \, (T_E - T_B) = C_B \cdot dT_B/dt \quad (7)$$

The walls themselves are irradiated by solar radiation Q_1 ; however, they lose part of this incident radiation to the colder environment by both radiation Q_2 and convection, as presented in Eq. (8).

$$Q_1 - Q_2 - \alpha_U \, F \, (T_E - T_U) - \alpha_B \, F \, (T_E - T_B) = C_E \cdot dT_E/dt \quad (8)$$

For a conservative estimation of the outbreathing requirements, the absorbed irradiation heat Q_1 is maximized and the reflected heat Q_2 minimized, see Eq. (9a) and (9b). The incident radiative heat is the product of the maximum energy flux on the earth surface, which the authors take from (Kuczera & Gunther, 1975), and the tank wall absorption coefficient ε_1. The absorption coefficient rises with the degree of dirt on the painted surface from about 0.4 for white or aluminum surfaces to 0.96 for a black painted tank as given in (VDI Wärmeatlas, 1974). The geometrical factor y_m considers that flatter and shorter tanks expose more area to solar radiation than taller and thinner ones of similar volume mostly due to exposure of the roof: values for y_m are expected to range between 0.3 and 0.57 as H/D decreases.

$$Q_1 \leq \max Q_1 = I_{H0} \, \varepsilon_1 \, y_m \, F \quad (9a)$$

Similarly, the maximum radiative heat emitted by the surface is the product of an emissivity coefficient ε_2, which decreases from 0.96 for newly painted tank surfaces to 0.4 for bare metal walls, and the specific radiative power of the walls. The walls specific radiative power is determined from the radiative power of a black body at a reference ambient temperature of 300K, named A, corrected with an additional term accounting for the increasing tank wall temperature.

$$Q_2 \geq \min Q_2 = \varepsilon_2 \, F \, [0.5 \, A + 0.75 \, \alpha_S (T_E - T_U)] \quad (9b)$$

The system of differential equations Eq. (7) and (8) can be solved analytically on behalf of Eq. (9a) and (9b) for the temperatures T_B and T_E if the thermodynamic quantities are assumed constant. In that context, the temperature T_B is obtained as an exponential function of time, which tends to an asymptotical maximum value, given in Eq. (10).

$$\max T_B = \lim_{t \to \infty} T_B(t) = T_U + (I_{H0} \, \varepsilon_1 \, y_m - 0.5 \varepsilon_2 A)/(\alpha_U + 0.75 \varepsilon_2 \alpha_S) \quad (10)$$

Simplified approach in function of the tank geometry

The analytical model can be simplified for cylindrical tanks with conical roofs constructed in agreement with (DIN-4119, 1979). The tank volume V_B and lateral surface F are given in Eq. (11) as a function of the diameter D and the cylindrical shell height H.

$$V_B = \pi/4 \; D^2 H + 0.02603 \, D^3 \qquad F = \pi D H + \pi \, 0.025434 \, D^2 \quad (11)$$

The minimal wall thickness has been set by the authors to 4 mm, which is thinner than the minimum required in DIN 4119 in Table 6. The choice of a thinner wall assumes smaller thermal capacity and, therefore, a conservatively larger maximum temperature gradient dT_B/dt. The tank contains only air at an initial temperature of 15°C. In order to maximize the radiation heat into the tank products, an aluminum coating with both ε_1 and ε_2 equal to 0.6 was chosen by the authors, which explains why these values are mentioned in (API-2000-VII, 2014) Annex E. The convective heat transfer coefficients to the walls α_U and α_B have been placed equal to 2 W/(m² K). This assumption is conservative for α_U in a range of tank volumes between 2 and 10 000 m³ and for ratios H/D between 0.2 and 2. The assumption of larger values for α_B, though conservative, is disclaimed by the authors as non-meaningful for temperature differences between walls and either product or ambient of less than 5K. The assumption of a white coating is less conservative than aluminum with increasing height over diameter H/D ratios. For the specified values of: absorption and emission coefficients ε_1 and ε_2 ; convective heat transfers α_U and α_B ; height over diameter ratio H/D ≥ 0.2, and for tank volumes between 10 and 10 000 m³, the maximum outbreathing flow rate can be estimated using Eq. (12). The similarity between Eq. (12) and Eq. (1) for Central-European latitudes is astonishing: the two formulae are identical for a height over diameter H/D ratio of about 0.5.

$$\max \dot{V}_{out} = 0.171 \cdot (H/D)^{-0.52} \cdot {V_B}^{0.89} \quad (12)$$

It should be noted that Eq. (12) has been integrally adopted within (TRGS-509, 2014).

Table 6 Minimal tank wall thickness as in DIN 4119.

Diameter range		Minimal wall thickness
Minimum tank diameter [m]	**Maximum tank diameter [m]**	**[mm]**
	30	5
30	40	6
40	50	7
50	60	8
60		9

Inbreathing requirements

From Eq. (6b) the maximum air inbreathing, to compensate for the gas volumetric contraction, is proportional to the maximum rate of cooling inside the tank. The authors approach neglects the cooling effect brought by the breathing of colder ambient air into the tank. This effect has been considered in recent publications as (Moncalvo, Davies, Weber, & Scholz, 2016). The energy equation for the tank content is identical to Eq. (7), as any cooling of the stored product occurs due to convective heat transfer to the colder tank shall and roof if the tank base is assumed as insulated. The energy equation for the tank walls in Eq. (13) differs from that in Eq. (8) as heat is removed from the stored product and transferred to the rain film.

$$\alpha_W\ F\ (T_E - T_W) - \alpha_B\ F\ (T_B - T_E) = -\ C_E \cdot dT_E/dt \quad (13)$$

In comparison to thermal outbreathing, where a system of two linear differential equations was satisfactory to determine the temperature at the walls and inside the tank, the modeling of the rain film on the tank shell is necessary to determine the thermal inbreathing. The cold rain film is generally warmed up by the heat subtracted from the tank, though some of this heat flow is transferred to the environment and to the colder raindrops impinging the film. Furthermore, the thin rain film on the broad tank wall surface may also be subject to some evaporation. All these phenomena are modeled in Eq. (14). Film evaporation is modeled as a function of steam pressure $p_{v\text{-}H2O}$, as shown in Eq. (15).

$$-\alpha_W\ F\ (T_E - T_W) + \dot{m}_e c_W F\ (T_W - T_{Wa}) +\ \alpha_U F(T_W - T_U) + F \cdot W_v = -c_W \rho_W d_W F\ dT_W/dt \quad (14)$$

$$W_v = \alpha_U \cdot \widetilde{H}_{v-H2O}/\tilde{c}_{p-air} \cdot \lfloor p_{v-H2O}(T_W) - p_{v-H2O}(T_U)]/[p_B - p_{v-H2O}(T_W)] \quad (15)$$

The initial temperature for both the tank walls T_{E0} and the stored product (air) T_{B0} is 55°C, while the ambient temperature is assumed at 15°C. For these values the internal heat transfer coefficient α_B is 5 W/(m²K), while the coefficient between the tank walls and the rain film α_W is 5000 W/(m² K) according to (VDI Wärmeatlas, 1974). The heat transfer coefficient between rain and the environment α_u is 15 W/(m²K) under the assumption of a wind velocity of 10 m/s. PTB W-22 reports some worst case scenarios for the rain intensity in Table (7) as referenced from the German Meteorological Service Deutscher Wetterdienst (www.dwd.de). According to this source, a rain mass flux of 225 kg/(m²h), which is also referenced in Annex E of API 2000 7th Ed., is an extreme rainstorm eventually occurring in Europe once a century. An estimated rain incidence angle on ground is also given and defined as the ratio between the wind velocity and the precipitation velocity of the rain drops. The more intense the rainfall, the heavier the individual droplets and the lower is the impingement angle. According to (Snyers, 1979), incidence angles between 30° and 90° are plausible. In general, as shown in the table, the simultaneous occurrence of heavy rain with large mass fluxes and large impendence angles, e. g. large wind velocities, seems to be rare. Rain film thickness increases with the tank size, and it is expected to be between 0.1 and 1 millimeter.

Table 7 Worst case rain flux of at least ten minutes duration for Central Europe according to Deutscher Wetterdienst.

Scenario	Rain Mass flux [kg/m² h]	Incidence angle [°]	Casuality [annum^{-1}]	Overall rain heat transfer coefficient for thin films $\tilde{\alpha}_W$ [W/(m²K)]
1	75	90	1	86
2	150	60	1/20	169
3	225	30	1/100	249

For the system of differential equations Eq. (7), (13) and (14), a numerical solution is generally available due to the nonlinear relationship between vapor pressure and temperature. The analytical solution for that system of differential equations is possible only in few cases when the evaporation term W_v in Eq. (15) is either linearized in function of temperature or suppressed. The latter is, for instance, the case of an infinite rain film thickness. For this specific case, using few approximations, the maximum inbreathing flow rate of an air filled tank can be estimated using the formula in Eq. (16).

$$\max \dot{V}_{in} = \alpha_B \, F \cdot (T_{B0} - T_{Wa})/T_{B0} \cdot 1/(\rho_B c_B) \quad (16)$$

Another case where film evaporation can be neglected is when the film thickness is very small. In that case, the energy equation for the rain film can be simplified into the linear relationship of Eq. (17). In that equation, the energy balance for the stored product is rewritten in terms of the temperature difference with the rain droplets, $\theta = T_B - T_{Wa}$, as shown in Eq. (18), using the overall rain heat transfer coefficient $\tilde{\alpha}_W$ from Eq. (19). The overall coefficient $\tilde{\alpha}_W$ is the inverse of the thermal resistance of the rain film which, in this case, consists of a convective and a capacitance term. The influence of the rain mass flux $\dot{m}_e$ on the overall rain heat transfer coefficient $\tilde{\alpha}_W$ is shown in Table 7. In this example, the heat transfer coefficient between the rain film and the walls α_W is assumed as 5000 W/(m² K) as given by the authors and specified in ISO 28300. Here, it is clear that the overall coefficient $\tilde{\alpha}_W$ is at least one order of magnitude smaller than α_W.

$$-\alpha_W \, F \, (T_E - T_W) + \dot{m}_e c_W F \, (T_W - T_{Wa}) = 0 \quad (17)$$

$$\ddot{\theta} + F \, (\alpha_B/C_B + \alpha_B/C_E + \tilde{\alpha}_W/C_E) \, \dot{\theta} + F/C_B \, F/C_E \, \alpha_B \tilde{\alpha}_W \theta = 0 \quad (18)$$

$$\tilde{\alpha}_W = 1/[1/\alpha_W + 1/(\dot{m}_e c_w)] \quad (19)$$

Another simplification applies for tanks which are less than 20K warmer than the environment. In that case, the evaporation rate W_v can be linearized as shown in Eq. (20). This linearization permits an analytical solution of the differential equation in Eq. (18) using the overall rain heat transfer coefficient $\tilde{\alpha}_W$ in Eq. (21).

$$W_v(T_B;\, T_{Wa}) = W_v(T_{B0};\, T_{Wa}) \cdot \theta/\theta_0 \quad if \;\; \theta_0 = T_{B0} - T_{Wa} \leq 20K \quad (20)$$

$$\tilde{\alpha}_W = 1/[1/\alpha_W + 1/(\dot{m}_e c_w + \alpha_U + W_v(T_{B0};\, T_{Wa})/(T_{B0} - T_{Wa}))] \quad (21)$$

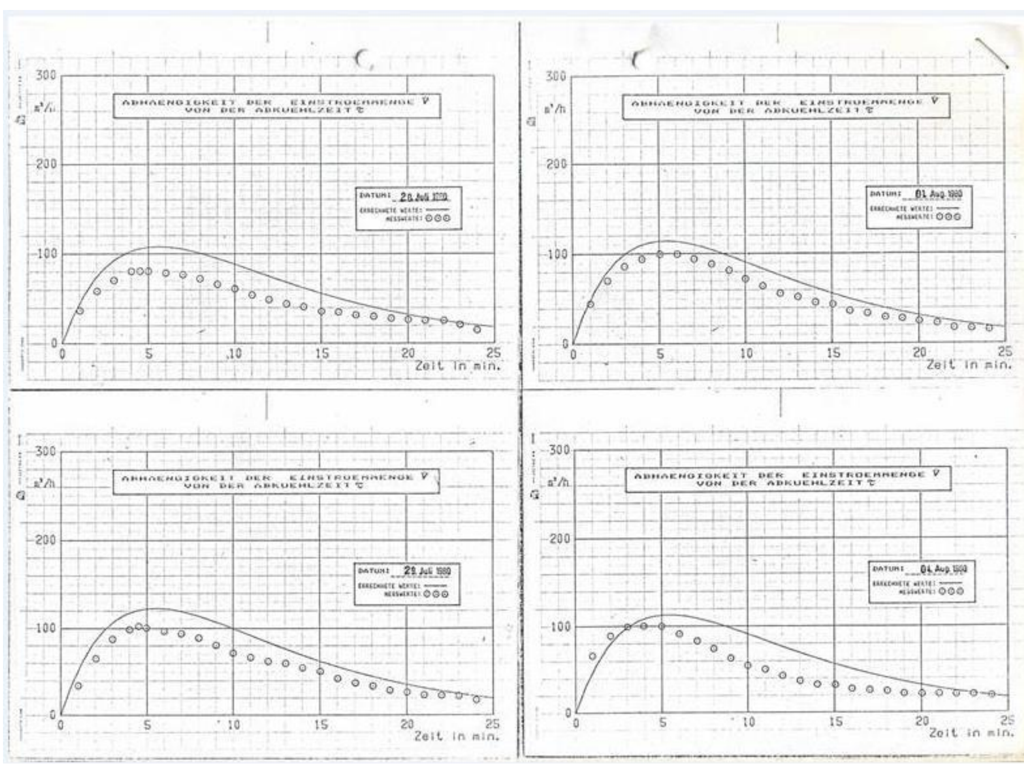
Validation and simplified approach in function of the tank geometry

The authors of the PTB document validated their method against measurements performed by (Sigel, Kuxdorf, Meiss, & Schwarz, 1983) with a cylindrical tank of a diameter and a height of 10.6 and 8.5 meter (tank volume 618 m³). For an experimental set with $\theta_0 \leq 20K$ and a rain intensity of 60 kg/(m² h), an increment of the external heat transfer coefficient α_U from zero to 5 W/(m² K), which is the value specified in API 2000 Annex E, caused an increment of about 20% of the maximum inbreathing rate. With increasing rain intensity, the convective wind cooling, which is proportional to the external heat transfer coefficient α_U , could become negligible as rain rinse cooling dominates. The measured thermal inbreathing is between 80 and 105 Nm³/h, while the one calculated by the authors is between 105 and 120 Nm³/h, see Fig. (3).

The authors concluded that, under the assumptions of a worst case scenario with a very large rain intensity of 225 kg/(m²h) and neglecting the tank wall thickness, the correlation in Eq. (22) can be developed, which matches with the formula for thermal inbreathing in API 2000 7th Ed. Main Body in Eq. (2) for C≈5. According to PTB W-22, the height over diameter ratio plays a marginal role. It should be noted that Eq. (22) has been integrally adopted within (TRGS-509, 2014).

$$\max \dot{V}_{in} = 0.12 \cdot (T_{B0} - T_{Wa}) \cdot {V_B}^{0.71} \quad (22)$$

Figure 3 Measurement of thermal inbreathing by Sigel et al. and comparison with the calculation



Salatino et al. (University of Naples)

The method of (Salatino, Volpicelli, & Volpe, 1999) considered the non-homogeneity of temperature of the metal walls inside the tank, as the shell is partially covered by the liquid, while the roof is usually in contact with the vapors. They derived a simplified method to estimate the maximum inbreathing for a tank with non-condensing products. In this method the maximum inbreathing is a function of the heat transfer coefficients h_{iG} between the gas and either the liquid or each individual tank surface A_i in contact with it, as shown in Eq. (23).

$$\max \dot{V}_{in} = \sum A_i \, h_{iG}/c \cdot R/p \, (T_{G1} - T_{G2}) \quad (23)$$

In this equation c is the molar specific heat and R the universal gas constant, while T_{G1} and T_{G2} are the asymptotical gas temperatures before and after the rainstorm which can be calculated from the individual surface temperatures T_i and the heat transfer coefficients h_{iG} , as shown in Eq. (24).

$$T_G = \sum A_i h_{iG} T_i / \sum A_i h_{iG} \quad (24)$$

For a maximum temperature drop in the gas phase of about 25°C and heat transfer coefficients h_{iG} of about 4 W/m²K, Eq. (23) can be simplified into Eq. (25), where K ≈ 2.6 m/hr.

$$\max \dot{V}_{in} = \sum K \cdot {V_{tk}}^{2/3} \cdot (1 + 2\,H/D)/\sqrt[3]{H/D} \quad (25)$$

As Salatino et al. pointed out, the determination of the heat transfer coefficients is crucial to the estimation of the temperature inside the tank. They performed a calculation for a tank with a total volume of about 63 000 m³, of which 56 000 m³ was occupied by liquid. Initial ambient and shell temperature was 309K, calculated roof temperature was 337K, and liquid temperature was 298K. Final roof, shell, and ambient temperatures were set to 293K. Using the heat transfer coefficients in Table (8), they obtained an initial and final temperature in the gas phase T_{G1} and T_{G2} respectively of 315K and 294K.

Table 8 Heat transfer coefficients between Gas and either Liquid or Tank Roof or Tank Shell

Heat transfer coefficient h_{iG} [W/m²K]	Gas & Liquid h_{LG}	Gas & Roof h_{RG}	Gas & Shell h_{SG}
Initial	3	3	3
Final	3	5	5

Table 9 Comparison among several calculation methods as done by (Salatino, Volpicelli, & Volpe, 1999)

Calculation method	Thermal inbreathing [m³/h]
ANSI/API code	4100
PTB model	12800
Pr EN 265001	6300 - 10500
Salatino dynamic	9600
Salatino complete static Eq. (23)	10000
Salatino simplified static Eq. (25)	9900

They used the calculated initial and final gas temperatures to estimate the inbreathing requirements by performing a dynamic simulation and calculating them using the complete static formula in Eq. (23) and the simplified version in Eq. (25). They compared these results against those calculated using the PTB method as well as an "ANSI/API code", which may be API 2000 5th Ed. or previous, and pr EN 265001, which is the forerunner of ISO 28300. This comparison is reported in Table (9). It should be noted that if the "API code" is API 2000 5th Ed. the application of the standard is outside its scope. Table (9) clearly shows that all three models developed by the authors approach the upper range of the pr EN 265001, and therefore, ISO 28300 and API 2000 7th Main Body.

Numerical simulations

In recent years, there have been several publications aiming to model tank breathing by means of numerical simulation using commercially available software. (Chakra, 2016) calculated the breathing requirements for the measurements done by (Sigel, Kuxdorf, Meiss, & Schwarz, 1983) with dry air using SuperChems™ and compared the results with the method in the Main Body of API 2000 7th Ed. For these simulations, the tank was placed in Montreal, Canada and Jubail City, KSA. Solar radiation and ambient temperature were adjusted for the location, but rain intensity was not specified. The results are summarized in Table (10). Compared to the simulations API 2000 Main Body is cautiously conservative.

Table 10 Comparison between Breathing requirements in SuperChems™ simulation and API 2000 Main Body

	Thermal inbreathing Chakra	Thermal outbreathing Chakra	Thermal inbreathing API 2000 7th Ed. Main Body	Thermal outbreathing API 2000 7th Ed. Main Body
	[Nm³/h]	[Nm³/h]	[Nm³/h]	[Nm³/h]
Montreal	134.5	30.3	264.1	79.1
Jubail City	148.4	32.6	352.2	101.3

Guideline for breathing of non-volatile tanks

In this guideline, some of the thermal parameters will be discussed. For non-condensing vapors, the internal heat transfer coefficient can be calculated from available scientific literature such as (Incropera, 2006) or (VDI Heat Atlas, 2010). Accordingly, values between 2 and 5 W/m²K can be expected for air at ambient condition in a wide range of tank volumes.

The heat transfer coefficient between wall and the rain shower α_W can be calculated using the equations in (VDI Heat Atlas, 2010) Chapter Md1 under the title "Heat transfer at vertical sprinkled surfaces". In that chapter, there are distinctive correlations for heat transfer coefficients for either evaporating or non-evaporating liquid films. Even for non-evaporating thin rain films, heat transfer coefficients between 2000 and 5000 W/m²K can be expected. Some authors, such as (Das, 2017), obtained values closer to the upper threshold for the heat transfer coefficient of an evaporating water film in a vertical evaporator.

The impact of wall thickness is crucial when the thermal capacitance of the walls, i.e., the product of the mass and the specific heat, exceeds the capacitance of the tank contents. Physically, the thermal capacitance of the walls is the relationship between the enthalpy loss, which is affected by the heat transfer rate, and the temperature gradient of the walls. Therefore, if two uninsulated tanks of the same volume containing the same product and exposed to the same rain shower, but with different wall thickness, are compared, the tank with the thicker walls will show the slower cooling rate and the lower inbreathing requirement.

Although the effect of wall thickness on tank inbreathing is similar to that of insulation, e. g. less inbreathing if the thickness of insulant is increased, the physical properties are different. Insulants have very poor thermal conductivity, which dampens the cooling rate from the inside to the outside.

In the calculation of inbreathing rates, the effects of heat transfer between the tank base and the soil, reflection of radiation, and tank surface absorption and emission are usually omitted. For the inbreathing scenario described in Annex E, the radiation emitted from the shell and the roof is an order of magnitude smaller than the convective heat lost through the walls to the rain film. For storage temperatures higher than that specified in Annex E, a more detailed engineering study needs to be performed. At high storage temperatures, conductive heat transfer between the tank base and the soil should be introduced into the heat balance, for instance, following the approach provided by (Hughes & Deumaga, 1974).

Most of the models and experiments described here involve dry air as a non-condensable gas. However, ambient air is usually humid and in certain locations and seasons even close to saturation. In other words, an opening vacuum breather valve is inbreathing not only dry air, but also water vapor, which may condense when the temperature drops. (Moncalvo, Davies, Weber, & Scholz, 2016) showed that a tank with saturated humid air may require a significantly larger inbreathing flow rate than what would be required for the same tank being filled with dry air.

Conclusions

The purpose of this project has been to develop a guideline for the breathing requirements of uninsulated low-pressure and atmospheric storage tanks exposed to either prolonged sunshine or intense rainfalls.

The results from the available literature have been introduced and compared for thermal outbreathing, when a cold tank is exposed to solar radiation, and inbreathing, when a hot tank is cooled down by a strong rain shower. For thermal outbreathing, there seems to be a good acceptance of the standardized calculation procedure in the Main Body of API 2000 7th Ed. and ISO 28300. The results using those standards are in good agreement with those calculated using previous editions of the API 2000 standards and now present as Annex A. For thermal inbreathing, the requirements calculated using the procedure in the Main Body lead to larger requirements for small and very small tanks than those calculated using the procedure in Annex A. From the existing literature, the method proposed by Förster et al. (PTB) delivers good reproduction of the measurements performed by Sigel et al. Given its solid theoretical model and its physically reliable thermal assumptions, the model from PTB has been elevated as European standard EN14015 prior to ISO 28300 and the actual API 2000.

Tests with storage tanks containing gases other than air, or placed outside Continental Europe are not available; however, computational simulations performed by Chakra for tanks located in Canada and KSA seem to confirm that the method in the Main Body of the API 2000 7th Ed. is acceptably conservative.

Furthermore, vacuum breathers and manholes of the smallest available sizes can breathe far more ambient air than what the method in the Main Body of the API 2000 7th Ed. would require for small and very small tanks, even for very low levels of design vacuum.

Even if the vapor space of a tank might be initially filled with non-condensable gases, humid air may enter into the tank during inbreathing. The impact of humid air inbreathing can increase the required flow rate significantly.

Considering all of these factors, it is prudent to follow the guidance provided in the Main Body of ISO-28300 and API-2000 7th Ed. for the sizing of breather valves for low pressure and atmospheric storage tanks.

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