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STUDY ABOUT THE RIGOR OF TIGHTNESS TESTS AND THE EXISTANCE OF AN EMPIRICAL CONSTANT TO VALIDATE THEORETICAL SIMULATIONS TO APPROXIMATE THE REAL MODEL

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ABSTRACT

Due to the great risk of contamination by leaking in underground fuel storage tanks (UST) of gas stations all over the world, the establishment of effective monitoring methods in this environment is extremely necessary. Among UST monitoring methods the tightness test is one of the most effective ones in identifying leaks, it can be done in two different ways, either wet part test or dry part test. But while both of the tests are permitted, they show a great difference in rigorousness, when it comes to approving or not a tank. This study envisions to deeply explore the causes of the difference of rigorousness between both tests, and discover ways in which simulations can approach the real situation. The research allowed us to identify not only the cause of such difference in rigor, but also to establish a constant that approximates the theory to the real situation.

Keywords: tank tightness, simulation, evaporation, vacuum.

1. INTRODUCTION

The notion of engineering and development being linked to sustainability and environment is an ever-growing trend. Researches with the goal of developing technologies to monitor and reduce pollution have been growing and gaining more and more space and visibility in the engineering world, as the connection between progress and environment has already become a necessity, rather than a luxury.

The vast number of leaking cases in gas stations, coming from USTs and piping systems, have caused damage to the environment, besides also harming the security, health and life quality of the population around these sites (SANDRES et al., 2002). This contamination can affect not only the soil, but also groundwater and cause explosion and fire risk.

In Brazil, there are more than 40 thousand gas stations and soil contamination is currently one of the main concerns, once around 30% of these stations present problems that can cause contamination. Multiple companies deal daily with the unforeseen coming from leaking and incorrect storage (TERRA BRASIL, 2017).

According to a research made by CETESB, gas stations are the main responsible for soil contamination in cities like São Paulo. The contamination occurs by leaking of fuel and gases due to bad installation of the USTs, which are fabricated in steel and do not possess protection against corrosion. Currently, multiple companies have invested in safer tanks, as seen that the investment in better installations it's much smaller than the costs of repairs for the problems caused by the contamination of the water and the soil (TERRA BRASIL, 2017).

Soil contamination by fuel its a big concern, seen that the fuel contains Benzene, Toluene, Ethylbenzene and Xylenes(BTEX) in its composition, that are all harmful to human health and can cause dangerous diseases. Soil contamination occurs when there isn't proper investment in quality equipment, mainly when it comes to metallic pipes and purely metallic tanks, because they are underground and it is necessary to pay attention to signs of corrosion and possible failures (TERRA BRASIL, 2017).

In a gasoline spill, one of the main concerns is the contamination of aquifers that are used as source of water for human consumption (TEIXEIRA, 2008). Due to the fact that it is very little soluble in water, spilled gasoline, containing more than 400 components, initially will be underground as a liquid of non-watery phase. In contact with underground water, gasoline will partially dissolve. The mono-aromatic hydrocarbons: benzene, toluene and xylenes, called BTEX are the components present in the gasoline that have the highest water solubility, thus, they are the first contaminants to reach the groundwater. These compounds are considered dangerous substances because they are depressor of the central nervous system. The benzene is proven carcinogenic, being able to cause leukemia (TEIXEIRA, 2008).

The gasoline commercialized in Brazil is mixed with alcohol in proportions that can go from 20% to 30%, according to the current legislation. That makes it different from the gasoline sold in other countries, where it isn't mixed with oxygenated compounds. The interaction between ethanol and BTEX can cause a rise in mobility and solubility, while also hindering the natural biodegradation of these compounds (TEIXEIRA, 2008).

The monitoring ends up being one of the main weapons to avoid contamination. The monitoring equipment for leak detection helps avoiding possible contaminations to the environment and accidents at work. Amongst USTs monitoring methods, the tightness test is one of the most effectives in the identification of possible leaks, being able to be executed in two different ways: the wet part test(i.e. part filled with fuel) and the dry part test.

This study compared the results obtained from the application of both types of test. The motivation for this paper came from the necessity of supplying scientific evidence conjugated with experimental data and theoretical concepts, some laboratory tests and real tank testing practice on site (gas stations) to better compare the rigor of the methods of leak detection regulated in Brazil by the standard ABNT 13784 in force and understand the causes of the difference found in the rigor of both methods while finding a way to approximate theoretical simulations to the practical results. This way, on site comparisons were made where, initially a wet part test was performed, and following that, the same tank was emptied and a dry part test was performed on the same tank. The research allowed us to identify a better performance in terms of rigor from the dry part test and confirmed the initial suspicion of the great influence of the evaporation rate of the fuel on the testing of partially full tanks (i.e. hybrid situation where there is a coexistence of a dry and a wet part in the tank). Based on the results obtained in both the theoretical simulations and practical tests we were able to find a constant that helps correcting any false positives and approximates both situations. We have proved that a well-made simulation(with the constant) implies on a more realistic model, and that was validated.

2. PETROBRAS’ COMMON GASOLINE

On this article, common gasoline was used as the study object, due to it having the highest evaporation rate among the liquid fuels commercialized by ANP, thus, the parcel of pressure increase allowed by a possible hole in the tank is lower, so the test executed with this fuel is essentially the most critic case.

Gasoline belongs in the group of the LNAPL (Light Non Aqueous Phase Liquids) e PMOS (Partially Miscible Organics Solubility) Mindrisz, et al., (2006). Derivative from petroleum, gasoline is composed by innumerable chemical compounds (olefins, aromatichydrocarbons etc.) among which stand out as the most water-soluble contaminants, the BTEX compounds, present in 18% of the gasoline weight. A particularity differentiates Brazilian gasoline from the ones in other nations, the considerable presence of ethanol, which currently corresponds to 27% of the volume (Portal G1, March 2016). This factor makes it possible for similar studies involving other countries’ gasolines to be able to present different results when compared to the ones made in Brazil.

Table 1: Main properties of this fuel at 1 atm

Properties	
Starting Boiling Point	>35°C
Flash Point	<-43°C
Partition Coefficient – noctanol/water	Insoluble in water. Soluble in organic solvents
Auto-Ignition Temperature	Log kow 2-7
Density	0.73-0.77
Viscosity	0,6x10-6 m2/s
Volumetric thermal dilation coefficient	1,2 x 10-3 °C-1

3. TIGHTNESS TEST: METHODOLOGY, THORETICAL FOUNDATION AND EQUATIONS.

3.1. Equations for Wet Part Tightness Test

According to the international standards and the standard ABNT 13784 in force in Brazil, the wet part test requires that, considering the tank out of operations (i.e. sales over) and the liquid at rest. The behavior of the volume of liquid inside the tank shall be observed for one uninterrupted hour and verified to see if the volumetric variation has not exceeded the permitted limit in that time. Thus, for academic purposes and to make comparisons between laboratorial and on site tests, a computational program was developed, capable of calculating the diameter of an hypothetical hole necessary for the limit situation to occur, which is 378 ml(0.1 gallon) “leaked” in one hour. Any loss of liquid volume that surpasses this limit is enough to deem the tank unfit. Thus the flow rate limit of 378 ml/h is used as input data on the implemented algorithm. Methodology illustration (figure1) and equations subsequently described.

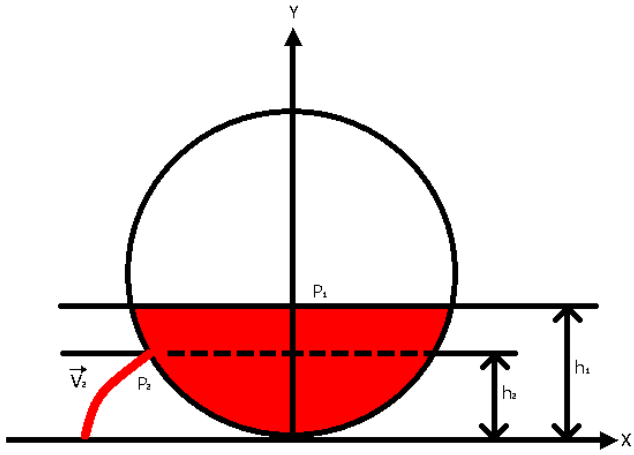

Figure 1 - Schematic representation of the methodology used to detect leaks in the wet part of a tank

Adopting a conservative approach, simplifying and considering, hypothetically, the fluid to be incompressible, non-viscous and a stationary flow(i.e. permanent flow can be considered due to the negligible variations of the water levels to obtain the admissible flow rate)., Using Bernoulli the following equations can be reached:

$$\frac{V_1^2}{2} + g * h_1 + \frac{P_1}{\rho} = \frac{V_2^2}{2} + g * h_2 + \frac{P_2}{\rho} \tag{1}$$

Where V_1 e V_2 are the velocities, P_1 and P_2 are the pressures, ρ is the specific mass of the fluid, h_1 is the height of the fuel at the start of the test and h_2 is the height of the theoretical hole.

It's observed on fig.1 that the pressures P_1 e P_2 are equal and manometric for the calculations, because they're in contact with the air and that the velocity V_1 is negligible, because the volume of liquid inside the tank is much bigger than what is leaking. Executing the algebraic manipulations necessary to isolate V_2:

$$V_2 = \sqrt{2 * g * (h_1 - h_2)} \tag{2}$$

To find out the diameter of the hole, the volumetric flow rate formula was used ($\dot{\forall}$):

$$\dot{\forall} = V * A \tag{3}$$

Where V is the flow velocity and A is the area of the hole described by:

$$A = \pi * \frac{d^2}{4} \tag{4}$$

Inserting eq.3 and eq.4 into eq.2:

$$\dot{\forall} = \sqrt{2 * g * (h_1 - h_2)} * \pi * \frac{d^2}{4} \tag{5}$$

Isolating d, the equation of the necessary hole diameter is reached. Given the heights of the hole and the fuel for a volumetric flow rate of 378 ml/h, the limit value for the permitted hole diameter is found:

$$d = \sqrt{\frac{4 * \dot{\forall}}{\sqrt{2 * g * (h_1 - h_2)} * \pi}} \tag{6}$$

3.2. Equations For Dry Part Tightness Test

For the execution of this type of test, a pump is coupled to the tank's breather, sealing all possible air inlets and cracks. With sales already over, as required per the ABNT 13784 standard, considering the tank completely sealed, the pump starts sucking air out of the tank untill a preassure drop between 90 to 100 mmHg is reached and then the pump is turned off. This procedure is repeated until after the pump's turning off the pressure drop value maintains itself stable inside the aforementioned range. After that te tank is put under observation for 30 minutes. During this time, the pressure raise cannot surpass 10mmHg (current standard) or 15mmHg (in study standard). In any case that the used limit value for the test is exceeded, the tank is deemed unfit. Thus, equally to the previous case of the wet part, for academic purposes, a computational program was developed, capable of calculating the diameter of the hypothetical hole necessary for the limit of pressure raise to be achieved. The procedures of the implemented algorithm, illustration of the methodology (fig.2) and equations are described next.

3.2.1. Dry Part Test on A Completely Empty Tank (i.e. only air)

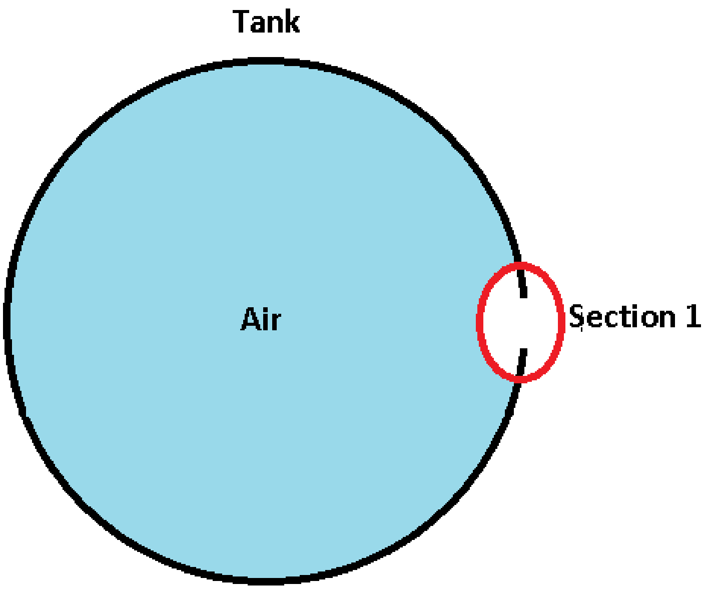

Figure 2: Schematic representation of the methodology to detect leaks on the dry part of a completely empty tank.

Initial considerations adopted:

(1) The air behaves as an ideal gas at pressures below 30 atm.
(2) The properties of air in the tank are uniform, but time dependent.
(3) Incompressible flow.

The continuity equation was used to approach the problem:

$$\frac{\partial}{\partial t}\int_{CV} \rho d\forall + \int_{CS} \rho \vec{V} d\vec{A} = 0 \tag{7}$$

Where: The first term represents the mass variation rate inside the control volume and the second term

represents the liquid rate of mass flow to the outside trough the control surface.
Once the properties in the tank are considered uniform, the specific mass (ρ) can be taken out of the integral:

$$\frac{\partial}{\partial t}\left[\rho_{CV}\int_{CV} d\forall\right] + \int_{CS}\rho\vec{V}d\vec{A} = 0, where, \int_{CV} d\forall = \forall \quad (8)$$

So,

$$\frac{\partial}{\partial t}(\rho\forall)_{VC} + \int_{SC}\rho\vec{V}d\vec{A} = 0 \quad (9)$$

The only place where mass crosses the control volume's boundary is at section 1, this way:

$$\int_{CS}\rho\vec{V}d\vec{A} = \int_{A1}\rho\vec{V}d\vec{A} \quad e \quad \frac{\partial}{\partial t}(\rho\forall)_{CV} + \int_{A1}\rho\vec{V}d\vec{A} = 0 \quad (10)$$

On the surface of section 1, the sign of $\rho\vec{V}d\vec{A}$ is negative,

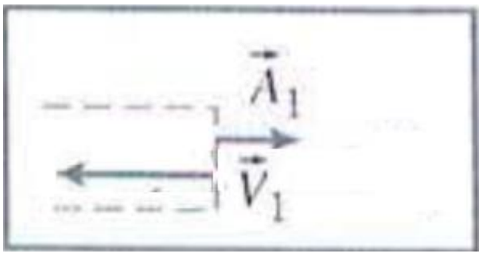

Figure 3: Surface 1

Thus,

$$\frac{\partial}{\partial t}(\rho\forall)_{CV} - \int_{A1}\rho V dA = 0 \quad (11)$$

As the flow is considered uniform on surface 1,

$$\frac{\partial}{\partial t}(\rho\forall)_{CV} - \rho_1 V_1 A_1 = 0 \quad or \quad \frac{\partial}{\partial t}(\rho\forall)_{CV} = \rho_1 V_1 A_1 \quad (12)$$

Once $\forall$ (tank's volume) isn't a function of time,

$$\forall\frac{\partial\rho}{\partial t} = \rho_1 V_1 A_1 \quad (13)$$

Isolating $\partial\rho/\partial t$, :

$$\frac{\partial\rho}{\partial t} = \frac{\rho_1 V_1 A_1}{\forall} \quad , where, A_1 = \pi * \frac{{d_1}^2}{4} \quad (14)$$

Where,

$\frac{\partial\rho}{\partial t} = Specific\ mass\ variation\ rate\ in\ time$
$\rho_1 = Specific\ mass$
$\forall = Tank's\ volume$
$V_1 = Air\ inlet\ velocity\ into\ the\ tank$
$A_1 = Section\ 1\ area$
$d_1 = Hole's\ diameter$

At this point the concept of stagnation pressure was used to find the velocity V_1 with which the air flows into the tank. Thus for an arbitrary point inside the tank, with the subscript "0" representing the stagnation conditions:

$$P_0 = P + \frac{\rho V^2}{2} \quad (15)$$

Isolating the velocity:

$$V = \sqrt{\frac{2*(P-P_0)}{\rho}} \quad , onde, \quad \rho = \frac{P[Pa]}{R_{air}[J*Kg^{-1}*K^{-1}]*T[K]} \quad (16)$$

Starting here, the subscript t_0 will be used to indicate the start of the test and $t_0 + \Delta t$ to indicate the end of it. The velocities were obtained through the developed equations. Thus, for a vacuum (induced pressure drop) of 100 mmHg and a temperature of 27 C (300 K):

$$\rho_{t_0} = \frac{87992.76}{287*300} = 1.022 \quad (17)$$

$$V_{t_0} = \sqrt{\frac{2*(87992.6-101325)}{1.022}} = 161.257\frac{m}{s} \quad (18)$$

And for an increase of 10 mmHg of pressure according to the in vigor standard:

$$\rho_{t_0+\Delta t} = \frac{89325.99}{287*300} = 1.037 \quad (19)$$

$$V_{t_0+\Delta t} = \sqrt{\frac{2*(89325.99-101325)}{1.037}} = 152.09\frac{m}{s} \quad (20)$$

For a velocity of the sound on air at 27 C, $c = \sqrt{kRT}$,
Where: K is the Volumetric Elasticity Module of the air:

$$c = \sqrt{1.4 * 287 * 300} = 347.189\frac{m}{s} \quad (21)$$

And the Mach numbers,

$$M_{t_0} = \frac{161.257}{347.189} = 0.46524 \quad (22)$$

$$M_{t_0+\Delta t} = \frac{152.09}{347.189} = 0.438 \quad (23)$$

The Mach numbers obtained (M>0.3) suggest that the flow is actually compressible, different from what was initially assumed.
Correcting the approach to a compressible flow the stagnation conditions for a compressible flow are:

$$P_0 = P * \left(1 + \frac{k-1}{2} + M^2\right)^{\frac{k}{k-1}} \quad (24)$$

And

$$\frac{T_0}{T} = 1 + \frac{k-1}{2} * M^2 \quad (25)$$

Isolating the Mach number in the stagnation pressure formula:

$$M = \frac{\sqrt{2} * \sqrt{P_0 * \left(\frac{P_0}{P}\right)^{-\frac{1}{k}} - P}}{\sqrt{k * P - P}} \quad (26)$$

The new Mach numbers will be:

$$M_{t_0} = \frac{\sqrt{2} * \sqrt{101325 * \left(\frac{101325}{87992.76}\right)^{-\frac{1}{k}} - 8799276}}{\sqrt{1.4 * 87992.76 - 87992.76}} = 0.4535$$
(27)

$$M_{t_0+\Delta t} = \frac{\sqrt{2} * \sqrt{101325 * \left(\frac{101325}{89325.99}\right)^{-\frac{1}{k}} - 89325.99}}{\sqrt{1.4 * 89325.99 - 89325.99}} = 0.4282$$
(28)

To find the flow velocities it is necessary to obtain the sound velocity and the temperature inside the tank.
Thus, using the stagnation condition of the temperature:

$$T = \frac{T_0}{1 + \frac{k-1}{2} * M^2}$$
(29)

For the start and the end of the test:

$$T_{t_0} = \frac{300}{1 + \frac{14-1}{2} * 0.4535^2} = 288.15\ K \quad (30)$$

$$c_{t_0} = \sqrt{1.4 * 287 * 288.15} = 340.26\ \frac{m}{s} \quad (31)$$

$$T_{t_0+\Delta t} = \frac{300}{1 + \frac{14-1}{2} * 0.4282^2} = 289.39\ K \quad (32)$$

$$c_{t_0+\Delta t} = \sqrt{1.4 * 287 * 289.39} = 340.99\ \frac{m}{s} \quad (33)$$

In possession of the Mach numbers e the respective sound velocities it is possible to calculate the inlet velocities of air into the tank:

$$V_{t_0} = c_{t_0} * M_{t_0} = 154.31\ \frac{m}{s} \quad (34)$$

$$V_{t_0+\Delta t} = c_{t_0+\Delta t} * M_{t_0+\Delta t} = 146.01\,\frac{m}{s} \quad (35)$$

Back to the continuity equation:

$$\frac{\partial \rho}{\partial t} = \frac{\rho_1 V_1 \pi D_1^{\,2}}{4\forall} \quad (36)$$

The time of the test can be represented by:

$$t = \frac{\Delta \rho}{\frac{\partial \rho}{\partial t}} \quad (37)$$

Where,

$$\Delta \rho = \rho_{t_0+\Delta t} - \rho_{t_0} \quad (38)$$

With,

$$\rho_{t_0} = \frac{P_{t_0}}{R * T_{t_0}} = \frac{87992.76}{287 * 28815} = 1.064 \quad (39)$$

$$\rho_{t_0+\Delta t} = \frac{P_{t_0+\Delta t}}{R * T_{t_0+\Delta t}} = \frac{89325.99}{287 * 289.39} = 1.076 \quad (40)$$

Having$\Delta \rho$, it is possible to find the diameter of the hole in an iterative way, varying it until the time of the test reaches the desired 1800 seconds (30 minutes).

3.2.2. Dry Part Test on A Semi Empty Tank

This is a very common situation, which corresponds to a hybrid case with simultaneous coexistence of both wet part and dry part. The standardizing and procedures adopted to test tightness of the dry part of partially filled tanks is similar to the one described in the previous section (i.e. completely empty tank).However, a fundamental detail has to be considered in this case. As there is fuel present in the tank, part of the pressure increase will happen simply due to the evaporation rate of the fuel during the test, with this, not necessarily indicating a leaking through a hole or something similar. By neglecting this fact, the testing companies risk mistakenly deeming a tank unfit without knowing the real cause of the vacuum drop (pressure increase).
This way, the challenge question that comes with applying this method to semi empty tanks is the following: A result that deems a tank non-tight and unfit has that result because of a real hole or simply due to fuel evaporation at low pressures generating a false positive of untightness? To help solve this problem, it is necessary to determine the parcel of the pressure raise in the tank caused only by atmospheric air inlet trough possible holes in the tank and, this way, distinguish with reliability the two possible sources of pressure increase. Throughout the research process, multiple laboratory tests were made simulating diverse situations, including on site tests contemplating real empty tanks from gas stations, which were certified tight. As a result of these experiments an empiric constant was obtained. It represents the median pressure increase in a tank on critical temperature conditions due to natural reasons (i.e. a perfect vacuum doesn't exist) and leftover gas after the emptying process of the tank. The value found was between 9 and 10 mmHg. This empirical constant was used on this paper as the way to correct possible distortions in the application of the method and due to theoretical approximations, besides other ideal laboratory considerations applied to the equations of the study. Ratifying the fact that a perfect vacuum does not exist. Thus, the empirical constant above served to

attenuate the main differences between theoretical conditions and the real situations on the gas stations.
To validate the computational simulation and the laboratory tests trough practical dry part tests of semi empty tanks, the following considerations were taken:

(1) At the start of the test, the control volume is completely occupied by gasoline vapor, as it is denser than air, thus, it tends to expel it.
(2) The control volume is the dry part of the tank not filled with fuel, and the variations on its dimensions are negligible.
(3) A maximum natural increase of 10mmHg(empiric constant) is expected on the simulation and happens, even if there are no detectable holes to allow air inlet or increase due to evaporation rate of the fuel throughout the duration of the test.

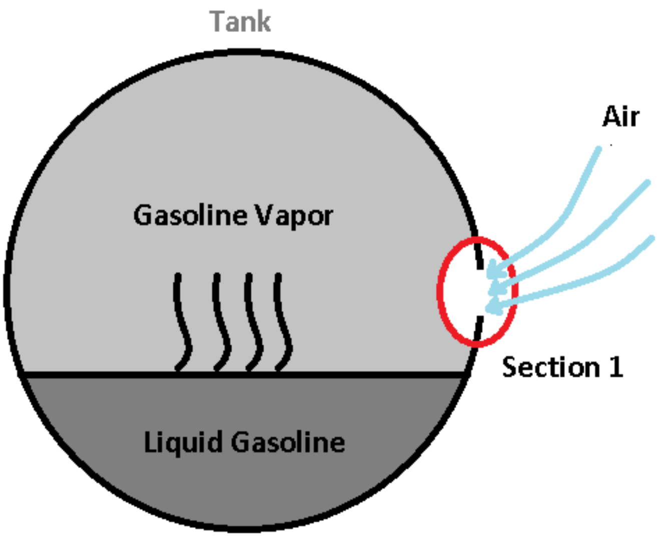

Figure 4 - Schematic representation of the semi empty tank at the start of the test

For this analysis, the Dalton law of partial pressures was used.

$$P_{Total} = P_{Air} + P_{GV} \quad (41)$$

$$P_{Air} * = \eta_{Air} * \frac{RT}{\forall}, air's\ partial\ pressure \quad (42)$$

$$P_{GV} = \eta_{GV} * \frac{RT}{\forall}, fuel\ vapour's\ partial\ pressure \quad (43)$$

As the objective is to know the parcel of the pressure increase for which the air inlet is solely responsible for, the equation to be used is:

$$P_{Air} = \eta_{Air} * \frac{RT}{V} \quad (44)$$

Which can be expanded as:

$$P_{Air} = \frac{m_{Air}}{M_{Air}} * \frac{RT}{V} \quad (45)$$

As the control volume does not contain any air at the start of the test, it is needed to find the final mass of air in the tank, which means, how much air entered the tank during the test.
To exemplify and validate the methodology and make the due approximations, it will be considered a tank with volumetric dimensions equal to the volume of the parcel of the tank not occupied by liquid gasoline (Ex.: for a tank with 30 thousand liters with 76 cm of column of liquid fuel, this theoretical volume will be of 22947 cubic meters) full of air. As the volume doesn't change, the variation of the mass of air will be given by:

$$\Delta m = \rho_{t_0+\Delta t} * \forall - \rho_{t_0} * \forall = Final\ Mass \quad (46)$$

The volume $\forall$ is the parcel of tank's volume that is not occupied by liquid gasoline.
In possession of the value corresponding to the mass of air that entered the tank the Eq. (45) can be used to calculate the increase of pressure by air inlet into the tank. This was always the procedure adopted on this research for multiple simulations e practical validation.
The temperature is the one at the end of the 30 minutes of the test. In this specific practical example the temperature was measured with the aid of well calibrated equipment of Veeder Root brand installed on the 30 de setembro gas station in Natal,RN, Brazil. The utilized tank in the simulation was of 30,000 liters with 76 cm of liquid column (common gasoline). The control volume (the volume not occupied by liquid fuel) corresponds to 22,497 liters.

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ISBN 978-88-85741-37-9; Bruzzone, Janosy, Nicoletti and Zacharewicz Eds.

Figure 5: Veeder Root Equipment

Having the value of pressure increase for which is responsible solely the inlet of air in the control volume, which is exactly what is necessary to certify tightness, a test situation of dry part test on an empty tank(only air) with a volume of 22,947 liters and a limit pressure raise corresponding to the pressure increase only by air inlet is simulated.

The approach used analyzing only the air is justified by the difficulties encountered in analyzing the pressure raise caused only by fuel evaporation (P_{VG}), due to the scarcity of data and lack of depth in studies about the properties of the gasoline in pressure conditions different from the atmospherical. Given these difficulties, the analysis with the Dalton law allows an approximation where only the air properties are necessary, these being easily accessible.

4. GENERAL RESULTS OF THE SIMULATIONS

The following tables present the data regarding the results obtained for the smallest hole diameter than can be identified by a specific test situation and the variables used in the simulation.

All the data from the simulations were obtained using the computational programs developed by the team simulating each of the previously detailed models (Wet part, empty dry part and semi-empty dry part).

Table 2: Results for the simulation of a wet part test with a completely full tank

Wet Part Test, Completely full tank (Theoretical)	
Volume of the Tank(m³)	30.607
Liquid Column Height(cm)	254
Volume of liquid(m³)	30.607
Flow Rate(l/h)	0.378
Gravity(m/ s2)	9.81
Diameter of the hole(mm)	0.1375

Table 3: Results for the theoretical ideal simulation of a dry part test with a completely empty tank

Dry part test, completely empty tank(Theoretical) (Neglecting empirical constant)	
Variation of specific mass(kg/m³)	0.017217
Volume of the tank(m³)	30.607
Initial Air Inlet Velocity(m/s)	154.3069
Initial Specific Mass(kg/ m³)	1.064019
Specific mass variation rate Calculated from the equation in the integral form (kg/m³/s)	9.53E-06
Time(s)	1800
Diameter of the hole(mm)	1.302

Table 4: Results for the theoretical simulation of a dry part test with a completely empty tank considering natural reasons

Dry part test on a completely empty tank(Theoretical) Considering the empirical constant	
Allowed pressure increase(mmHg)	15
Empirical constant(mmHg)	10
Variation of specific mass(kg/m³)	0.017217
Volume of the tank(m³)	30.607
Initial Air Inlet Velocity(m/s)	154.3069
Initial Specific Mass(kg/ m³)	1.064019
Specific mass variation rate Calculated from the equation in the integral form (kg/m³/s)	9.53E-06
Time(s)	1800
Diameter of the hole(mm)	0.753

Table 5: Results for the theoretical simulation of a wet part test with a semi empty tank

Wet Part Test, semi empty tank (Theoretical)	
Volume of the Tank(m³)	30.607
Liquid Column Height(cm)	76
Volume of liquid(m³)	7.660
Flow Rate(l/h)	0.378
Gravity(m/ s2)	9.81
Diameter of the hole(mm)	0.186

Table 6: Results for the real analysis of a dry part test with a semi empty tank

Dry part test, semi empty tank (Real)	
Mass Variation(Kg)	0.131979
Final temperature(K)	292.7
M(Molar mass) of the air(kg/Mol)	0.029
n(number of mos) of Air	0.102979
Allowed pressure increase(mmHg)	15
Empirical constant(mmHg)	10
Pressure increase by air inlet(Pa)	10.92
Diameter of the hole(mm)	0.097

With the obtained results, it is possible to verify the clear effect of the evaporation rate on the total pressure increase during the test. When comparing the smallest identifiable hole of the dry part test with a semi empty tank versus a completely empty one, the 0.753 mm from the latter is approximately 8 times bigger, thus less rigorous, than the 0.097 mm found on the former. It is also possible to state the superiority of the dry part test when compared to the wet part one, as seen that in the same situation (semi empty, 76 cm of liquid column of gasoline) the dry part test showed itself two times more rigorous than its rival, thus, much stricter when judging the integrity of the tank. The difference in results between the tests using and neglecting the empirical constant is alarming, and shows how important its use is to approximate the simulation to the real situation.

5. CONCLUSIONS

This study verifies the suspicions of the great difference in rigorousness between the two types of tightness test for USTs in gas stations allowed by the in force standard in Brazil.

The results of the study confirmed the superior rigorousness of the dry past test. It also showed the necessity to adopt an empirical constant (obtained through extensive practical tests) to approximate the theoretical simulations and the real tests. This constant being the natural raise of pressure in any tank due to natural reasons, something that also allows us to reinforce the inexistence of a “perfect vacuum”. The results were also capable of confirming the initial suspicion of the great influence of the evaporation rate of the fuel on the total pressure increase throughout the duration of the test**.**

A major hindering factor on the dry part analysis of a semi-empty tank was the lack of information about the properties of the gasoline on non-standard conditions(atmospheric pressure and average temperature). Deeper studies into the properties of this fuel would allow greater precision in determining the empiric constant and are an interesting prospect.

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