Simplified Thermal Analysis of Reinforced Concrete Fuel Storage Tanks Exposed to Fire

Fernanda S. Santos¹, Alexandre Landesmann¹
fersantos@coc.ufrj.br, alandes@coc.ufrj.br

¹ Civil Engineering Program, Federal University of Rio de Janeiro, PEC-COPPE/UFRJ, Rio de Janeiro, Brazil

Abstract. This paper proposes the application of a numerical-computational methodology via FEM for a simplified thermal analysis of a fuel storage tank, made of reinforced concrete, under fire conditions. The analyzes are carried out in two phases, including: (i) determination of the radiative fluxes emitted by a pool-fire flame occurring over an adjacent tank filled with flammable liquid fuel which strike the side of the tank under study. In this step, by defining the thermal radiation characteristics of the fire and the configuration factors, the geometrical characteristics of the pool-fire, the rate of burning and the physical dimensions of the flame are determined, and (ii) simplified heat transfer analysis for the definition of the temperature field variation as a function of the elapsed time of the fire, taking into account the mechanisms of heat exchange by conduction, convection and radiation, but neglecting the temperature dependents properties of concrete, such as spalling, cracking, and dehydration general. The procedures are developed from simulations in the software Abaqus. This paper is the initial part of a research whose aim is to provide answers to the productive sector of the petroleum industry, encompassing aspects of design and detailing of storage tanks.

Index Terms – fire, fuel storage tanks, reinforced concrete, FEM

1 INTRODUCTION

Reinforced and prestressed concrete tanks were, throughout history, widely used as reservoirs for storage of potable water, domestic and industrial effluents, as components of water treatment and sewage plants, and as silos and deposits of solid material (cereals, sugar, cement). In the global context, its application was extended to the storage of materials such as liquefied natural gas, liquefied petroleum gas, oil and oil products, industrial chemicals and various inflammable, reactive, corrosive and toxic products, which represent risks to public health and environment. The main advantages of this type of tank are: (i) its constructive and lower operating costs, (ii) the reduction of evaporative losses due to thermal insulation of concrete and, (iii) its durability and resistance to impacts, earthquakes, external attacks, atmospheric discharges and fire. The use of these tanks for storage of oil products is not recent. In the 1940s, the United States Department of Defense invested in the construction of dozens underground concrete cylindrical tanks, for safety reasons. As examples, can be cited the 25 concrete cylindrical tanks, buried and internally coated with steel, present in the port of Los Angeles, California, used to store petroleum products such as oils used in ships and aircraft fuels, and 12 similar tanks located on San Francisco Bay (Close and Jorgensen, 1991).

Only in the Brazilian petroleum industry, is estimated the existence of thousands steel tanks of different capacities. Tanks in concrete, however, have not been effectively explored. One reason for this lies in the lack of studies in the literature to respond questions related to the design, detailing, specification, construction, durability and resistance to extraordinary loads. Moreover, concrete tanks were studied as a constructive solution of lower cost and greater operational reliability (Fontes, 2013).

The fire is one of the main risks associated with storage tanks containing flammable liquids. Industrial accidents in tank farms composed of multiple tanks are recurrent and the grouped disposition promotes a phenomenon known as the “Domino Effect”, characterized by the propagation of the fire to adjacent tanks. Steel tanks are more susceptible to this effect than concrete tanks. Van Breugel (1990) ensures the assumption that, depending on the duration of the fire, concrete tanks would be capable to resist the exposure to a large fire. Thus, the probability of subsequent failure of tanks would be significantly minimized.

This work aims the implementation of a numerical-computational methodology via FEM for a simplified analysis of the structure of a reinforced concrete tank exposed to external pool-fire conditions. The analyzes are developed through computational heat transfer simulations in the software Abaqus (2011). This study is part of an ongoing investigation COPPE/UFRJ, which intends to serve as a reference in the formulation of specific technical standards, contributing to the fulfillment of structural safety criteria in fire situations (Santos, 2014).

1.1 Literature review

Guidelines for the design of concrete fuel storage tanks (Close and Jorgensen, 1988), specific recommendations on basic design criteria, inspection, maintenance, materials, construction techniques, coatings (Close and Jorgensen, 1991, Fontes, 2013), and design criteria established by technical standards (FIP, 1978, ACI 350.2 R, 2004) are available in the literature.

According to author’s best knowledge, the behavior of concrete tanks for storage of flammable products under fire conditions has been poorly addressed in publications. No relevant published research on this particular topic was identified. The main studies in the context of this work were performed on steel tanks. On the other hand, there is substantial number of available publications related to different applications con-
crete exposed to high temperatures caused by fires. Without clamping to complete, the works of Sadaoui and Khennan (2009), van der Heijden et al. (2012) and Gao et al. (2013), deserve to be specially mentioned since they proposed experimentally-based analytical (and numerical) approaches to incorporate the effects of creep, spalling and physical modeling of concrete structures exposed to fire conditions.

Based on simplified models, McGrattan et al. (2000) present a methodology to compute the radiative flux emitted by large fires in liquid and gaseous fuels. Through his method it is possible to define the acceptable separation distance between buildings and flammable deposits more accurately, as it takes into account that the smoke generated in fires of this nature, acts as a barrier that blocks part of the thermal radiation and prevents it from reaching people or structures located nearby. Fossa (2008), in turn, through a model of solid flame, studied the effects of a pool-fire, in a storage tank of liquid hydrocarbons, on adjacent tanks due to radiant heat.

Boot (2012) presents a new research being undertaken in the field of consequence modeling, focusing on modeling the fire. According to the author, the formulas and methods practiced in the safety assessment of storage areas for flammable products do not respond to specific questions of implantation and do not check the possibility of the Domino Effect, requiring more detailed calculation methods to explore the main phenomena observed in tank farms. Fontenelle (2012), via a Computational Fluid Dynamics (CFD) methodology, examined the thermal behavior of steel storage tanks, filled with ethanol, under fire conditions, also evaluating the effectiveness of the security measures adopted and prescribed by the standard.

1.2 Pool-fire simplified models

The mathematical tools capable of predicting the thermal consequences associated with pool-fires, in which the fire spreads on the horizontal surface of the liquid fuel, can be divided into three classes: Computational Models of Fluid Mechanics, semi-empirical models, and integration models.

The semi-empirical models are currently the most suitable to predict the radiative fluxes that reach objects located outside a flame. They do not focus on providing a detailed description of the fire, but focus on defining only the parameters that are relevant to the behavior of the target object. Figure 1 illustrates the three fire models established in the literature: point source model, solid flame model and modified solid flame model.

The point source model is simple and assumes that a fraction of the whole energy of the fire is released as thermal radiation. The intensity of this radiation varies proportionally with the inverse square of the distance to the source. This model, described by (1) overestimates the heat fluxes at points near the fire.

\[
q_{r,\text{inc}}(j) = \frac{\chi rQ}{4\pi x^2}
\]

(1)

Where: \(q_{r,\text{inc}}(j)\) [kW/m²] is the intensity of thermal radiation striking an element \(j\), \(\chi r\) is the fraction of the total energy released by the fire, \(x\) [m] is the distance between the source and the elements and \(Q\) [kW] is the fire's heat release rate. The fraction of the total energy released by the fire \((\chi r)\) depends on the fuel, on the size and configuration of the flame, and on the amount of smoke generated from burning. McGrattan et al. (2000) correlated \(\chi r\) with the diameter of the fire according to (2):

\[
\chi r = \chi_{r,\text{max}} \frac{D}{20}, \text{with } \chi_{r,\text{max}} = 0.35 \text{ for } D < 40 \text{ m}
\]

(2)

which shows that \(\chi r\) decreases strongly when the diameter \(D\) [m] increases. Constant values of 0.07 to 0.1 are established for \(D>40\) m. An alternative way to obtain the value of \(\chi r\) is given by (3):

\[
\chi r = 0.21 - 0.0034 D, \text{for } D < 50 \text{ m}
\]

(3)

for larger diameters, values between 0.03 and 0.06 are adopted. The heat release rate of the fire \((Q)\) can be determined by (4), proposed by Iqba and Salley (2004):

\[
Q = \frac{\pi D^2}{4} \cdot m_w \cdot \Delta H_\text{c} (1 - e^{-kD})
\]

(4)

where: \(m_w\) [kg/m².s] is the mass burning rate per unit area of the liquid surface, \(\Delta H_\text{c}\) [kJ/kg] is the combustion heat of fuel and \(k\) [m⁻¹] is an empirical constant.

The solid flame models, in turn, assume that the flame can be represented by a solid geometry, usually a cylinder. The thermal radiative flux emitted from its surface that reaches an element \(j\) is given by (5):

\[
q_{r,\text{inc}}(j) = E_{av} F_{ag} \tau
\]

(5)

where: \(E_{av}\) [kW/m] is the average emission power of the flame, \(F_{ag}\) is the configuration factor and \(\tau\) is the atmospheric transmissivity. The emission power of the flame \((E\text{[kW/m²]})\) can be calculated through different correlations based on experimental data. Iqba and Salley (2004), as in (6), assumed that the flame emits radiation uniformly along its surface:

\[
E = 58(10^{-0.00823D}), \text{for } D < 60 \text{ m}
\]

(6)
Moorhouse and Pritchard (1982) also defined $E$ as constant along the flame in (7), and Muñoz et al. (2007) suggested formulations, according to (8), for the determination of $\chi_t$ which takes into account the influence of the diameter through weighting factors:

$$E = \frac{\chi_t m_r \Delta H_r}{1 + 4 \frac{H_f}{D}}$$  \hspace{1cm} (7)$$

$$\chi_t = \begin{cases} 0.158D^{0.15} & \text{for } D \leq 5 \text{ m} \\ 0.436D^{0.58} & \text{for } D > 5 \text{ m} \end{cases}$$  \hspace{1cm} (8)$$

where: $H_f$ [m] is the height of the flame.

When the fire takes place in spaces with large diameters, however, it cannot be considered that all the visible flame emits radiative fluxes uniformly. In fact, part of its surface is blocked and, in certain parts of the flame, only the smoke emissions radiate (Mudan, 1984). The modified solid flame model takes into account this phenomenon, and uses the concept of the average emission power ($E_a$) as a relation between the power emission of the flame ($E$) and the power emission of smoke ($E_{\text{soot}}$ [kW/m²]) as suggested by Muñoz et al. (2007) in (9):

$$E_{\text{av}} = \chi_{\text{lum}} E + (1 - \chi_{\text{lum}})E_{\text{soot}}$$  \hspace{1cm} (9)$$

where: $\chi_{\text{lum}}$ is the fraction of the total energy released by the fire, determined as a percentage of the visible flame.

The solid flame height ($H_f$) can be determined by correlations obtained experimentally, that relate it with parameters of the liquid and of the flame geometry. Equation (10) expresses the correlation obtained by Heskestad (1983) in tests on pool fires over different fuels:

$$\frac{H_f}{D} = 0.235 \frac{Q^2}{D} - 1.02$$  \hspace{1cm} (10)$$

Equation (11), developed by Thomas (1962), is based on laboratory experiments in the absence of wind and considers the apparent height of turbulent diffusion flames:

$$\frac{H_f}{D} = 42 \left( \frac{m_{\infty}}{\rho_{\infty} \sqrt{gD}} \right)^{0.61}$$  \hspace{1cm} (11)$$

where: $\rho_{\infty}$ [1.2 kg/m³] is the ambient air density and $g$ [9.8 m/s²] the acceleration of gravity. There are also formulations for fires in the presence of cross-wind.

2 Methodology

The heat transfer thermal analysis was performed by the software Abaqus and can be divided into two different phases, namely:

(i) External fire: According to the concepts described in Beyler (2002), the geometric characteristics of the liquid pool are determined and, by defining the thermal radiation characteristics of the fire, are obtained the burning rate of the fuel, the physical dimensions of the flame and its temperature. The calculated heat source is then introduced into Abaqus and interacts with the tank in study, emitting radiative fluxes that hit its external face according to the configuration factors.

(ii) Heat transfer analysis: With the heat fluxes obtained in the previous step, the temperature gradients variations along the exposed tank sections are determined during the elapsed time of the fire. Numerical models developed based on FEM are employed, and the variation of the material’s thermal properties under high temperatures is considered.

2.1 External Fire

In order to simulate a typical fire scenario in a tank farm, it was considered that a tank at a distance of 30 m, with the same characteristics of the tank in study, is on fire. Wind conditions and the introduction of extinguish methods to combat the fire are disregarded during the analysis period.

The software Abaqus calculates the portion of the radiative flux emitted by the heat source that strikes the adjacent tank, being its magnitude dependent on the configuration factor, the tanks and flame geometry, and the emissivity of the flame and of the receiving surface. The heat source is implemented as an open cavity (open cavity radiation), positioned within a distance from the target (Abaqus, 2011). The determination of the flame characteristics were based on the method described in Beyler (2002), a solid flame model capable of estimating the impact of the radiation emitted by a pool-fire on a particular target. In this method, the flame height is determined by (11), a correlation developed by Thomas (1962). The intensity of thermal radiation that reaches an element located outside the limits of the flame, assumed to be cylindrical and vertical, is given by (5). According to Mudan (1984), experiments described in the literature showed that large fires over fuels with carbon and oxygen ratio greater than 0.3, for example petroleum products, produce large amounts of smoke, which is capable of blocking part of the radiative fluxes emitted in the direction of the targets outside the flame. Thus, we can calculate the average emission power of a flame ($E_{\text{av}}$) combining the thermal radiation emitted by the smoke ($E_{\text{soot}}$), calculated by Hagglund and Persson (1976) as 20 kW/m², and the radiation emitted by the visible flame ($E$) considering the ratio of these components as in (12):

$$E_{\text{av}} = (\%_{\text{visible flame}}) E + (%_{\text{soot}}) E_{\text{soot}}$$  \hspace{1cm} (12)$$

where: $\%_{\text{visible flame}}$ represents the ratio of the flame that is visible, $\%_{\text{soot}}$ the proportion of the flame that is covered by smoke.

Based on qualitative observations of video recordings of gasoline fires, Mudan (1984) estimated that the luminous parts covered approximately 20% of the flame surface. On the other
hand, an ethanol large scale fire is characterized by a significantly different burning behavior when compared with oil fire. An experiment conducted in Sweden, on 200 m² mixtures of acetone and ethanol, showed that the heat flux emitted was about two times higher than a gasoline fire in the same scale. Different from the gasoline fire, the acetone/ethanol fire was virtually free of smoke (Schill, 2012). Considering this information, it is estimated for this paper that the smoke blocks 20% of the visible parts of the flame on an ethanol fire.

The emission power of the flame is calculated by (13):

$$E = E_b \varepsilon_f$$  \hfill (13)$$

where: $\varepsilon_f$ [1.0] is the flame emissivity and $E_b$ [kW/m] is the equivalent emission power of a black body.

Equation (14), where $s$ [m⁻¹] is the extinction coefficient, is used to calculate the emissivity according to the fuels extinction coefficient (Sparrow, 1973). Mudan (1984), however, attested that these value approaches the unity for large fires.

$$\varepsilon_f = 1 - e^{sD}$$  \hfill (14)$$

The value of $E_b$ depends on the maximum temperature that the flame can reach during the fire. Experiments conducted to measure this temperature found that, for pool diameters between 0.1 and 50 m, the average flame temperature oscillates between 900 and 1100°C and is independent of the fuel (Beyler, 1986). Thus, using (15), the most critical situation was adopted in the calculation of the emission power of the flame from an ethanol pool-fire. Flame temperature in a gasoline pool-fire, in turn, was obtained experimentally by Haggglund and Persson (1976) as being equal to 967°C for pools with a diameter between 1 and 10 m, value adopted in this analysis.

$$E_b = \sigma (T_f^4 - T_{a}^4)$$  \hfill (15)$$

Where: $\sigma$ [5.67x10⁻¹¹ kW/m² K⁴] is the Stefan-Boltzmann constant, $T_f$ [K] is the flame radiation temperature and $T_a$ [293.15 K] the ambient temperature.

As shown in Fig. 2(a), in Abaqus, the numerical FEM methodology involved the discretization of two discontinued vertical cylinders (sub-models) to represent the target tank shell structure and the flame (above source tank) , with height ($H_f$) and with a temperature capable of generating the radiative fluxes correspondent with the calculated emission power of the flame ($E_{0f}$). The configuration factor ($F_{ij}$), which measures the fraction of total radiation of heat that leaves a radiative surface and reaches a receiving surface, is calculated from the concept presented in Annex G of EN 1991-1 - 2 (2002), according to (16) and Fig. 3. This calculation is performed by Abaqus, employing a heat transfer open cavity radiation FEM approach, considering the contributions of fluxes sent to the receiver in different directions depending on the orientation of the infinitesimal areas.

Finally, the thermal analysis was performed by means of a transient heat transfer model combining automatic time stepping with Newton-Raphson strategy technique and a backward difference algorithm time integration method.

$$A_i F_{ij} = \int \frac{\cos \theta_i \cos \theta_j}{\kappa \mathbf{S}_{ij}} \ dA_i dA_j$$  \hfill (16)$$
Where: $\theta$ [°] is the angle that the normal vector of the surface makes with the direction of $q_{r}$ (i), $A_{j}$ [m$^2$] is the surface area of the emitter, $dA_{j}$ [m$^2$] is the area of an infinitesimal plane of the emitter, $dA_{j}$ [m$^2$] is the area of an infinitesimal plane of the receiver and $dA_{i}$ - $j$ [m$^2$] is the distance between this two planes.

Equation (16) indicates that the configuration factor between an emitter (i) and a receiver (j) depends on the discretization of its faces of infinitesimal areas ($dA$). An infinitesimal area of the receptor interacts with each of the visible infinitesimal areas of the emitter, and the factor depends on the relative position between them, which is expressed in the formula through the angle ($\theta$) that the normal vector makes with the direction between the planes and the distance between them ($S_{ij}$).

The atmospheric transmissivity ($\tau$) was defined according to the graph shown by Beyler (2002) based on the distance between the surface of a heat source at a constant temperature of 1400 K, and the target, assuming that the ambient temperature remains constant at 293 K. The relative humidity was considered as 79.1 %, according to climatological data of the city of Rio de Janeiro (INMET, 2013), resulting in $\tau = 0.75$. In Abaqus, the heat source must have a temperature ($T_{equiv}$ [K]) capable of generating radiation fluxes ($q_{r,inc}$) as calculated. The temperature is obtained from (17):

$$-E_{av}\tau = \frac{e_{i}\sigma(T_{a}^{4} - T_{equiv}^{4})}{T_{equiv}} = \frac{\frac{e_{i}\sigma T_{a}^{4} + E_{av}\tau}{e_{i}\sigma}}$$

where, by the Stefan - Boltzmann law, the flux emitted by the surface of the fire is described corresponding to the average emission power of the flame reduced by the atmospheric transmissivity.

The convective flux emitted by the heat source is disregarded in this analysis since, according to Fontenelle (2012), isotherms above 20°C do not reach a constant tank face when it is within a distance of 30 m from the source tank. The temperature increase in the air in contact with the heated outer wall of the target tank and the temperature increase of the liquid inside it are also disregarded. Therefore, the temperatures of the gas and liquid ($T_{g}$) around the tank will be considered in this study as remaining at 20°C during the fire, according to Fig. 4, thus resulting a negative heat flow and the consequent heat loss from the wall tank for the environment and for the liquid inside. The air convection coefficient ($h_{con}$) is adopted as 2.0 (W/m$^2$K) and the coefficient of the liquid inside the tank ($h_{lit}$) is adopted as 2.0 (W/m$^2$K).

### 2.2 Heat transfer analysis

The total heat flux ($q_{tot}$), responsible for the transmission of thermal energy from a flame to a given structure is defined by the sum of the radiation flux ($q_{r,tot}$) and the convective flux ($q_{c,tot}$) according to (18).

$$q_{tot} = q_{r,tot} + q_{c,tot}$$

As in (19), the radiative heat flux is obtained by the difference between the radiant energy absorbed by the surface ($q_{r,abs}$) and the radiant energy emitted from it ($q_{r,emi}$). The absorbed radiation is characterized for being a part of the received radiant energy, whose value is dependent on the absorptivity coefficient ($\alpha$). The radiation emitted by the surface depends on its temperature ($T_{r}$ [K]), and is obtained by the Stefan-Boltzmann law.

$$q_{r,tot} = q_{r,abs} - q_{r,emi} = \alpha q_{r,inc} - \varepsilon\sigma(T_{r})^{4}$$

Therefore, considering that the Kirchhoff's law defines that the absorptivity and the emissivity of a surface ($\varepsilon$) are equal, the radiative heat flux can be defined by (20).

$$q_{r,tot} = \varepsilon q_{r,inc} - \sigma(T_{r})^{4}$$

The convective heat flux, as described in (21), is defined by Newton's law of cooling and depends on the difference between the gas temperature ($T_{g}$) and the temperature of the exposed surface. $h$ [W/m$^2$K] is assumed to be the heat transfer coefficient, which is proportional to the velocity of the gases in the near-surface region.

$$q_{c,tot} = h(T_{g} - T_{r})$$

The total heat flux is, then, defined by (22):

$$q_{tot} = \varepsilon q_{r,inc} - \sigma(T_{r})^{4} + h(T_{g} - T_{r})$$

Through the surface heat fluxes, it is possible to determine the thermal gradients variation within the sections exposed to fire. The numerical model presented in this research follows the thermal properties specified in EN 1992-1-2 (2004), except for the density, which was assumed to be constant and independent of the temperature. Table 1 shows the considerations assumed in the analysis.
The specific heat, $c_v$ [J/kgK], for concrete with 3% moisture content, was determined by (23). The function of the specific heat of concrete with siliceous aggregate can be molded from a constant value, the $c_{p,\text{peak}}$ between 100°C and 115°C with linear decrease from 115°C to 200°C, and from a linear relation to other temperatures. The thermal conductivity of the concrete, $\lambda_c$ [W/mK], was calculated from (24), where $T$ is the temperature of the concrete. The corresponding graphs are shown in Fig. 5.

$$c_v(T) = \begin{cases} 
900 & (20°C \leq T < 100°C) \\
\frac{c_{p,\text{peak}} - 1000}{85} & (100°C \leq T < 200°C) \\
900 + \frac{9}{2} & (200°C \leq T < 400°C) \\
1100 & (400°C \leq T < 1200°C) 
\end{cases}$$

(23)

$$\lambda_c(T) = 2 - 0.2451 \left( \frac{T}{100} \right) + 0.0107 \left( \frac{T}{100} \right)^2$$

(24)

The mesh implemented has elements of approximately 20 cm. In the thermal analysis, 8 nodes solid elements (DC3D8) are used for concrete. Previous numerical evaluations (Santos, 2014) showed that the meshing size of ~1.2 x 1.2m for solid flame and ~0.25 x 0.25m for tank wall, it provides accurate results for temperature variation on different regions of the tank; see Fig. 2(b). Indeed, considering the small tank sidewall thickness with regard to the other dimensions (e.g., diameter and height), only one element across the tank thickness is required. Due to the computational cost, it was decided to calculate the temperature rise occurring only in half of the tank. Moreover, the reinforced bars of concrete were not considered in the present simplified thermal analysis (Santos, 2014) – they are listed in Table 3 for further structural assessment. It is considered, thus, that the unexposed part of the tank remains at 20°C during the fire. An increment time of 5 minutes for a total of 10 hours of transient thermal analysis is used.

### 3 CASE STUDY

In the case study, a tank of reinforced concrete designed by the method described by Montoya et al. (2004) was adopted. Table 2 and Table 3 present the data used for the calculation.

The tank is supported on the ground surface and has an external floating roof, as shown in Fig. 6. For safety criteria, the specific weight of water was used in the calculations, since it is higher than the specific weight of oils.

### Data adopted for the design of the tank

- **Agressive enviromental class**: III
- **Concrete**: C30
- **Concrete cover**: 4.5 cm
- **Maximum crack width ($W_{\text{max}}$)**: 0.1 mm
- **Steel**: CA 50
- **Specific weight of the liquid ($\rho_{\text{liq}}$)**: 1000 kg/m³
- **Concrete safety factors ($\gamma_c$)**: $\gamma_c=1.5$; $\gamma_c=1.15$; $\gamma_c=1.5$
- **Concrete cover ($\gamma_c$) and weighting coefficients ($\gamma_f$)**

### Tank design

<table>
<thead>
<tr>
<th>Data adopted for the design of the tank</th>
<th>Tank design</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Aggressive enviromental class</strong></td>
<td>III</td>
</tr>
<tr>
<td><strong>Concrete</strong></td>
<td>C30</td>
</tr>
<tr>
<td><strong>Concrete cover</strong></td>
<td>4.5 cm</td>
</tr>
<tr>
<td><strong>Maximum crack width ($W_{\text{max}}$)</strong></td>
<td>0.1 mm</td>
</tr>
<tr>
<td><strong>Steel</strong></td>
<td>CA 50</td>
</tr>
<tr>
<td><strong>Specific weight of the liquid ($\rho_{\text{liq}}$)</strong></td>
<td>1000 kg/m³</td>
</tr>
<tr>
<td><strong>Concrete safety factors ($\gamma_c$)</strong></td>
<td>$\gamma_c=1.5$; $\gamma_c=1.15$; $\gamma_c=1.5$</td>
</tr>
<tr>
<td><strong>Concrete cover ($\gamma_c$) and weighting coefficients ($\gamma_f$)</strong></td>
<td></td>
</tr>
</tbody>
</table>

**Figure 6. Concrete tank with external floating roof - adapted from FIB (1978).**

As shown in Fig. 7, a thermal analysis for reinforced concrete tanks was performed, with the target tank positioned 30 m apart from the burning tank. Different fuel tanks are adopt-
ed: ethanol and gasoline for scenarios 1 and 2 respectively, allowing a comparison of the results. The average emission power of the flame in the gasoline fire ($E_{av}$) was calculated as 42.726 kW/m², resulting in a flame temperature ($T_f$) of 596.72°C. The geometry adopted for the flame was a cylinder with 33.64 m of height ($H_f$). For ethanol, was obtained a 164.928 kW/m² average emission power of the flame, which results in a flame temperature of 943.20°C. The geometry adopted for the flame was a cylinder with a 33.64 m height.

**RESULTS**

The increase of temperature was monitored at 12 points along the external and internal surfaces of tanks, as shown in Fig. 8. The equidistant points are distributed vertically, on the face exposed to the fire. The selection of these points was based on the assumption that they are the points with the highest temperatures.

**Figure 7.** Tank dimensions and fire scenarios.

The increase of temperature was monitored at 12 points along the external and internal surfaces of tanks, as shown in Fig. 8. The equidistant points are distributed vertically, on the face exposed to the fire. The selection of these points was based on the assumption that they are the points with the highest temperatures.

Figures 9 and 10 present the configuration factors on the target tank external faces. As expected, it can be observed that the faces closer to the flame, located near the upper edge of the wall and on the frontal planes, have superior configuration factors. Comparing the two figures it can be noticed that the gasoline flame, with a greater height, generates higher configuration factors than the ethanol flame. The observed increase was expected, considering that in the gasoline fire, an additional emission area is added to the configuration factor of each receptor plan.

**Figure 9.** Scenario 1 (ethanol) – Configuration factors on the target tank.

**Figure 10.** Scenario 2 (gasoline) – Configuration factors on the target tank.

Figures 11 and 12 show the temperature on the surface of the target tank for scenarios 1 and 2, respectively, after 10 hours of fire. It is observed that for scenario 1, at a given area, the external face has temperatures above 320°C, whereas for scenario 2, the temperatures in the same area are close to 140°C. The internal faces, heated by conduction, reach temperatures of 50°C for scenario 1 and of 30°C for scenario 2.

Analyzing the graphs shown in Fig. 13 and Fig. 14, and Table 4, it is possible to compare the increase of temperature on the external and internal surfaces of the target tank in both scenarios during the fire. As expected, the scenario 1 promotes the higher temperatures on the target face. The results obtained are consistent with experiments in which it was estimated that an ethanol fire emits heat fluxes two times higher than a gasoline fire of the same scale (Schill, 2012).

Higher temperatures are observed at intermediate points of the target’s face in the two scenarios simulated. The lower temperatures at the edge of the wall (monitoring point 6), result from the contribution of the upper portion of the tank in the heat dissipation due to its contact with air.
TABLE 4. MAXIMUM TEMPERATURES REGISTERED AT THE MONITORED POINTS

<table>
<thead>
<tr>
<th>Maximum temperatures (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Monitored points</td>
</tr>
<tr>
<td>Ext.1</td>
</tr>
<tr>
<td>Scenario 1</td>
</tr>
<tr>
<td>Scenario 2</td>
</tr>
<tr>
<td>Monitored points</td>
</tr>
<tr>
<td>Int.1</td>
</tr>
<tr>
<td>Scenario 1</td>
</tr>
<tr>
<td>Scenario 2</td>
</tr>
</tbody>
</table>

5 CONCLUSIONS

This paper is part of an ongoing research aimed at applying the FEM heat transfer computational model for safety analysis of reinforced concrete tanks exposed to an external fire. From the results of the analysis, the following concluding remarks can be summarized:

1) The analyzes were performed in the software Abaqus by means of a heat transfer model able to: (i) simulate the radiation flux emitted by an external heat source that strikes a target by calculating the configuration factors, and (ii) estimate the temperature evolution along the elements exposed to fire.

2) The maximum temperatures on the external sides occurred at a 25 cm distance from the top edge in scenario 1, reaching 387.38°C and at the upper edge in scenario 2, reaching 191.18°C.

3) The maximum temperatures on the inner faces occurred on the top edge for the two scenarios, reaching 165.99°C for the simulation with ethanol and 97.53°C for the simulation with gasoline. The next step of this study is the implementation of a methodology intended to conduct parametric analyzes with different distances between the source and target tanks, aiming to determine a critical distance between the units, considering: (i) the two selected fuels (ethanol and gasoline), (ii) the influence of wind in the direction of the flame and (iii) the tank material.

ACKNOWLEDGMENT

The first author of this paper acknowledges FAPERJ - Carlos Chagas Filho Foundation for Supporting Research of Rio de Janeiro State the support granted in the development of this paper through the program "Bolsa Nota 10-2013", and CAPES - Coordination for the Improvement of Higher Education Personnel the support between March/2012 and March/2013.

REFERENCES

Boot, H., Developments in Consequence Modelling of Accidental Releases of Hazardous Materials. Transactions of the VSB-