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Pool Fires: a Model for Assessing Meteorological Parameters Influence on Thermal Radiation

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When a flammable liquid is accidentally released, for example due to the rupture of a plant storage tank or a transportation incident, there is a possibility of ignition, which results in a pool fire. In these cases the estimate of the impact areas of pool fires assumes a great importance, especially when there is the real possibility of generating serious consequences (domino effects). The paper illustrates a case study, referred to an Italian biorefinery, that produces bioethanol from biomasses. In the last years there has been a growing use of bioethanol in order to replace fossil fuels with renewable fuels. A very important issue is that the heat flux from an ethanol fire can be significantly higher than that of a petroleum fire. The reason of this difference is that gasolines and hydrocarbons fires generate larger amounts of soot, which tends to block the visible parts of the flames, thereby reducing the heat flux. On the contrary an ethanol fire is almost free from soot and therefore the associated heat flux is not dissipated by smoke. In particular the paper describes a semi-empirical pool fire model, which uses a selection of sub-models correlations, aimed at determining the geometrical configuration of the flame (average height, tilt angle and elongated diameter) and thermal radiation as functions of meteorological parameters such as air humidity and wind velocity.

1. The case study: bioethanol release in a biorefinery

The examined biorefinery is located at Crescentino and produces about 40,000 t/y of ethanol. The biofuel is stored in outdoor fixed roof tanks, having an internal floating roof. The industrial settlement has three daily tanks ($V_{daily\ tank}$=193 m^3), two weekly tanks ($V_{weekly\ tank}$=1,450 m^3) and one denaturant tank ($V_{denaturant\ tank}$= 300 m^3 and its diameter is 7 m), which is located in the basin, including the daily tanks. The bioethanol, which must carry out the quality tests, is stored in the daily tanks, while the biofuel, that has passed the tests, is stored in the weekly tanks. The daily and weekly tanks are included in two different basins, which respectively have an area of 896 m^2 and 1,248 m^2. Diameter of the daily tank is 6 m and its height is equal to eight metres. The weekly tank has the following dimensions:

- diameter = 12 m;
- height = 14 m.

Ethanol storage temperature ranges from 30°C to 33°C, whereas its pressure is about 0.6 barg for the daily tanks and 1 barg for the weekly tanks. A bioethanol release from the flange (Figure 1), which connects biofuel outlet pipe with the weekly tank, is analysed. This choice depends on two factors:

1) flange failure has a relatively high frequency (about 10^{-4} events/year);
2) this flange is the nearest the ground (it is situated at 0.05 m from the tank bottom) and the weekly tank has the largest volume. It follows that the flange is exposed to the highest hydrostatic pressure and therefore, under the same emission time and hole diameter, biofuel mass flow (kg/s) is maximum as well as the released volume in the basin (the pool is not affected by the presence of the tank and it has been assumed that ethanol pool fire occurs after the release is stopped).

In order to study the ethanol release from the flange, the following hypotheses have been considered:

- D_h (hole diameter) = 0.01 m;

- t_r (release time) = 600 s.

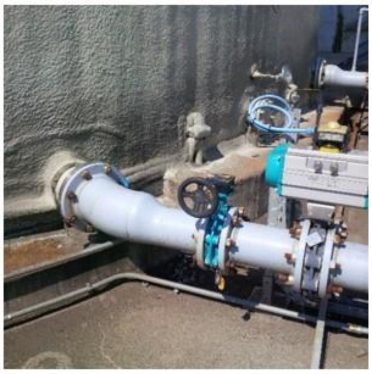

Figure 1: Flange

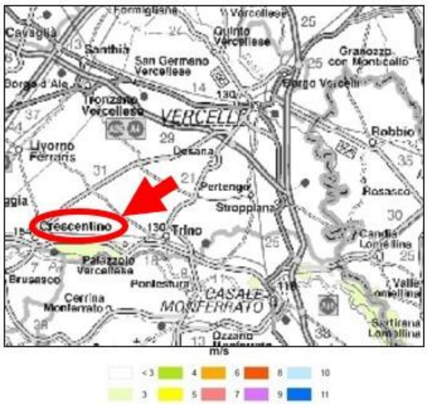

Figure 2: Crescentino (annual mean wind velocity)

The ethanol outflow velocity (v_2) is controlled by the hydrostatic pressure of the biofuel in the weekly tank and is calculated by Bernoulli equation (1 is referred to the initial condition, whereas 2 is referred to the final condition, that is the biofuel release):

$$\frac{p_2 - p_1}{\rho_{et}} + \frac{{v_2}^2 - {v_1}^2}{2} + g \cdot (z_2 - z_1) + R = 0 \tag{1}$$

Where:
- p indicates ethanol pressure (Pa);
- v is ethanol outflow velocity, expressed in m/s (v_1=0);
- z is the height from the ground (m);
- ρ_{et} is the ethanol density (800 kg/m^3);
- R indicates the local head losses (m^2/s^2).

R has been determined by 2 K Method. In particular pressure drop has been taken as equivalent to velocity head for the outlet (K_f =1) and 1/2 velocity head for the inlet (K_f =0.5):

$$R = \sum_i K_{f_i} \cdot \frac{{v_2}^2}{2} = \frac{1}{2} \cdot \frac{1}{2} {v_2}^2 + \frac{1}{2} {v_2}^2 = \frac{3}{4} {v_2}^2 \tag{2}$$

In this way R becomes function of outflow velocity, which can be calculated by equation 1. At this point the released volume (V_{et}) is determined by the expression:

$$V_{et}\left(m^3\right) = v_2 \cdot \frac{{D_h}^2}{4} \cdot \pi \cdot t_r \tag{3}$$

The thermal radiation (expressed in kW/m^2) at certain distances has been calculated as function of air humidity (AH) and wind velocity. AH has been assumed equal to 50%, 60% and 70%, whereas velocity assessment has been based on data, reported in Italy wind Atlas (Figure 2). As Crescentino is characterized by an annual mean wind velocity, that is lower than 3 m/s (RSE, 2016), 3 m/s, 4 m/s and 5 m/s have been considered in order to study the influence of wind velocity on thermal radiation. The Atlas shows velocity values, which are referred to the height of twenty-five metres, whereas semi-empirical model uses a wind velocity (v_{10}), referred to 10 metres above the ground. The conversion has been carried out by the following equation:

$$v(z) = v_{10} \cdot \left(\frac{z}{10}\right)^p \tag{4}$$

p is a dimensionless parameter, depending on the ambient (urban or rural). The plant is located at a rural zone and therefore p is equal to 0.07. In Table 1 the converted velocities are shown.

Table 1: Wind velocities referred to 10 metres above the ground

	Height (z=25 m)	Height (z=10 m)
	3	2.8
Wind velocity (m/s)	4	3.74
	5	4.7

2. Semi-empirical pool fire model

Semi-empirical modelling is a relatively simple technique for predicting the heat flux (at a distance), associated with pool fires (Morgan Hurley, 2016). The model has been focused on the prediction of flame dimensions and heat flux to external objects. A surface emitter model (Rew et al., 1997), which assumes that heat is radiated from the entire surface of solid (flame) object (tilted cone or cylinder), has been chosen, because point source models do not attempt any shape prediction and assume that the source of the heat radiation is a point. Semi-empirical model is composed by a number of calculation steps as shown in Figure 3. The pool diameter (D_p) is determined by the following equation:

$$D_p(m) = \sqrt{\frac{4V_{et}}{\delta_p \cdot \pi}} \tag{5}$$

Where δ_p is the pool thickness, which has been assumed equal to 0.015 m. The subsequent step consists in calculating the pool burning rate (v_B), expressed in kg/(m^{2}s):

$$v_B = c_8 \cdot \frac{H_c}{\Delta H_{vap} + c_p \cdot (T_b - T_a)} = 0.025 \text{ kg/(m}^2\text{s)} \tag{6}$$

in which:

- c_8 (constant)= 0.001 kg/(m^{2}s) (TNO, 2005);
- H_c (the heat of combustion of ethanol)=26,800 kJ/kg;
- ΔH_{vap} (the heat of vaporization of ethanol at its boiling point) = 920 kJ/kg;
- C_p (the ethanol heat capacity) = 2,430 J/(kg K);
- T_b (ethanol boiling temperature) =351.15 K;
- T_a (ambient temperature) = 293.15 K.

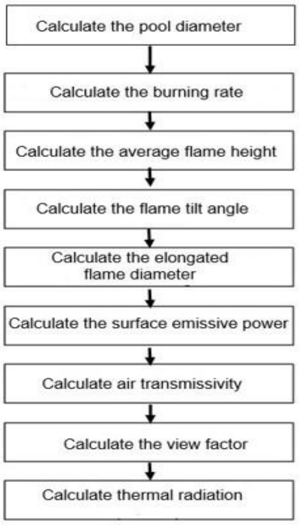

Figure 3: Calculation diagram of thermal radiation (confined pool)

At this point the characteristic wind velocity (v_c) is introduced in order to determine the flame geometry:

$$v_c(m/s) = \left(\frac{g \cdot v_B \cdot D_p}{\rho_a}\right)^{\frac{1}{3}} \tag{7}$$

Where ρ_a (1.205 kg/m^3) is the air density at 293.15 K. The subsequent step determines the scaled wind (TNO, 2005) velocity (v_{sc}):

$$v_{sc}=v_{10}/v_c \tag{8}$$

where v_{10} indicates wind velocity at height of 10 metres. Thomas correlation (TNO, 2005) is used to calculate the flame height (h_f):

$$\frac{h_f}{D_p} = 55 \cdot \left[\frac{v_B}{\rho_a \cdot (g \cdot D_p)^{1/2}}\right]^{0.67} \cdot (v_{sc})^{-0.21} \tag{9}$$

The flame tilt angle (θ) is calculated, using the Froude and Reynolds numbers, referred to $v_{10:}$

$$Fr_{10} = \frac{v_{10}^{2}}{g \cdot D_p} \tag{10}$$

$$\mathrm{Re} = \frac{v_{10} \cdot D_p}{\upsilon_a} \tag{11}$$

In which:

- υ_a (kinematic viscosity of air at 293.15 K) = $1.5 \cdot 10^{-5}$ m^2/s.

Flame tilt can be predicted using a correlation given by Welker and Sliepcevich (Welker and Sliepcevich, 1966):

$$\frac{tg\vartheta}{\cos\vartheta} = 0.666 \cdot \left(Fr_{10}\right)^{0.333} \cdot (\mathrm{Re})^{0.117} \tag{12}$$

if tanθ/cosθ= c, the tilt angle can be analytically calculated by:

$$\vartheta = \arcsin\left[\frac{\left(4c^2+1\right)^{\frac{1}{2}}-1}{2c}\right] \tag{13}$$

Because of wind influence the flame elongates and its base assumes an elliptical shape. The elongated pool diameter (D'_p) is calculated (TNO, 2005) by the following expression (cylindrical flame):

$$D_p^{'}(m) = D_p \cdot 1.5 \cdot \left(Fr_{10}\right)^{0.069} \tag{14}$$

At this point flame dimensions are completely determined. In order to calculate surface emissive power (SEP), a multi-layer model has been used, because multiple layers of surface emissive power give more accurate predictions of near-field incident radiation, especially downwind of the flame where a single-layer model can underpredict the heat flux at ground level. In this study a two-layers model has been applied. The emissive power of the lower zone is taken as the maximum surface emissive power (SEP_{max}) of clear flame, multiplied by a factor, which depends on the fraction (ζ) of the flame surface covered by soot due to incomplete combustion:

$$SEP_{\max}\left(kW/m^2\right) = F_s \cdot v_B \cdot \frac{H_c}{\left(1+\frac{4h_f}{D_p}\right)} \tag{15}$$

Where F_s is the radiation fraction, that generally ranges between 0.1 and 0.4. In this study F_s has been assumed equal to 0.25. The upper zone (layer) is assumed to be partially obscured by soot (Rew et al., 1997), which reduces the thermal radiation. An ethanol fire is almost free from soot and therefore the heat flux is not particularly dissipated by smoke. For this reason, in the semi-empirical model ζ has been chosen equal to 0.15:

$$SEP\left(kW/m^2\right) = SEP_{\max} \cdot (1-\xi) + \left(SEP_{soot} \cdot \xi\right) \tag{16}$$

In which:

- SEP_{soot} (surface emissive power of soot)= 20 kW/m^2.

Air transmissivity (τ_a) and view factor (F) must be determined for calculating the thermal radiation (q) at a certain distance. The first parameter is given by the following formula, in which p_w only depends on air humidity:

$$\tau_a = 2.02 \cdot \left(p_w \cdot x\right)^{-0.09} \tag{17}$$

p_w is the partial pressure of water vapor, whereas x indicates the distance between the flame and target. The view factor is the fraction of thermal energy, emitted by a source, which is intercepted by a target. In case of cylindrical flames, it depends on vertical (F_v) and horizontal (F_h) view factors and is given by their vectorial sum:

$$F_V = \frac{1}{\pi \cdot S} \cdot \left[tg^{-1}\left(\frac{H}{\sqrt{S^2-1}}\right)\right] - \left(\frac{H}{\pi \cdot S}\right) \cdot \left[tg^{-1}\left(\sqrt{\frac{(S-1)}{(S+1)}}\right)\right] + \left(\frac{AH}{\pi \cdot S\sqrt{A^2-1}}\right) \cdot \left[tg^{-1}\left(\sqrt{\frac{(A+1)\cdot(S-1)}{(A-1)\cdot(S+1)}}\right)\right] \tag{18}$$

$$F_h = \frac{\left(B-\frac{1}{S}\right)}{\pi\sqrt{B^2-1}} \cdot \left[tg^{-1}\left(\sqrt{\frac{(B+1)\cdot(S-1)}{(B-1)\cdot(S+1)}}\right)\right] - \left(\frac{A-\frac{1}{S}}{\pi\sqrt{A^2-1}}\right) \cdot \left[tg^{-1}\left(\sqrt{\frac{(A+1)\cdot(S-1)}{(A-1)\cdot(S+1)}}\right)\right] \tag{19}$$

$$F = \sqrt{F_h^{\,2} + F_V^{\,2}} \tag{20}$$

In equations 18 and 19, S and H respectively are equal to $2x/D_p$ and $2h_f/D_p$ (Lautkaski, 1992). A and B are calculated by the following formulae:

$$A = \frac{H^2 + S^2 + 1}{2S} \tag{21}$$

$$B = \frac{S^2 + 1}{2S} \tag{22}$$

The thermal radiation is given by the following equation:

$$q\left(kW / m^2\right) = SEP \cdot \tau_a \cdot F \tag{23}$$

3. Results and discussion

In order to assess the influence of meteorological parameters (wind velocity and air humidity) on thermal radiation, various scenarios have been considered and for every case the profiles of heat flux as a function of the mentioned parameters have been reported (Figures 4, 5 and 6) and compared.

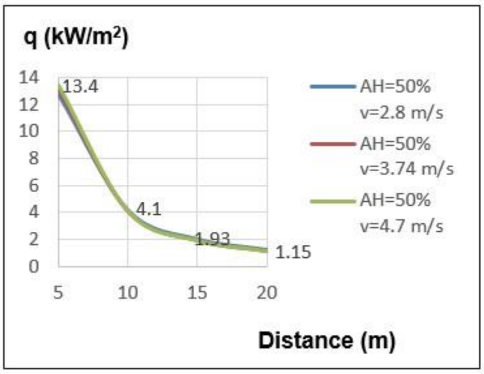

Figure 4: Thermal radiation (AH=50%)

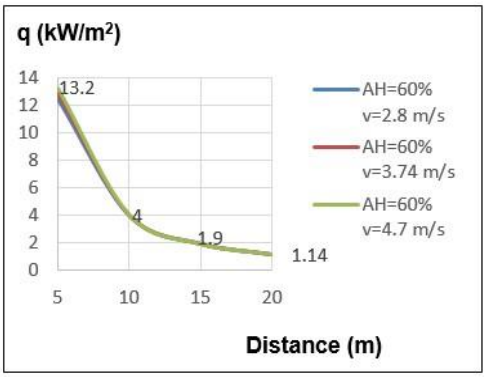

Figure 5: Thermal radiation (AH=60%)

Air humidity only influences the heat flux by the atmospheric transmissivity, whereas wind velocity influences both flame dimensions (elongated diameter, tilt angle and height) and thermal radiation (in equation 15 h_f only depends on v_{10}). In this study the same conditions of ethanol release from flange have been considered and therefore biofuel outflow velocity (v_2), released volume (V_{et}) and pool diameter (D_p) do not vary in the nine examined cases. These parameters are respectively equal to 13.7 m/s, 0.645 m^3 and 7.4 m.

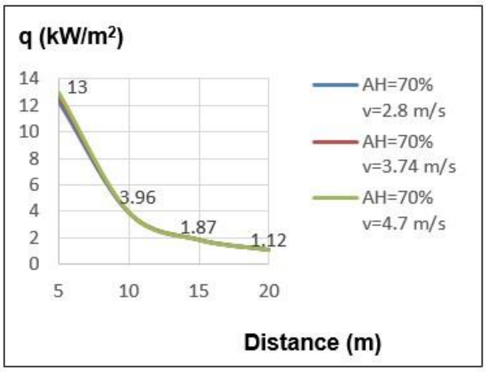

Figure 6: Thermal radiation (AH=70%)

Under the same humidity and distance, wind velocity passage from 2.8 m/s to 4.7 m/s determines a SEP increase of about 7.9% (Eq. 16), an (average) view factor decrease of about 6.6% and the maximum thermal radiation is equal to 13.4 kW/m^2 (AH=50%, d=5 m and v=4.7 m/s). On the contrary, under the same velocities and distances (SEP and view factor do not depend on humidity), the air humidity passage from 50% to 70% determines an exiguous average transmissivity decrease of about 3.4% and an average percentage of thermal radiation decrease of about 2.8%. The profiles of heat flux have been limited to twenty metres, because, at this distance, thermal radiation is lower than value, which could injure human health (Table 2). It follows that impact area of pool fire is lower than 20 m. In the examined ranges of variability of meteorological parameters, the curves of heat flux are characterized by exiguous shiftings and tend to be superimposed. In spite of the low value of fraction (ζ) of the flame surface covered by soot, thermal radiation rapidly decreases in the range of distances included between 5 m and 10 m, whereas its decrease becomes more gradual over 10 metres. In Table 3 the wind velocity influence on flame geometry is shown. The velocity passage from to 2.8 m/s to 4.7 m/s determines a flame height decrease of 10%, a tilt angle increase of about 14.6% and a pool diameter increase of about 7.4%.

Table 2: Fire effects

Effects	Thermal radiation (kW/m^2)
no effect for clothed men	1.4
first degree burns	1.8
second degree burns	2
damages to steel tanks	12.6
50% lethality	19
100% lethality	40

Table 3: Geometrical characteristics of flame

v_{10} (m/s)	h_f (m)	Θ (°)	D_p'(m)
2.8	6	48	9.5
3.74	5.7	52	9.9
4.7	5.4	55	10.2

At 5 metres from the radiant source, the heat flux is able to cause damages to the adjacent tanks and this has particularly influenced the choice of firefighting systems, applied to the ethanol storage area in order to prevent domino effects.

4. Conclusions

The risk analysis of a pool fire must be aimed at characterizing the geometrical configuration of the flame and thermal radiation intensity from the fire centre, considering the influence of meteorological parameters such as wind velocity and air humidity. The illustrated model is able to meet these requirements. It has been developed after a review of literature and published full-scale measurements in order to assess the current status of modelling of thermal radiation. The phenomenon complexity requires a model, which uses a selection of sub-models correlations. Nowadays there is a large quantity of published large-scale data, that can be used to validate pool fire models, because they cover the majority of fuels, pool sizes and ambient conditions. However the main deficiencies of the semi-empirical models depend on experimental errors and inaccuracy, which can be reduced by improved modelling of soot obscuration. An alternative to semi-empirical modelling of pool fires is the use of CFD models (Lauri, 2015). However, these require sub-models for combustion, soot production and radiative heat transfer. These sub-models contain some level of empiricism and therefore CFD models also require validation. Once validated, CFD models have the potentiality to address effects such as enclosure of the fire and obstructions within the flame. These benefits must be balanced against the relative ease of use of semi-empirical models, which, within their range of validation, provide an efficient methodology, aimed at calculating the heat flux for hazard assessment purposes.

Acknowledgments

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