Fire Safety Journal 78 (2015) 229–237

Contents lists available at ScienceDirect

Fire Safety Journal

journal homepage: www.elsevier.com/locate/firesaf

Thermal exposure from large scale ethanol fuel pool fires

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ARTICLE INFO

Article history:
Received 4 March 2015
Received in revised form
3 September 2015
Accepted 19 September 2015
Available online 30 October 2015

Keywords:
Ethanol
E85
Alcohol
Large scale pool fire
Radiation
Test
Plate thermometer
Tank fire

ABSTRACT

The growing use of ethanol as fuel for combustion engines has dramatically increased the need for large scale storage of ethanol in tanks. There are new risks related to fires in storage tanks having larger volumes. Very little experimental data exist to support risk assessments regarding emitted radiation and burning rate for large pool fires. Experience from small scale tests show that the exposure to nearby surroundings is less for alcohols than for hydrocarbon fuels like gasoline and these results are often extrapolated to fires of large sizes. This paper describes the results of two pool fires conducted within the frame of the ETANKFIRE project, one with 97% ethanol and 3% gasoline and the other with 85% ethanol and 15% gasoline, both with a surface area of 254 m^2. The results show, contrary to experience from small scale pool fires, that the exposure to nearby surroundings is much larger for ethanol-rich fuels compared to the calculated radiative heat flux from a pure gasoline fire of same fuel area.

1. Introduction

The primary hazard of large pool fires is their thermal impact to the nearby environment, which constitutes a risk of infrastructure damage, injuries, spread of the fire, and difficulties in firefighting. Common large pool fires of liquid fuels include spill fires and fires in storage tanks. The research on large pool fires is often concerned with hydrocarbon fuels detailed in e.g. handbooks [1] and reviews [2–5]. Generally speaking, an increase in the size of a fire burning with an optically thick flame decreases the radiative fraction, χ_r, which is the fraction of the radiative heat release rate (HRR) to the total HRR, $\dot{Q}_{tot}$[1]. This decrease in χ_r is due to incomplete combustion as oxygen diffusion becomes insufficient for combustion throughout the flame. In addition, when combustion becomes less complete, more soot is formed which absorbs more radiation from the luminous flames, further lowering the thermal impact on the surroundings [3,6].

The extensive work on hydrocarbon based fuels is based on a number of large scale tests including fuels like gasoline [7], crude oil [2,8], kerosene [[2] and references therein], jet propellant (JP-4) [9,10] and liquified natural gas (LNG) [11]. Large scale fire tests on water miscible fuels, including ethanol are, however, very rare. These are a very important class of liquids that are very difficult to extinguish when burning [12]. The rapidly growing demand for ethanol fuels[2] (e.g. E85 or E5 – as standard inclusion of ethanol in normal gasoline) has increased the number of ethanol storage tanks around the world. These tanks have burned on several occasions but none of the fires have been successfully suppressed; all known fires of this type have resulted in the total loss of both the ethanol content and the storage tank. One of the most famous examples is the 800 m^2 Port Kembla Fire in Australia where about 4000 m^3 of ethanol was lost and 50,000 l of alcohol resistant foam was used [13]. Other more recent examples are the tank fire in Ourinhos, Brazil, that burned for more than 30 h in 2013 or the fire at Dois Córregos later the same year [14]. The former was presumably started by a lightning strike. This fire ended in a complete burn-out even though 10,000 m^3 of water was used [15].

The burning behaviour of ethanol pools having fuel areas of less than 0.25 m^2 has been thoroughly investigated during the past several decades [16–18]. Since large scale test results are, to this point, non-existing, correlations from small scale tests are often used to extrapolate to larger ethanol fires. This approach results in significantly underestimating the thermal impact from ethanol-

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[1] The theoretical HRR if all evaporated gases burn completely, $\dot{Q}_{tot}=\Delta H_c \bullet \frac{dm}{dt}$, where ΔH_c is the theoretical heat of combustion.

[2] The nomenclature is the percent by volume of ethanol in the fuel, with the balance of the fuel made up of gasoline.

http://dx.doi.org/10.1016/j.firesaf.2015.09.003
0379-7112/

rich fuels compared to hydrocarbon fuels like gasoline. Correlations have been applied for large fires of 100 m^2 [19] and even full scale tank fires [20]. The underestimation of thermal impact is due to an underestimation of the radiant fraction for alcohol rich fuels, which we show here is much larger that for typical hydrocarbon fuels.

The 197 m^2 pool fire tests conducted in 1990 [12] showed that a burning mixture of 70% acetone and 30% ethanol radiated far more than a gasoline fire of the same fuel surface, despite having a lower heat of combustion. This result indicates that increased combustion efficiency and decreased soot formation can have large effect on the thermal impact to the surrounding area. Therefore, the same behaviour is to be expected for ethanol pool fires. Unfortunately, a complicated pool geometry and gusty wind conditions made the 1990 data too uncertain for quantitative analysis [12]. Thus, the degree (if any) to which the radiation from large ethanol pool fires increases in relation to equivalent gasoline fires has remained unknown.

The following work is intended to address this knowledge gap [21]. New experimental results on E97 and E85 pool fires (97% and 85% ethanol by volume mixed with gasoline) are presented. For 2 m^2 pool fires the thermal impact of E97 and E85 is much less than that of gasoline. However, by increasing the fuel area to 254 m^2, the ethanol fuels produced radiant heat fluxes between two and three times higher than the calculated radiation from an equivalent gasoline fire, despite having a much lower total HRR.

2. Experimental procedure

To estimate the combustion efficiency and the radiant fraction we define the theoretical heat of combustion, ΔH_c, as the weighted averages of the tabulated data for ΔH_c for ethanol (26.8 MJ/kg) and gasoline (43.7 MJ/kg), from the SFPE handbook. This yields 27.3, 29.3 and 35.3 MJ/kg for E97, E85 and E50, respectively.

2.1. Thermal exposure measured by plate thermometers

It has been shown that data from plate thermometers (PT), originally designed to control furnaces in fire resistance testing [22], can be used to accurately calculate the incident radiant heat flux towards a surface [23–26]. The simple design and low cost of PT therefore enables a large numbers of instruments to be used to cover both long distance, high resolution and many dimensions. In this test, a modified version of the standard plate thermometer was used in order to improve accuracy and response time. The exposed surface was made of Inconel 600 alloy, was 0.4 mm thick, and had dimensions of 100 mm by 100 mm. It was insulated with 30 mm of ceramic wool insulation and supported on the backside with a stainless steel plate. The Inconel 600 and stainless steel plates were connected via four thin strips of Inconel in the corners [25]. A 1 mm shielded type K thermocouple was attached to the inside of the exposed surface.

At steady state, the PT records a temperature close to the adiabatic surface temperature, defined as the temperature of a perfectly insulated surface subject to a specific fire exposure and surface emissivity. In addition, it can be used to calculate the incident radiant heat flux, $\dot{q}''_{inc}$, normal to the surface orientation, through the relation [23]

$$\dot{q}''_{inc}=\sigma T_{PT}^4+\frac{(h+K_{PT})(T_{PT}-T_\infty)+C_{PT}\frac{dT_{PT}}{dt}}{\varepsilon_{PT}} \quad (1)$$

where h is the convective heat transfer coefficient between exposed surface and air, T_{PT} and T_∞ are temperatures of PT and ambient air, respectively, ε_{PT} is surface emissivity of the exposed surface and σ is Stefan–Boltzmann's constant. K_{PT} and C_{PT} are correction parameters for heat loss and storage, respectively, in the PT. These were determined through calibration measurements during the same and previous tests using water cooled heat flux metres (HFM), which in turn were calibrated according to ISO 14934-2. The HFM were Medtherm total heat flux 64-series of model 18, sapphire window and with ranges of 0–20 and 0–50 kW/m^2. For this type of PT the calibration yields K_{PT}=4 W/m^2K and C_{PT}=3000 J/m^2K. Using these parameters the PT and HFM measurements correlated satisfactorily in a number of previous tests [26]. The convective heat transfer coefficient was calculated based on forced convection and the measured air temperature and wind speed, and was set to 18 W/m^2K for all PT. The comparison between PT and HFM data validates this simplification; see Results section and the appendix.

2.2. 2 m^2 tests

The small scale tests were conducted indoors. Three fuels: E50, E85 and E97 burned in a 2.0 m^2 circular pan, one test for each fuel. The composition of E97 was 97% ethanol by volume and 3% gasoline (corrected for the 5% ethanol already present in commercial European gasoline). E50 and E85 was mixed according to the same principle. The pan was placed on a scale under an oxygen consumption calorimetry hood system. Other instrumentation included video recording of the flame height and room gas temperature measurements. A 100 mm tall thermocouple (TC) tree with 10 mm interspatial shielded thermocouples was placed in the centre of the pool to measure the fuel surface regression rate. PT were positioned around the pool vertically (V), facing the fire, and horizontally (H), facing the sky. The PT were placed in two opposing directions at 0 (V), 1 (V+H) and 3 m (V+H) from the pool rim. HFM were positioned at 1 m (V), see Fig. 1 for details.

2.3. 254 m^2 tests

The large scale pool fires were conducted at the Dala Mitt Fire Brigade training centre in Sweden. A flat surface concrete pool, 18 m in diameter with a 150 mm high rim, was used to contain the fuel. About 20 m^3 (corresponding to a fuel depth of just under 8 cm) of E85 and E97 was used in each of the two tests but approximately 5% more fuel was used in the E97 test.

The same TC tree as used in the small scale test was placed in the pool centre. In the downwind direction PT were positioned at 0 m (V), 5 m (V+H), 10 m (V+H), 20 m (V+H), 30 m (V) and 40 m (V) from the pool rim. Two PT (V+H) were also positioned elevated 5.5 m over the pool rim plane. In the up- and sidewind directions PT were positioned at 0 (V), 5 (V+H), 10 (V), 20 (V) and 30 m (V). HFM was also used at 5, 10 and 30 m distance in order to confirm correlation with the PT measurements. Fig. 1 and Table 1 summarise the measurements.

The visible flame was documented by video camera. Gas temperatures were recorded using small TC that were placed behind the PT at 5 and 10 m to protect them from direct exposure to radiation from the fire. TC were also placed in a weather station situated 50 m from pool centre, which also recorded wind speed and direction. There was no precipitation, the prevailing wind direction was from the south for both tests and the wind speed averaged 1 m/s for the E85 test and 2.5 m for the E97 test.

2.4. Uncertainty analysis

All temperatures are measured with shielded type K-thermocouples of 1 mm diameter, bought from Pentronic, Sweden. They are all calibrated to an uncertainty of $\pm$ 1.5 °C. The burning rate estimations do not suffer from this uncertainty of temperatures