Flow Visualization within the Liquid Phase of Pool Fire

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The effects of altering the lower thermal boundary condition of a methanol pool from −5 °C to 50°C was investigated within a 90 mm diameter and 12 mm deep quartz burner under steady state burning condition in a quiescent air environment. The temperature and velocity within the liquid were measured by a single thermocouple traversed through the pool and PIV, respectively, in order to better understand the transport of mass and energy in the liquid. Temperature measurements revealed a distinct two-layer vertical thermal structure with the upper layer of the pool being almost uniform and near the boiling temperature of the fuel, while in the lower layer experienced an increasing temperature gradient as the bottom boundary temperature was lowered. The thickness of the thermally uniform layer increased as the bottom temperature was increased. The measured fluid velocity showed a complementary two-layer structure with the upper layer being dominated by a pair of counter-rotating vortices that kept this portion of the liquid well mixed and transferred heat from the hot pool wall to the pool center, while the flow in the lower layer was uniformly low in value and vertical.

INTRODUCTION

A pool fire can be defined as the combustion of fuel vapor emitted from a horizontal layer of liquid fuel. The pool fire is a coupling between the gas phase (flame) and the liquid phase (fuel layer) as the rate of combustion is controlled by the rate of fuel evaporation and vice versa. The vaporized fuel and the oxidizer react in a diffusion flame, and heat is transferred from the flame and hot combustion products to the liquid pool through conduction, convection and radiation [1]. This energy is required for the fuel heating and evaporation to sustain the combustion.

Most of the pool fire studies so far have focused on the gas phase of the pool fire leaving the liquid phase poorly presented in the analysis. For example, numerical models of pool fires mostly consider only the gas phase explicitly, using the fuel’s liquid-gas interface as a boundary condition with some assumptions regarding the local mass flux of fuel vapor across that boundary [2]. As the most advanced model so far, the liquid phase was assumed stationary neglecting any motion within the liquid fuel [3]. These assumptions have an unknown influence on the model’s outcome, but could be profound as the mass flux through the liquid surface is often used as a surrogate for the burning rate, contingent upon the usual assumption that all the fuel vapors emitted participate in the combustion.

In a previous study [4] it was shown that the changing of boundary conditions on the liquid side had a noticeable effect on burning characteristics (e.g. burning rate and flame height) of the pool fire indicating that it was not only the gas phase which controlled pool fire dynamics. Temperature distributions also give evidence that there might be large scale fluid motion and mixing within the liquid fuel and is too profound to be neglected [4].

Others have raised the importance of the non-uniform evaporation rates, and as such the liquid surface may experience convection induced by both buoyancy and thermocapillary forces [5]. This phenomenon is well known in the flame-spread stage of pool fires. The region beneath the flame is hotter than other regions and motion is induced by thermocapillary stresses on the fuel surface from hot to cold regions to accelerate the flame spread over the liquid fuel layer [6, 7]. To simulate steady pool fire burning, the system was modeled as a free surface liquid layer over a cold solid base that was heated from above by a non-uniform heat source [8]. This work showed that for liquids with a Prandtl number of unity or larger, under conditions which were similar to the steady burning of liquid pools, thermocapillary and buoyant flows were induced within the liquid layer.

To further investigate the liquid phase phenomena, the motion within the liquid fuel is examined using high speed particle image velocimetry (PIV). These results are unique as there is little experimental data available for liquid phase of steady pool burning. The study is aimed to reveal the presence of recirculation zones within the fuel.

EXPERIMENTAL SETUP AND PROCEDURE

Experiments were conducted under steady-state, steady-flow conditions associated with maintaining a constant fuel level in the pool while burning. The pool was kept full of liquid fuel to the top edge of the confining walls to eliminate any effects of ullage [9]. The temperature at the bottom of the liquid layer was held at a prescribed constant temperature. The tests were conducted in a quiescent environment with no transverse airflow and at atmospheric pressure. The fuel used in this study was methanol (CH₃OH) which at atmospheric pressure has a flash point of 11°C and boiling point of 64.7°C [10].
A schematic of the burner used in this study is shown in Fig. 1a. The burner was circular with an inner diameter of 90 mm and a depth of 12 mm. The burner wall was made of 2.5 mm thick quartz tube (95 mm outer diameter), which was exposed to the room conditions on the outside. The bottom of the burner was made of 3 mm thick porous bronze plate with an average pore size of 10 μm. The porous plate provided a uniform inlet fuel flow into the bottom of the pool while it was heated/cooled from underneath by a heat exchanger. The heat exchanger was a flat spiral coil made of 6 mm diameter copper tube that was in contact with porous plate. The fluid circulated through this coil was 50% ethylene glycol 50% water solution and its temperature was set by a water bath (Model 12111-21, Cole Parmer Canada Inc.) controllable between -20ºC and 50ºC. As shown in Fig. 1a, the fuel was supplied to a cavity occupied by the heat exchanger beneath the porous plate.

![Schematic of the burner and experimental setup](image_url)

Fig. 1 (a) a section view of the burner and (b) a schematic of the experimental setup

Fuel was supplied to the burner at a rate to keep the fuel level within the burner constant. Assuming that the evaporated fuel from the pool was burned completely in the flame, the burning rate was estimated by measuring the fuel flow rate to the burner. The schematic diagram in Fig. 1b shows the fuel delivery system consisting of an ultrasonic level sensor (Model 098-10001, ML-101, Cosense Inc.), a custom-designed software (LabWindows/CVI, National Instruments Corporation) PID controller, and a peristaltic pump (MasterFlex L/S digital driver with Easy Load II head, Cole Parmer Canada Inc.).

The level sensor monitored the fuel height with an accuracy of 0.01 mm in a small (6 mm diameter) non-combusting, inter-connected shunt-pool located immediately adjacent to the main pool as shown in Fig. 1b. Then, the peristaltic pump was used to set the flow rate of the fuel with an accuracy of 0.05 mg/s to compensate the evaporated fuel of the pool and keep the fuel level steady. The fuel flow rate transferred to the pool was measured as the set flow rate by the pump and recorded for 20 min after giving enough time (~10 min) to ensure that a steady burning condition was achieved.

Temperature within the liquid fuel was measured with a Type K thermocouple probe (TSS series, Omega Engineering Inc.) with an exposed 0.25 mm junction. The thermocouple was traversed by a 3-axis motorized stage to measure temperature in a grid pattern of 18 points in the vertical and 21 points in the horizontal directions across the region of interest shown in Fig. 1a as the field-of-view (FOV). The region of interest for temperature measurements is 30 mm wide adjacent to the pool wall covering the pool depth (12 mm deep).

The temperature measurements were collected using a computer controlled data acquisition system (Model NI 9219, National Instruments Corporation) at 10 Hz. One hundred temperature samples were recorded at each point after pausing 10 s at new point to assure that the thermocouple had adjusted to the temperature of the new location point (thermocouple response is less than 1 s). To avoid heat conduction along the thermocouple probe and the associated uncertainty in the temperature readings, the probe was shaped in a way that the last 35 mm of its length was positioned at the same depth within the liquid pool. The temperature gradient in the x-direction was reported to be insignificant [15] assuring that heat conducted along the probe was negligible.

Although the bath temperature was settable from -20ºC to 50ºC, the fuel temperature entering at the bottom of the pool was different as a result of heat transfer either to or from the room. Using the same thermocouple used for temperature mapping, the temperatures at the bottom of the pool were measured at different radial positions. The representative fluid bottom temperature at the bottom of the pool, $T_{bot}$, was found by an area-averaged temperature to range from -5ºC to 50ºC.

The velocity field within the liquid fuel was determined by PIV. The liquid fuel in the pool was seeded with 10 μm hollow glass sphere particles with the specific gravity of 1.1, the settling velocity of the particles was in the order of 0.1
A light sheet was produced by a 2D mirror scanning system composed of two mirrors of which one was fluctuating at 500 Hz and distributing laser beam (LRS-0532-TF, 1.6 W, 532 nm, Laserglow Technologies) to illuminate the region-of-interest. A lens located at its focal length from the fluctuating mirror was used to make a parallel and uniform light sheet that passed through the pool’s center. A time series of particle images were collected at 100 fps by a 1024×1024 pixel CMOS high speed camera (MV-D1024E-160, Photon Focus) equipped with an SLR camera lens (65 mm, f/11, Nikon).

The field-of-view for velocity measurements, shown in Fig. 1a, was the same used for temperature mapping, which was 30 mm wide × 12 mm deep adjacent to the pool wall where strongest fluid motion was expected due to the large temperature difference between the wall and the liquid fuel [4]. Due to the strong image distortion for the circular pool, the velocity measurements toward the pool’s center were associated with a large uncertainty. Therefore, the width of the FOV was restricted to the first 1/3 of the pool diameter as a tradeoff over having a large FOV from the wall to the center of the pool to reduce the results uncertainty.

To consider the effects of the pool curvature on the images, the velocity measurements were corrected for image distortion. A custom-designed 2D calibration target was placed within the pool and aligned with the laser sheet location. The target was a 2D array of circular dots of known diameter (0.5 mm) located 1.5 mm apart from each other. The distorted target image is shown in Fig. 2a. The target images were processed using commercial software (DaVis 8.0.6, La Vision GmbH) and an algorithm detected the dots in the target images. Knowing the size and the distance of the dots, a third-order polynomial mapping function was calculated from images to de-warp the camera images, which was used to correct the images for any distortion. The de-warped image of the calibration target is presented in Fig. 2b. The root-mean-square error of the mapping function calculated from the imaged dots was 0.29 pixel which was around 5% of the averaged particle displacement between two successive images.

The collected particle images were processed and the velocity vector field was calculated using commercial software (DaVis 8.0.6, La Vision GmbH). As a wide range of velocity scales was observed (from near zero to up to 25 mm/s) especially near the wall, a multi-pass processing scheme was used starting from a large (32×32 pixel) interrogation window with 50% overlapping. The result was then improved by using 12×12 pixel window size and 75% overlapping. The mean velocity field was calculated from averaging 1000 instantaneous vector fields.

RESULTS AND DISCUSSION

The temperature within the liquid fuel was measured at different bottom boundary temperatures. Contours of temperature maps across the FOV (depicted in Fig. 1a) are shown in Fig. 3 with the interior of the pool wall being located at \( x = 0 \) mm. For the presented four different bottom temperatures, there is an identifiable thermal structure in the liquid fuel layer. In all cases, it can be seen that the liquid surface temperature is measured to be a few degree (~3-4°C) below the expected boiling point of methanol. The temperature distributions when the bottom temperature is 24°C or less show a distinct two-layer thermal structure within the liquid fuel. The lower layer has a relatively steep temperature gradient in the vertical direction, while the upper layer is relatively uniform in temperature. The thickness of the near-uniform temperature layer grows as the temperature at the bottom of the liquid fuel is increased. In the case of \( T_{bot} = 40°C \) this two layer structure is less obvious to identify as the overall temperature difference is small (~20°C) compared to other bottom temperatures.

The dimensionless temperature profiles at the pool’s central axis at \( x = 45 \) mm for the four different bottom temperatures are shown in Fig. 6. The liquid dimensionless temperature is defined as:

\[
\theta = \frac{T - T_{bot}}{(T_s - T_{bot})}
\]

where \( T \) is the measured temperature and \( T_s \) is the liquid surface temperature. The vertical coordinate is scaled against the pool depth

\[
y^* = y/L
\]

where \( L \) is the thickness of the fuel layer, which is equal to the burner depth.
It can be seen in Fig. 4 that the general thermal structure of the liquid phase relative to bottom temperature variation is similar. However, the dimensionless temperature, \( \theta \), rises faster at higher bottom temperatures. For example, for \( T_{\text{bot}} = 40^\circ\text{C} \), \( \theta \) reaches 0.6 when \( y^* \) is 0.34 whereas for \( T_{\text{bot}} = -5^\circ\text{C} \) it happens at \( y^* = 0.43 \). This is an indication of the increasing thickness of the thermally well-mixed layer at higher boundary temperatures.

![Image of temperature distribution](image)

**Fig. 3** Temperature distribution within the liquid phase of steady methanol pool fire at boundary temperature of (a) \( T_{\text{bot}} = -5^\circ\text{C} \) (b) \( T_{\text{bot}} = 7^\circ\text{C} \) (c) \( T_{\text{bot}} = 24^\circ\text{C} \) and (d) \( T_{\text{bot}} = 40^\circ\text{C} \)

![Image of velocity field and vorticity](image)

**Fig. 4** Dimensionless temperature profiles at the central axis of the methanol pool fire at different bottom boundary temperature

In order to investigate the origins of the observed thermal structure, the mean velocity field and averaged vorticity of the liquid fuel within the pool were examined. In Fig. 5, for the same bottom boundary temperatures as Fig. 3, results of the measured 2D velocity vector field overlaid with the computed vorticity, \( \omega \). For clearer presentation, only every fourth vector is displayed in the figure. The velocity field shows the existence of a large-scale mixing motion in the top layer of the liquid fuel with two strong counter-rotating vortices located adjacent to the pool wall. In contrast to the top layer of the liquid, the bottom layer is seen to have a low vertical velocity and no discernible vorticity. The thickness of the mixing motion layer and the size of the recirculation zone increase with the bottom boundary temperature, which is in agreement with the liquid thermal structures shown in Fig. 3.
Fig. 5 Vector maps of the average velocity field (every 4\textsuperscript{th} computed vector shown) with a background color map of the mean vorticity field within the combusting methanol for boundary temperatures of (a) $T_{bot} = -5^\circ C$ (b) $T_{bot} = 7^\circ C$ (c) $T_{bot} = 24^\circ C$ and (d) $T_{bot} = 40^\circ C$

The two main vortices depicted in Fig. 5 recirculate fluid in the top region of the liquid layer. The first vortex close to the pool wall rotates clockwise directing fluid away from the wall toward the center of the pool on the upper side of the vortex and toward the wall from the bulk pool on the lower side of the vortex. The driving force for this vortex is attributed to the top of the pool wall being hot because of its proximity to the base of the flame and in contact with the
products of combustion. Fuel in contact with this hot upper portion of the wall will create a buoyant flow upwards toward the liquid-vapor interface. Once this buoyant flow reaches the top of the liquid layer it must be diverted radially inward to the pool’s center in order to remain part of the pool. Along with the need for continuity in the liquid, those two motions of upwards at the wall and then inwards create this vortex. 

The second vortex has a counter-clockwise rotation driving fluid from the central region of the pool along the surface to form a stagnation point when it meets the wall vortex. The origins of this second vortex is less clear in whether it is simply fluid responding to viscous shear stress due to the existence of the buoyancy-driven vortex or whether there are separate forces on the fluid that need to be taken into account. A potential mechanism for maintaining the counter-clockwise vortex is the thermocapillary stress on the surface. The local evaporation of small scale non-luminous pool fires (e.g. methanol) is believed to be highest close to the wall [11] that can cause that motion within the liquid. However, more investigations are required to determine the driving mechanisms of these vortices. 

Both vortices contribute strongly to energy transfer in the liquid pool. The wall vortex transfers heat from the wall to the liquid. As suggested by [12], for a vessel constructed of low thermal conductivity material (e.g., quartz), the heat flux from the wall to the fuel will mostly occur immediately below the fuel surface because there is a high resistance to heat transfer down the wall material. That limited ability to transfer heat down the height of the wall results in the temperature difference between the fuel and the wall is only substantial enough near the top of the pool to establish the buoyant flow in that location. 

The thickness of the convection layer can be derived from the temperature distribution, thermal thickness \( l_t \), as the distance from the liquid surface to the depth that the temperature gradient is smaller than 20% of the overall liquid layer temperature variation throughout the pool depth. This thickness can be estimated from the velocity field, velocity thickness \( l_v \), from liquid interface to where the averaged velocity magnitude is larger than a threshold based on the fuel burning rate. The thermal and the velocity thicknesses are compared in Fig. 6 for the range of applied bottom temperatures. Linear regressions are used to describe the relationship between the mixed-layer thickness and the pool bottom temperature. In Fig. 6, the dashed line \( l_t = 0.06 T_{bot} + 5.9, R^2 = 0.87 \) is the fitted line to the velocity thickness and the solid line \( l_t = 0.09 T_{bot} + 4.1, R^2 = 0.91 \) represents the thermal thickness. It is apparent that both thermal and velocity thicknesses increase with \( T_{bot} \). It can be also seen that \( l_t \) is smaller than \( l_v \) at various bottom boundary temperatures, which can be explained by considering the properties of the fuel. The Prandtl number, \( Pr \), is ~6-7 for methanol which is much greater than unity. This indicates that heat diffuses more slowly than momentum does in the liquid phase leading to a thinner thermal thickness.

CONCLUSION

In this study the thermal and flow structures within the liquid phase of pool fire were examined as the liquid bottom temperature was changed. The liquid fuel was burning from the top while the fuel was injected to the burner from bottom at a rate matching the fuel evaporation rate to keep a constant fuel level in the burner. The temperature distributions within the liquid fuel showed that a two-layer thermal structure was developed within the pool at steady state. The top layer close to the fuel surface was almost uniform while the lower layer had a steep temperature gradient, particularly when the bottom temperature was 24°C or less. This temperature distribution within the liquid phase maintained its structure, but the thickness of the near-uniform top layer increased with increasing the boundary temperature. 

Velocity measurements within the liquid phase revealed the presence of large mixing motions within the liquid pool. Two main vortices were detected. One vortex close to the wall associated with the buoyant flow near the hot burner walls was seen as responsible for transferring heat away from the hot wall. The other vortex was counter
rotating and participated in transporting this energy toward the pool’s center. Both vortices were observed to be only present in the top region of the liquid layer. The presence of these vortical structures appears to be the reason for the near-uniform temperature seen at the top layer of liquid thermal structure. In the lower region of the pool, no significant motion was observed (other than vertical motion needed to keep the pool level fixed, which is of the order of 1 mm/min), which could explain the steep temperature gradient in the lower part of the liquid fuel thermal structure.

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REFERENCES