

IGNITION, FLAME SPREAD AND MASS BURNING CHARACTERISTICS OF LIQUID FUELS ON A WATER BED

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Abstract

An experimental technique has been developed to systematically study the ignition, flame spread and mass burning characteristics of liquid fuels spilled on a water bed. The final objective of this work is to provide a tool that will serve to assess a fuel's ease to ignite, to spread and to sustain a flame, thus helping to better define the combustion parameters that affect in-situ burning of oil spills. A systematic study of the different parameters that affect ignition, flame spread and mass burning has been conducted in an attempt to develop a bench scale procedure to evaluate the burning efficiency of liquid fuels in conditions typical of oil spill scenarios. To study ignition and flame spread, the Lateral Ignition and Flame Spread (LIFT) standard test method (ASTM E-1321) has been modified to allow the use of liquid fuels and a water bed. Characteristic parameters such as the critical heat flux for ignition, ignition delay time and flame spread velocity as a function of the external heat flux have been obtained. A series of "fire properties" corresponding to the fuel can be extrapolated from these tests and used to assess the tendency of a fuel to ignite and to sustain flame spread. Mass burning has been studied by determining the burning efficiency of different fuels (χ) under conditions where a simple one-dimensional heat conduction model describes the surface regression rate.

1.0 Introduction

The burning of oil in water is of great interest as a result of off-shore exploration, production and transportation of petroleum. This combustion phenomenon may constitute a hazard, i.e. and accidental burning slick drifting towards a platform, but it may also serve as a measure to minimize the environmental damage of an oil spill (Twardus *et al.*, 1981, Evans *et al.*, 1990, Walavalkar *et al.*, 1996).

The available information on this phenomena is quite limited. Although great effort has been devoted to the understanding of pool fires (Drysdale, 1985) and flame spread over liquid pools (Williams, 1985, Ross, 1994, Glassman *et al.*, 1981) the specific issues related to a fuel burning over a water bed have deserved little attention. Most of the work being related to fires in fuel tanks and the phenomena commonly referred as "boilover" (Ito *et al.*, 1991, Arai *et al.*, 1990). Only a few studies have dealt with the burning of a thin layer of fuel on a water bed. A good summaries of the existing knowledge are provided by Evans *et al.* (1990) and Walavalkar and Kulkarni (1996).

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In an attempt to provide an adequate methodology for ignition of oil-spills a review of the ignition methods commonly used for oil spill clean-up was provided by Jason (1989). Ignition source temperatures and successful ignition conditions have also been a subject of interest (Bech *et al.*, 1990, Thompson *et al.*, 1979). A cone calorimeter was used by Putorti *et al.* (1994) to quantify the heat flux necessary to accomplish ignition of different fuel. In this work emphasis was given to the effects of weathering and emulsification on the ignition delay time.

Assuming that the fuel layer is ignitable an important concern is thin layer boil-over. The term "boilover" has been usually applied to a fire scenario in which an open top tank containing burning crude oil, after a long period of quiescent burning, shows a sudden increase in fire intensity associated with the expulsion of burning oil from the tank (Henry *et al.*, 1983). The term boilover has also been applied to the burning of thin layers of fuel on the surface of water in order to limit the spread of oil after an accident (Koseki *et al.*, 1991a, 1991b). This scenario is commonly referred as thin layer boil-over. Although somehow different in nature, both cases result from the onset of boiling nucleation at the fuel/water interface and therefore, the time from ignition to the onset of boilover correlates well with the time needed for the thermal wave to reach the water (Garo *et al.*, 1994).

Geometrical considerations pertaining to burning rate are of great importance when considering the use of burning for oil spill cleanup. If the oil spill is not contained, the fuel layer thickness decreases till self-sustained burning is no longer possible. Typical values on the order of 0.5 mm have been identified as a minimum thickness for self-sustained burning (Arai *et al.*, 1990, Garo *et al.*, 1994, Alramadhan *et al.*, 1990). The effect of fuel thickness, pool diameter and fuel boiling point on the burning rate has also been studied by Garo *et al.* (1994, 1996) who observed that the burning rate does not depend on the initial fuel layer thickness for fuel layers thicker than 10 mm and decreases for thinner layers.

One of the first attempts to model this type of problem was made by Twardus and Brzustowski (1981), who developed a simple one-dimensional model to describe the combustion of oil slicks on water. This model describes the burning process as that of a pool fire with a heat loss term from the fuel to the water underneath. Heat losses from the fuel towards the water will increase as the fuel layer thickness decreases, therefore, a minimum thickness for self-sustained burning can be established. In a later model Brzustowski and Twardus (1982) incorporated the effects of radiative absorption in the fuel and the effect of tilting by the wind. A more realistic model that incorporates radiative feedback and the effects of turbulent buoyant motion was subsequently developed by Alramadhan *et al.* (1990), emphasis was given to the regressing surface and the gas phase and no account for heat transfer towards the water bed was made. Although these simple theories obtain expressions for the burning rate and minimum thickness for self-sustained burning, they all fail to describe the evolution of the burning rate as the fuel thickness decreases below 10 mm.

A simple way to classify all studies relevant to in situ burning of an oil slick over a water bed is by dividing the combustion process in its three different stages, ignition, flame spread and self sustained burning (or mass burning). An external source of energy will lead to ignition, which will be followed by the spread of the flame across the fuel surface. Although flame spread might be an instantaneous process for many

crude oils in their natural state, the loss of highly volatile compounds, due to weathering, and the presence of water in emulsions might lead to flame spread that needs to be assisted by external radiation. For thin fuel layers, heat losses to the water bed, might lead to a similar situation. For these particular cases a minimum size might be necessary to provide the necessary radiative heat feed back to self-sustain flame spread. Once the flame spread process is self sustained mass burning will follow. For all three distinctive processes boiling of the water bed underneath the fuel might occur and its effect on the characteristics of the combustion process needs to be considered.

This work attempts to identify an ideal configuration in which the ease by which a fuel can burn can be evaluated. The three aspects of the combustion process, ignition, flame spread and mass burning will be studied in an attempt to obtain results that depend only on the fuel. The results should be independent of specific burning characteristics and geometrical constraints, and thus, extrapolation to a large scale should be possible.

2.0 Methodology

Fuel properties vary significantly when subject to a strong heat insult, therefore, they need to be evaluated under fire conditions. Evaluation of the fuel "fire properties" that are independent of the length scale will permit the ranking of fuels in their natural state, weathered, emulsified and with additives and will allow the reduction of the number of large scale experiments necessary to determine in-situ burning protocols and procedures. By focusing on the fuel and introducing external radiation, large scale conditions can be simulated (radiation feedback from the flame to the fuel increases with length scale). It has to be noted that this is not a study of the burning characteristics but of the fuel burning efficiency. Ignition and flame spread will be studied by using the Lateral Ignition and Flame Spread Test (ASTM E-1321) and mass burning by observing the regression rate under conditions where the characteristics of the flame can be predicted adequately.

2.1 The Lateral Ignition and Flame Spread Test (ASTM E-1321)

The Lateral Ignition and Flame Spread Test (LIFT) provides characteristic "fire properties" for ignition and flame spread. Results are normally presented on "Flammability Diagrams" that provide ignition delay times and flame spread velocities as a function of the external heat flux (or characteristic length scale of the fire). The LIFT has the advantage that it allows for ignition and flame spread to be studied together which provides a more realistic scenario than other test methods, such as the cone calorimeter. The theoretical background behind this test method is extensive and more clear than for other tests therefore it provides an adequate framework for the study of complex fuels such as composite materials or crude oils.

2.1.1 Theoretical Background

The basis of the theoretical model behind this test method can be described as ignition and flame spread as a result of inert heating of a thermally thick homogeneous solid to an ignition temperature. The flame configuration applies to a flame spreading into an opposed ambient flow which corresponds well to flame spread occurring in in-situ burning. The fuel is considered thermally thick and its initial temperature and that

of the ambient is constant at T_i . An inert fuel layer is assumed with negligible pyrolysis before ignition, and an ignition temperature (T_{ig}) is employed as criterion for flame spread. The position of the flame or pyrolysis front is identified by x_f as where the surface temperature has reached T_{ig} . Heat is transferred to the solid ahead from the pyrolysis front from the external source ($\dot{q}_e''(x, t)$) and from the flame (\dot{q}_f''). Flame heat transfer ahead of the pyrolysis front is considered to occur over a region δ_f with a uniform heat flux of \dot{q}_f'' unaffected by $\dot{q}_e''(x, t)$. Although the flame heat flux is represented as a surface heat flux, more general heat transfer effects could be considered without changing the form of the final results.

The problem is analyzed by considering the heat transfer history, due to flame and external fluxes, at the flame front $x_f(t)$. This constitutes a conduction problem with an arbitrary time varying heat flux $\dot{q}_e''(x, t)$ which depends on the position of the flame front, $x=x_f(t)$, as well. The solution to this problem is readily derivable from results given by Carslaw and Jaeger (1963) or by standard analytical techniques. An elaboration of additional assumptions and the derivation of the solution are given by Quintiere (1981). There it is shown that the transient solution for the flame spread velocity can be derived from the following expression

$$T_{ig} - T_i = \frac{\dot{q}_f''}{h} \frac{2}{\sqrt{\pi}} \sqrt{\frac{a\delta_f}{V_f}} \left(1 - \frac{\sqrt{\pi}}{2} \sqrt{\frac{a\delta_f}{V_f}} \right) + \frac{\sqrt{a}}{h\sqrt{\pi}} \int_0^t \frac{\dot{q}_e''(x_f, s)}{\sqrt{t-s}} ds - \frac{a}{h} \int_0^t \dot{q}_e''(x_f, s) \exp(a(t-s)) \operatorname{erfc}(\sqrt{a(t-s)}) ds \quad (1)$$

where

$$V_f = \frac{dx_f}{dt} \quad (2)$$

The characteristic time to reach thermal equilibrium can be expressed as $1/a$ where $a = \alpha (h/k)^2$. Typically, a is of $O(10^{-3})$ to $O(10^{-2}) \text{ s}^{-1}$, and $a\delta_f/V_f$ is of similar order of magnitude.

The definition of "h" deserves special attention. The parameter "h" represents the summation of the convective heat losses and the fraction of the external heat flux not absorbed by the surface. Both heat loss terms can be expressed by $h(T_{ig}-T_i)$ as a result of a linear approximation to convective and radiative heat losses (Mikkola *et al.*, 1989).

Equation (1) is then an integro-differential equation for the flame front position x_f . The left-hand side represents the temperature rise required to sustain flame spread and the right-hand side represents the sum of the temperature increases due to the flame heat transfer, \dot{q}_f'' , and the heating imposed by the flux field, $\dot{q}_e''(x, t)$. Making the imposed flux field independent of time will result in significant simplification of equation (1) and of the experimental procedure, therefore, \dot{q}_e'' will be considered only a function of x and thus, equation (1) can be written as:

$$h(T_{ig} - T_i) - \dot{q}_e''(x_f)[1 - \exp(at)\operatorname{erfc}(\sqrt{at})] = \frac{2}{\sqrt{\pi}} \dot{q}_f'' \sqrt{\frac{a\delta_f}{V_f}} \left(1 - \frac{\sqrt{\pi}}{2} \sqrt{\frac{a\delta_f}{V_f}} \right) \quad (3)$$

2.1.2 The Ignition Mechanisms

The gas phase ignition of a solid combustible is generally the combined result of an externally imposed heat flux (radiation and convection) that causes the gasification of the solid, and the presence of conditions (in the gas or external to it) that will lead to the onset of a sustained combustion reaction. If the reaction is initiated by an ignition source (open flame, electrical spark, flying ember, etc.), ignition is normally referred to as piloted ignition. If ignition occurs without a pilot, the process is normally referred to as spontaneous, or auto, ignition. For both cases, when the solid combustible is suddenly exposed to a sufficiently strong external heat flux ignition occurs after a certain time, this time is commonly referred in the literature as ignition delay time. The ignition delay time has been defined by Fernandez-Pello (1995) as:

$$t_{ig} = t_p + t_{in} \quad (4)$$

where t_p is a characteristic time for fuel gasification and t_{in} is a characteristic time for gas phase ignition. The mechanisms leading to gas phase ignition under these circumstances are complicated and difficult to predict. In a phenomenological way the process is as follows: after fuel pyrolysis, the vapor (pyrolysate) leaves the surface, and is diffused and convected outwards, mixing with the ambient oxidizer and creating a flammable mixture near the solid surface. This period corresponds to the pyrolysis time and depends uniquely on the material and the heating conditions. If the mixture temperature is increased, either by heat transfer from the hot ambient gas, a pilot or any other mechanism, the combustion reaction between the fuel vapor and the oxidizer gas may become strong enough to overcome the heat losses to the solid and ambient, and become self-sustained at which point flaming ignition will occur. This period corresponds to the induction time and is derived from a complex combination of fuel properties and flow characteristics. Under ideal conditions, introducing a pilot reduces the induction time making it negligible when compared to the pyrolysis time. Thus, the fuel and oxidizer mixture becomes flammable almost immediately after solid pyrolysis starts and therefore, pyrolysis temperatures and times are commonly referred as ignition temperature and ignition time (Quintiere, 1981, Quintiere *et al.*, 1983, 1984).

$$\text{if } t_{in} \ll t_p \Rightarrow t_{ig} = t_p \quad \text{and} \quad T_{ig} = T_p \quad (5)$$

Although such a definition is not physically correct (Alvares *et al.*, 1971) it can be very useful in some practical applications since it not only yields fairly accurate results but also provides a reference parameter that could serve to characterize the ignitability of a solid material. For most practical situations, the flow over the fuel surface will control the mixing of fuel and oxidizer (t_m) as well as the transport (t_r) of this mixture towards the pilot, therefore, can have a significant effect on t_{in} and on the validity of equation (5). The relative effect of the flow on t_{ig} will decrease as the characteristic velocity of the system increases (characteristic time for mixing and transport decrease) and as \dot{q}_e'' decreases (t_p increases).

Before ignition, no heat is being supplied by the flame, $\dot{q}_f'' = 0$, therefore, equation (3) can be reduced to

$$h(T_{ig} - T_i) = \dot{q}_e''(x_f)[1 - \exp(at)\text{erfc}(at)] \quad (6)$$

for values of \dot{q}_c'' lower than a critical value $\dot{q}_{0,ig}''$, the system reaches thermal equilibrium (T_{EQ}) before the surface temperature arrives to the ignition temperature therefore no ignition occurs and for $t \rightarrow \infty$ equation (6) becomes:

$$T_{EQ} = T_i + \frac{\dot{q}_c''}{h}$$

if $T_{EQ} \geq T_p$, ignition is expected to occur and a critical heat flux for ignition can be derived from equation (6) if $T_{EQ} = T_p$, and is given by:

$$\dot{q}_{0,ig}'' = h(T_p - T_i) \quad (7)$$

The practical implications of equations (6) and (7) are many. A characteristic ignition delay time (t_{ig}) results from equation (6). By assuming that

$[1 - \exp(-at)\text{erfc}(\sqrt{at})] \approx \frac{2}{\sqrt{\pi}} (at)^{1/2}$ (for $\dot{q}_c'' > \dot{q}_{0,ig}''$) then:

$$t_{ig} = \frac{\pi}{4a} \left(\frac{h(T_p - T_i)}{\dot{q}_c''} \right)^2 \quad (8)$$

and used to attempt the prediction of t_p (Putorti *et al.*, 1994). It needs to be noted that for this expression to be valid, both mixing (t_M) and transport (t_T) times have to be neglectable when compared to t_p ($t_M \approx t_T \ll t_p$). This will be satisfied best as the external heat flux approaches the critical heat flux for ignition ($\dot{q}_c'' \approx \dot{q}_{0,ig}''$) and $t_p \rightarrow \infty$. This is important because it implies that the error incurred in the experimental determination of t_{ig} , (due to the unknown nature of the flow) will decrease as \dot{q}_c'' approaches $\dot{q}_{0,ig}''$. Therefore, $\dot{q}_{0,ig}''$ is a property of the fuel that can be extrapolated, independent of the flow. Instead, equation (8) could be extrapolated only if the experimental conditions at which “a” and “h” were obtained satisfy the assumption that $t_M \approx t_T \ll t_p$.

From equation (7) a value of “h” can be extracted from the experimental results and is given by:

$$h = \frac{\dot{q}_{0,ig}''}{(T_p - T_i)} \quad (9)$$

As mentioned before, $h(T_p - T_i)$ accounts for the total convective and radiative heat losses. The relative importance of convective and radiative heat losses is extremely difficult to determine, although, if the fuel requires a high external heat flux to ignite or the flow velocity over the fuel surface is very low, it can be assumed that external radiation will be the dominant element and heat losses could be effectively represented as a fraction of the external heat flux that only depends on the fuel and is independent of the flow. For this particular case, h, will become a property of the fuel that characterizes the fraction of the incident heat flux that is absorbed by the fuel. Beyond a critical value, changes in the flow velocity will affect the value of “h” and will serve to quantify the effect that a wind might have on the ignition characteristics of a fuel.

2.1.3 Flame Spread

Once the flame is ignited and propagation starts all terms in equation (3) have to be used to describe the temperature evolution of the solid. For most fuels of interest the term $\sqrt{a\delta_f/V_f} \ll 1$ and if the pilot is not started till the solid has been preheated to thermal equilibrium, the time dependency is eliminated from equation (3) and the following expression describes the flame spreading process

$$V_f^{-1/2} = \left[\frac{\sqrt{\pi}}{2\sqrt{a\delta_f}\dot{q}_f''} \right] [h(T_{ig} - T_i) - \dot{q}_e''(x_f)] \quad (10)$$

It has to be noted, from equation (3), that $V_f \rightarrow \infty$ when $\dot{q}_e'' = \dot{q}_{0,ig}''$, meaning that for an external heat flux of $\dot{q}_{0,ig}''$ the heat from the flame is not needed any more for the reaction to propagate through the solid fuel. Under these conditions, the flame will instantaneously establish over the entire heated surface.

2.1.4 Flame Extinction Mechanisms

The process leading to non-spreading and subsequent extinction of the diffusion flame is the result of a complex combination of the flow field characteristics and fuel properties with a finite chemical reaction. Solutions to the extinction problem have been previously reported, and it has been demonstrated that an appropriate description demands an elliptic resolution of the complete Navier-Stokes equations (Kodama *et al.*, 1987, Chen *et al.*, 1986). Non-spreading or extinction of a flame occurs when the heat generated by a finite chemical reaction (Frey *et al.*, 1979) added to the heat flux from an external source can not balance the heat necessary to increase the temperature of the fuel and oxidizer flow to that of the flame plus heat losses. Among the heat losses are those to the geometrical boundaries, flame radiation to the environment and surface radiation (McCaffrey, 1979).

The solution to equation (1) has a lower limit, $V_f = \pi a \delta_f$, that, under the condition of thermal equilibrium, yields a minimum external heat flux, $\dot{q}_{0,s}''$, necessary for the flame to spread.

$$\dot{q}_{0,s}'' \equiv \dot{q}_e'' = h(T_{ig} - T_i) - \frac{\dot{q}_f''}{\pi} \quad (11)$$

2.1.5 Summary

The above exposed theory represents the basis of an experimental procedure to give a relative assessment of the "flammability" of a fuel (Quintiere, 1981, Quintiere *et al.*, 1983, 1984). Materials can be ranked based on three different principles, readiness to pilot ignition, susceptibility to flame spread and extinction characteristics. The experimentally obtained parameters should be environment independent so as to be considered properties of the material.

By heating the combustible material with a constant heat flux till piloted ignition occurs, a diagram of the characteristic ignition time can be obtained. From these experiments the minimum heat flux for ignition ($\dot{q}_{0,ig}''$) can be extracted and, by using equation (7), incorporated in equation (6) leading to

$$\frac{\dot{q}_e''}{\dot{q}_{0,ig}''} = \frac{1}{[1 - \exp(-at)\operatorname{erfc}(\sqrt{at})]} \quad (12)$$

For simplicity, the approximate expression given by equation 8 can be used for $\dot{q}_e'' > \dot{q}_{0,ig}''$.

Verification of the minimum external heat flux for ignition can be obtained by conducting experiments with variable external heat fluxes of magnitude smaller than $\dot{q}_{0,ig}''$. By substituting equation (7) in equation (10) the following expression is obtained

$$V_f = \frac{\phi}{[\dot{q}_{0,ig}'' - \dot{q}_c'']^2} \quad \text{where} \quad \phi = \frac{4a\delta_f(\dot{q}_f'')^2}{\pi} \quad (13)$$

where ϕ is a material property determined from the experiments and $\dot{q}_{0,ig}''$ can be obtained by increasing the external heat flux till $V_f \rightarrow \infty$, thus, by conducting ignition and flame spread experiments the value for minimum heat flux for ignition can be verified. Reducing the external heat flux will eventually lead to extinction, thus a minimum velocity ($V_{f,min}$) and external heat flux ($\dot{q}_{0,s}''$) for flame spread can be recorded. As mentioned before $\dot{q}_{0,s}''$ does not necessarily coincide with the minimum external heat flux for flame spread predicted by equation (11).

Using the values of $\dot{q}_{0,ig}''$ and $\dot{q}_{0,s}''$, obtained experimentally and equations (12) and (13) a "flammability diagram" (Quintiere *et al.*, 1983, 1984) can be obtained for each material of interest. These diagrams, under known experimental conditions, will serve as a useful way to assess and rank the fire performance of fuels and to identify the parameters that dominate their fire characteristics.

2.2 Experimental Apparatus

2.2.1 The L.I.F.T.

The experimental configuration used as starting point for the design of these experiments is the Lateral Ignition and Flame Spread Test (L.I.F.T.) which is an ASTM standard for the determination of material ignition and flame spread properties (ASTM-E-1321, 1993). Details on the dimensions and geometrical characteristics of this apparatus can be found in the above referred standard.

The procedure and theory behind this test was established by Quintiere *et al.* (1981, 1983, 1984) and consists of two independent tests, a pilot ignition test and a lateral flame spread test. For both tests the fuel sample is placed in front of a radiant panel (483 mm x 280 mm) forming an angle of 15° with the fuel surface with a minimal distance between fuel and panel of 125 mm. The radiant panel provides a heat flux distribution to the fuel surface which is almost constant where sample and specimen are closer and decays as the distance between the panel and the sample increases. Characteristic heat flux distributions can be found in references (Quintiere, 1981, Quintiere *et al.*, 1983, 1984, and ASTM Standards, 1993). The ignition specimen (155 mm x 155 mm) is placed in the region of nearly uniform heat flux and the full sample (155 mm x 806 mm) is used to study the effect of external radiation on lateral flame spread. Recent tests with PMMA have shown that the fuel sample size can be

significantly reduced without any significant changes (Cordova *et al.*, 1997) in the “fire properties” this observation has motivated the reduction of the sample size for the experiments to be conducted with liquid fuels.

2.2.2 The H.I.F.T.

The experimental apparatus described above has been used to study the ignition and flame spread characteristics of liquid fuels on a water bed. The L.I.F.T. hardware had to be significantly modified for this purpose. Figure 1 shows a schematic of the modified hardware. Since both the panel and fuel tray are, in this case, horizontal, the modified hardware is commonly referred as H.I.F.T. (Horizontal Ignition and Flame Spread Test). This geometrical configuration has been previously used to study materials from which the vertical configuration was not convenient (Motevalli *et al.*, 1992).

When the fuel sample is placed parallel to the gravity vector natural convection serves to transport the fuel towards the pilot flame. By placing the sample horizontally the dominant direction for the flow is lost. This change in geometry imposes significant disadvantages to the study of ignition. Natural convection over a horizontal hot surface is much more complex than a vertical natural boundary layer and makes ignition more susceptible to environmental changes and to geometry. For the L.I.F.T. the pilot flame can be placed away from the sample, since buoyancy will carry the pyrolysis products towards the pilot. In the absence of a forced flow (for the H.I.F.T.) it is necessary to place the pilot directly above the fuel surface. Many studies have shown that although pilot flames are the most consistent mechanism for ignition they tend to enhance local evaporation (Ross, 1994, Glassman *et al.*, 1981) therefore spark ignition is generally preferred (Putorti *et al.*, 1994). A number of different pilot locations have been tested to try to show this effect and the results will be presented in following sections. A detailed study that involved changes in pilot location, flow structure and tray geometry lead to the configuration shown in figure 1.

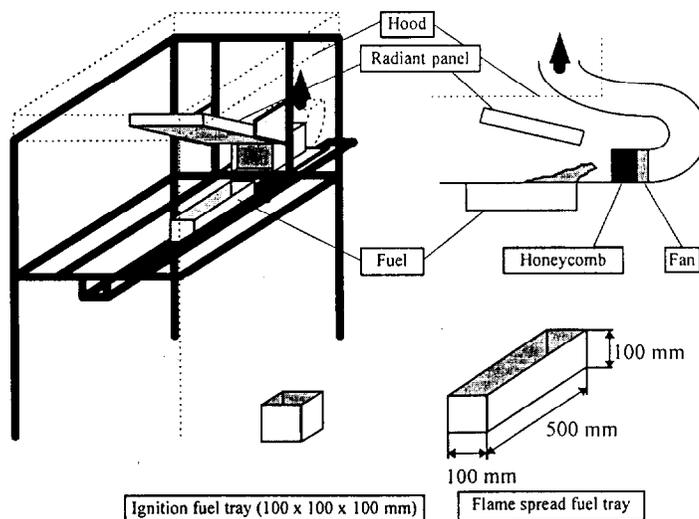


Figure 1. Schematic of the Experimental Apparatus

Consistent results were obtained by including a small fan that induces, by aspiration, a flow with a velocity of 0.1 m/s. The pilot can be removed from the fuel surface and the use of a spark can be avoided. A structured flow is created simulating the natural boundary layer present with the L.I.F.T.

2.3 Mass Burning

A simple way to assess the relative potential of a fuel to sustain mass burning is by using a burning efficiency (χ) extracted from a simple one-dimensional heat conduction model under conditions where the flame characteristics are known. The model relies on the concept that a fraction (χ) of the energy released by the flame is effectively used to support burning of the fuel. The higher the value of χ the more effective the combustion process. The value of χ is independent of the geometry (size, fuel layer thickness, etc.) and is only a function of the fuel. This value represents the mass burning efficiency of the fuel, is independent of the ignition and flame spread parameters and serves as a complement to the H.I.F.T. data.

2.3.1 Formulation

Heat release rate from a pool fire has been documented extensively (Drysdale, 1985, Cox, 1995, Garo, 1996) and it has been found that the expression

$$\dot{Q} = \rho_{\infty} C_p (T_{\infty} g (T_f - T_{\infty}))^{1/2} d^{5/2}$$

correlates well with values measured experimentally (Alramadhan *et al.*, 1990). The total heat release from the combustion process is denoted by \dot{Q} , C_p is the specific heat at constant pressure and ambient temperature for air, T_{∞} is the ambient temperature, T_f is an average flame temperature (for this work $T_f \approx 1100$ K, Cox, 1995), g is the acceleration of gravity ($g=9.81$ m/s²), d (diameter of the fuel pool) is the characteristic length scale, ρ is the density and the sub-index ∞ stands for ambient conditions.

The net heat fed back to the fuel represents a small fraction of the total heat release, this fraction (χ) has been found to be independent of the pool diameter (Drysdale, 1985, Cox, 1995) and thus, the heat flux per unit area reaching the surface can be expressed as

$$\dot{q}_s'' = \chi \frac{4 \rho_{\infty} C_p (T_{\infty} g (T_f - T_{\infty}))^{1/2} d^{1/2}}{\pi} \quad (14)$$

If conduction is assumed to be the dominant heat transfer mechanism and if the thermal wave has not reached the fuel/water interface the fuel can be considered as semi-infinite and an expression for the temperature as a function of time and position can be obtained. This treatment can also be used when fuel and water have similar thermal diffusivities (Arai *et al.*, 1990, Garo *et al.*, 1994). This case will be referred as the "One-Dimensional Single Layer Conduction Model". If fuel and water have significantly different thermal diffusivities and the thermal wave has already reached the fuel/water interface, fuel and water layers need to be treated independently. The water can still be assumed as semi-infinite but the fuel layer needs to be treated as a layer of

finite thickness. This case will be referred as the “One-Dimensional Two Layer Conduction Model”.

Radiation through the fuel layer can be of importance (Alramadhan *et al.*, 1990) but for the fuels of interest it has been demonstrated that most of the radiative heat flux is absorbed very close to the surface (Garo, 1996). Natural convection inside the fuel and water layers can significantly enhance heat transfer close to the fuel surface but seems to affect only weakly steady burning for highly viscous fuels (Ross, 1994). The importance of natural convection decreases with viscosity and its effects can be neglected for those fuels relevant to this study, this is not the case for less viscous fuels (i.e. octane, xylene, etc.) (Garo *et al.*, 1996). For simplicity, this analysis will assume conduction to be the dominant heat transfer mechanism.

Assuming no convective motion and that radiation is fully absorbed at the surface, the following energy balance can be made at $y=y_s(t)$

$$\dot{q}_s'' = H_v \rho_F r(t) + \dot{q}_c'' \quad (15)$$

where $\dot{q}_c'' = -\lambda_F \left. \frac{\partial T}{\partial y} \right|_{y=y_s(t)}$ is the heat conducted into the fuel layer, H_v is the latent heat of vaporization, t is a specific time, T is the temperature, λ is the thermal conductivity, $r(t) = \frac{\partial}{\partial t} (y_s(t))$ is the regression rate, $y_s(t)$ is the location of the fuel surface at a specified time, and the sub-index F stands for fuel. For the entire analysis, it is assumed that the ignition source brings the surface temperature to T_s (vaporization temperature of the fuel) instantaneously. The vaporization temperature is considered to remain constant throughout the entire burning time.

2.3.2 One-Dimensional Single Layer Conduction Model

This analysis is an extension of the works of Arai *et al.* (1990) and Garo *et al.* (1994). Details of the formulation will not be presented here and therefore, the reader is referred to these works for further information.

The heat conduction equation for a one-dimensional semi-infinite element is given by:

$$\frac{\partial^2 T}{\partial y^2} = \frac{1}{\alpha_F} \frac{\partial T}{\partial t} \quad (16)$$

with initial condition at

$$t=0, \quad T = T_\infty$$

and boundary conditions

$$y = y_s(t), \quad T = T_s$$

$$y \rightarrow \infty, \quad T = T_\infty$$

if α_F is the thermal diffusivity of the fuel and “r” is assumed to be constant then the following expression for the temperature distribution can be obtained

$$\frac{T - T_\infty}{T_s - T_\infty} = \exp\left(-\frac{r}{\alpha_F} (y - y_s(t))\right) \quad (17)$$

this expression will be accurate if the thermal diffusivity of the fuel is approximately equal to the thermal diffusivity of water ($\alpha_F \approx \alpha_w$) or for a time period "t" earlier than the characteristic time for the thermal front to reach the fuel/water interface ($t < t_c$). The characteristic time, t_c , can be derived by scaling equation (14)

$$t_c \approx \frac{y_{s,i}^2}{\alpha_F + y_{s,i} r}$$

where $y_{s,i}$ is the initial thickness of the fuel layer. Knowing the temperature distribution it is possible to calculate \dot{q}_C'' and by substituting in equation (15) an expression for the average regression rate can be calculated.

2.3.3 One-Dimensional Two Layer Conduction Model

The thermal diffusivity of water is significantly bigger than that of the fuels of interest and as soon as the thermal wave reaches the fuel/water interface, the water bed starts acting like a heat sink ($t > t_c$). A full description of this scenario is given by the following set of differential equations and boundary conditions.

$$\frac{\partial^2 T}{\partial y^2} = \frac{1}{\alpha_F} \frac{\partial T}{\partial t} \quad (18)$$

$$\frac{\partial^2 T}{\partial y^2} = \frac{1}{\alpha_w} \frac{\partial T}{\partial t} \quad (19)$$

with initial condition at

$$t=0, T = T_\infty$$

and boundary conditions

$$\begin{aligned} y = y_s(t), T &= T_s \\ y = 0, -\lambda_F \frac{\partial T}{\partial y} \Big|_{y=0^+} &= -\lambda_w \frac{\partial T}{\partial y} \Big|_{y=0^-} \\ y \rightarrow \infty, T &= T_\infty \end{aligned}$$

The use of equation (15) and a complex solution of the equations (18) and (19) is necessary to obtain the corresponding temperature distributions and regression rate. The data available in the literature generally provides only average regression rates, therefore, only an order of magnitude comparison will be possible. A simplified approach that incorporates the main physical characteristics is, thus, used to solve the above system of equations in place of a numerical solution.

The average regression rate has been reported of the order of 10^{-5} m/s (Garo *et al.*, 1994) and the thermal diffusivity for the liquids of interest is small, therefore, at each stage of the regression process only the steady conduction equation needs to be solved. The resulting temperature profiles are linear and an equivalent thermal

diffusivity can be obtained. The equivalent thermal diffusivity results from matching the thermal penetration distance through a two layer bed with thermal diffusivities α_f and α_w and the thermal penetration distance in one single layer of thermal diffusivity α_{EQ} . The characteristic length for the fuel layer will be $y_{s,i}$ and the average regression rate, r , is assumed to be constant. From equations (18) and (19):

$$\alpha_{EQ} = \frac{r y_{s,i}}{\alpha_f} (\sqrt{\alpha_w} + \sqrt{\alpha_f})^2$$

The use of an equivalent thermal diffusivity enables formulation of the problem with one differential equation. The single differential equation corresponds to equation (16), where α_f has been substituted by α_{EQ} and the following expression for the average regression rate (r) is obtained:

$$r = \frac{1}{H \nu \rho_f} \left[\chi \left(\frac{4 \rho_\infty C_p (T_\infty g (T_f - T_\infty))^{1/2}}{\pi} \right) d^{1/2} - \frac{\alpha_f \lambda_f (T_s - T_\infty)}{y_{s,i} (\sqrt{\alpha_f} + \sqrt{\alpha_w})^2} \right] \quad (20)$$

Equation (20) although simple and approximate provides an engineering tool that could be of great practical use.

2.3.5 Experimental Apparatus

The experimental apparatus, measurement methods and experimental procedures are those described by Garo *et al.* (1994, 1996) and therefore will only be described briefly here. Pool burning tests of a layer of liquid fuel floating on water were conducted in a large test cell vented by natural convection. Fuel and water were placed in stainless steel pans of 0.15 m, 0.23 m, 0.30 m and 0.50 m in diameter and 0.06 m deep. Some experiments were conducted with pans of different depths to verify that the results were independent of the pan depth. The pans were placed on a load cell to measure the fuel consumption rate. The load cell had a response time of 60 ms and an accuracy within +/- 0.5 g.

3.0 Experimental Results and Discussion

3.1 Ignition

To calibrate the H.I.F.T., SAE 30 W oil was used for ignition and flame spread tests. This fuel was used to make possible comparison with previously reported results on ignition delay time (Putorti *et al.*, 1994) and also because of its high flash point (approximately 250°C). A higher flash point results in a longer ignition delay time providing a longer period to observe the different flow structures formed inside the liquid fuel and on the gas phase. Radiation absorption is also lower with SAE 30 oil favoring boiling of the water bed. Although the viscosity of SAE 30 oil is generally higher than that of crude oils, it is very sensitive to temperature (approximately 1×10^{-2} at 0°C and 1×10^{-5} at 100°C) reaching comparable values after only a small temperature increase.

To assess the effect of the geometry on the ignition delay time different configurations were tested. Two different trays were used, a first tray with a 5mm lip

surrounding the entire upper edge of the tray and a second tray where the lip was eliminated. Both trays were tested with and without a flush floor.

The main assumption regarding the theoretical models presented above is that the fuel will behave as a semi-infinite solid. The same methodology could still be applied if that was not the case but the solution will acquire unnecessary complications. It has been previously observed that the container has significant effects on the formation of recirculation currents inside the liquid. These recirculation currents enhance heat transfer inside the liquid resulting in a more homogeneous temperature distribution (Venkatesh *et al.*, 1996) and in longer ignition delay times. Eight chromel-alumel thermocouples (0.5 mm in diameter) were placed in the liquid with the tip at the center of the tray to verify the effects of heat transfer from the tray towards the fuel. The thermocouples were placed at different depths and spaced to provide a finer grid close to the surface and to cover the entire depth of the tray.

Radiative heat flux from the panel increases the temperature of the fuel but also of the container. The inclusion of a 5 mm lip surrounding the upper edge of the tray increased the solid surface receiving radiation from the panel. The temperature of the upper part of the lip tray was observed to be significantly higher than that observed for the no-lip tray. Thermocouple measurements in the liquid showed that fuel surface temperature was consistently higher for the no-lip tray, while the temperatures recorded deeper in the fluid were higher for the lip tray.

Ignition was consistently observed when the surface temperature attained 254°C. Experiments using the no-lip tray will attain this surface temperature faster than those using the lip tray. Therefore, ignition delay times for experiments using the 5mm interior lip tray were 30% slower than that those corresponding to similar tests with the tray that had no-lip. Despite identical fuel layer thickness and external heat flux, boiling occurred only in the lip tray.

Conduction into the metal walls of the container was suspect of the dramatic ignition delay time differential. A fine metallic powder was used to coat the surface of the sample in both trays. Observations of the flow in the lip tray indicated increased eddy activity of the fuel layer. By selecting the no-lip configuration, effect of conduction through the boundaries was minimized.

It has been shown by Kolb *et al* (1997) that the flow surrounding the flame has a significant effect on the structure of the entrained air, this might lead to significant differences in the ignition delay time as well as on the flame height, once the flame is established. The importance of the mixing (t_M) and transport times (t_T) with respect to the pyrolysis time (t_p) was assessed above. When the hot surface is placed horizontal, characteristic flow velocities are very low, therefore, t_M and t_T can be expected to have a significant effect on the ignition delay time therefore, there is a need to observe the flow characteristics.

To study the air entrainment into above the fuel sample, a 2W red diode laser (SDL-820) was used to create a light sheet for visualization of the smoke emerging from the fuel surface. The images have been processed using an EPIX video card. A threshold value was established below which all pixels were assigned a 0 value, to obtain a more clear image of the flow structure, as evidenced by the smoke. The threshold value is arbitrary and is chosen with the single purpose of eliminating all background images. Figures 2 and 3 are two typical images.

In the absence of a flush floor both trays displayed identical air entrainment and gaseous fuel evolution patterns. Clear eddies could be observed at the edges of the tray (figure 2), as observed these eddies grow to cover the entire surface of the fuel tray. When a floor surrounded the tray the eddies disappeared and a random flow of gases was observed (figure 3). The absence of eddies (for the flush floor case) deters the mixing of fuel and oxidizer at the surface and as a consequence the ignition delay time increased by approximately 20% over experiments conducted with no floor. By introducing a 0.1 m/s flow parallel to the surface a boundary layer is formed and the all eddies were eliminated. By introducing the forced flow t_M and t_T are reduced significantly and t_{ig} approaches t_p . The choice of a small velocity (0.1 m/s) is not arbitrary, as the velocity increases the convective component of “h” increases and will have an effect on the value of the critical heat flux for ignition. This issue will be discussed below.



Figure 2. Smoke Visualization for a Tray with No-Lip and No Flush Floor.

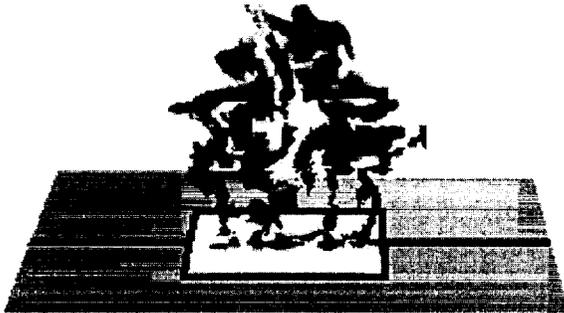


Figure 3. Smoke Visualization for a Tray with No-Lip and Flush Floor.

Premature ignition can be caused by a pilot flame and if the pilot needs to be placed over the fuel surface, it can only be reduced by decreasing the unaccountable radiation from the pilot to the surface. A small propane diffusion flame established on a 4 mm diameter stainless-steel nozzle was used as a pilot. The pilot size was change and its effect on the ignition delay time recorded, it was observed thAt for a specific location on the fuel surface, no change in the ignition delay time could be observed for pilots

smaller than 10 mm, instead extinction of the pilot was commonly seen when fuel ignition occurred. The distance from the fuel surface was observed not to affect the ignition delay time between 10 and 20 mm, decreasing if the pilot is placed closer and increasing as the distance goes beyond 20 mm. It was therefore concluded that the heat contributed by a 10 mm pilot flame at a distance of 10 mm from the surface can be considered negligible. This size and distance from the surface was used to vary the pilot location. The pilot was placed at the center of the tray, at the edges, corners and 10 mm outside the tray. Significant differences among ignition delay times were observed for different pilot locations. Changing the pilot location has no effect on t_p , for pilot flames smaller than 10 mm and more than 10 mm away from the fuel surface, but directly affects the delay for the fuel and oxidizer to reach the pilot, having therefore relating directly to t_M and t_T .

A set of characteristic values is presented in Table 1 to show typical ignition delay times for fixed heat flux (14 kW/m^2) and fuel bed characteristics (SAE 30, layer thickness: 10 mm). All values presented are averages of no less than 5 tests. It can be noted that the ignition delay time is a strong function of all the parameters shown in Table 1. It was concluded that a stable laminar flow is necessary, both to eliminate the need to keep the pilot flame over the fuel surface and to create a robust flow structure that can be considered independent of the environment. A series of experiments were conducted in this configuration, for different heat fluxes, pilot sizes and pilot location and in all cases the ignition delay time remained inside a 7% deviation from the mean.

Pilot Location	Time (sec)			
	No-Lip Tray (with no floor)	Lip Tray (with no floor)	No-Lip Tray (with floor)	Lip Tray (with floor)
0 mm	342	777	1075	1470
50 mm	455	820	415	727
100 mm	510	830		

Table 1. SAE 30 Weight Oil Ignition Tests at 14 kW/m^2

The results from these experiments are presented in Figure 4 together with data obtained for the same fuel by Putorti *et al.* (1994). The ignition delay time is presented as $\tau^{1/2}$, following equation (8). It can be observed that, although the ignition delay time significantly differs from the values found by Putorti *et al.* all data converges to a unique critical heat flux for ignition. Since these experiments were conducted using different ignition procedures and under significantly different environmental conditions, t_M and t_T are expected to be different, thus, affecting the ignition delay time. On the contrary, t_p should not be affected if convective losses are similar in magnitude or can be neglected. As the external heat flux approaches $\dot{q}_{0,ig}''$, t_M and t_T become neglectable compared to t_p and all data converges to a unique point ($\dot{q}_{0,ig}'' \approx 6 \text{ kW/m}^2$), as observed in Figure 4. This observation implies that convective losses are either similar or neglectable for both cases, increasing the value of the forced flow will enhance the convective heat losses and will result in a shift of $\dot{q}_{0,ig}''$. Due to the low velocities

characteristic of this particular configuration, convective heat losses are expected to be negligible, but verification by systematic variation of the velocity still needs to be done.

For the particular case of an oil-slick on a water bed, the water underneath the fuel might attain boiling before ignition occurs. Heating of the bed can be treated as a semi-infinite solid and temperature distributions can be predicted quite accurately (Garo *et al.*, 1994). The analytical prediction of a characteristic time to boiling goes beyond the scope of this work, but the determination of a minimum heat flux that will lead to boiling ($\dot{q}_{0,B}''$) before ignition can occur is of great practical importance therefore needs to be included as a complement to the critical heat flux for ignition.

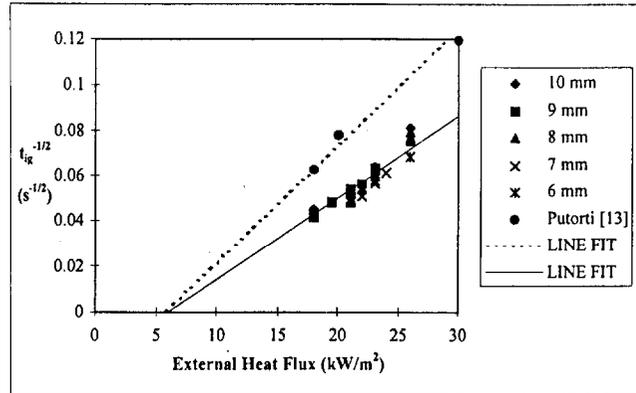


Figure 4. Ignition Delay Time for Different External Heat Fluxes (SAE 30W oil) (The delay times from reference (Putorti *et al.*, 1994) were extracted as an average of the values obtained for 43 mm, 15 mm and 10 mm fuel layers).

As the external heat flux decreases the temperature gradient at the surface decreases and thermal penetration increases before the surface attains T_p . If the thermal wave can increase the water temperature at the fuel/water interface to the boiling point before the surface reaches T_p , boiling will prevent ignition from occurring. The minimum heat flux that will allow the surface temperature to reach T_p before boiling is given by $\dot{q}_{0,B}''$ and presented in figure 5. Under the assumption that convective heat losses are negligible, $\dot{q}_{0,B}''$ can be considered independent of the environmental conditions and only a property of the fuel and the fuel layer thickness. Figure 5 shows the progression of $\dot{q}_{0,B}''$ as a function of the fuel layer thickness. As the fuel layer decreases in thickness, the heat wave will reach the water faster allowing for a shorter available time for the surface to reach T_p and consequently requiring a higher temperature gradient at the surface (higher $\dot{q}_{0,B}''$).

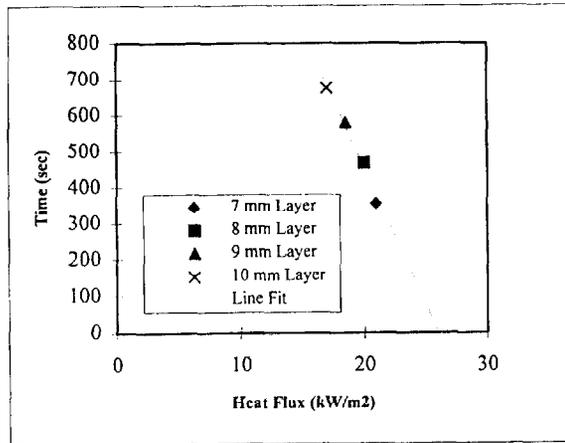


Figure 5. Critical Heat Flux and Time for Boiling (SAE 30W oil)

To further demonstrate the validity of this experimental methodology a series of tests were conducted with Cook Inlet crude oil. Figure 6 shows these results and other obtained for ANS crude oil by Putorti *et al.* (1994). The data presented for the Cook Inlet crude oil is an average of at least five experiments conducted under identical conditions. It was observed that Cook Inlet crude oil in its natural state ignited at ambient temperature, therefore no external heat flux was necessary, when evaporated to a 10% mass loss $\dot{q}''_{0,ig}$ increased to approximately 6.5 kW/m^2 showing a significant increase in the difficulty to ignite. The critical heat flux for ignition extrapolated from Putorti *et al.* (1994) for ANS crude oil (evaporated to a 30% mass loss) was approximately 2.5 kW/m^2 indicating greater ease of ignition than Cook Inlet crude oil (10% evaporated) and being more difficult to ignite than the same oil in its natural state.

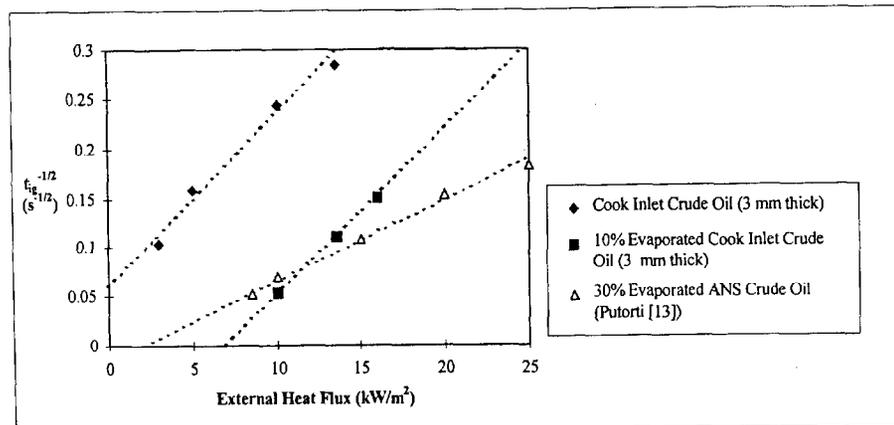


Figure 6. Ignition Delay Time for Different External Heat Fluxes.

3.2 Flame Spread

Several flame spread tests were conducted with SAE 30 W oil to validate the use of the H.I.F.T. It was observed that flame propagation transitions from a continuous flame spread mode, for the higher heat fluxes, to a pulsating mode as the external heat flux decreased. For $\dot{q}_e'' < 6 \text{ kW/m}^2$ propagation ceased. Figure 7 shows a series of characteristic results for SAE 30W oil for different fuel layer thicknesses. It can be noted that flame spread regime significantly exceeds $\dot{q}_{0,ig}''$, specially for thinner fuel layer thickness. The flame spread velocity corresponding to a specific external heat flux increases with the fuel layer thickness showing that the fuel layer thickness has a significant effect on the propagation velocity. Further testing is still required to fully clarify this phenomena. Flame spread over liquid is a complex phenomena that is significantly affected by surface tension and buoyancy driven flows (Ross, 1994, Glassman *et al.*, 1981), the basic criteria presented in equation (13) has been demonstrated to be a viable way to determine the propensity of a fuel to sustain flame spread.

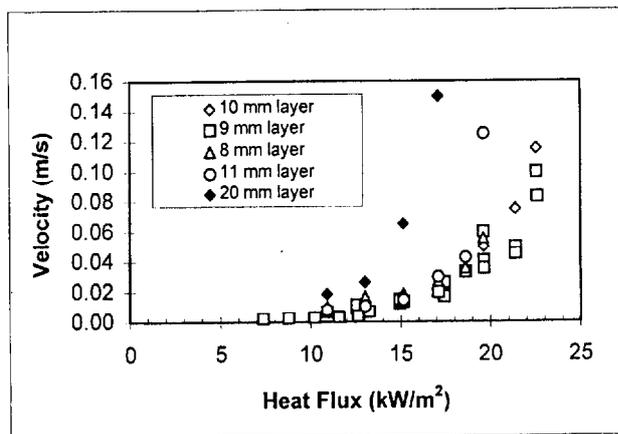


Figure 7. Flame Spread Velocity for Different External Heat Fluxes (SAE 30W).

3.3 Mass Burning

The thickness of the initial fuel layer was varied from a maximum of 20 mm to a minimum thickness of 2 mm. Before each test water was first poured in the pan and next the fuel until it reached 1 mm below the pan lip. During the combustion process the location of the fuel/water interface remained constant, therefore, the freeboard length increases during the experiment. The freeboard length changes were found to have very little effect on the steady-state burning rate.

A typical experiment can be described as a short unsteady ignition period followed by a steady state burning period. During steady burning the surface temperature increased slightly as the experiment progressed (lighter volatile tend to burn off first). The steady burning period was followed by thin layer boil-over characterized by an increase in the burning rate as well as intense splashing of water and fuel. It is important to note that steady burning was followed by boilover and not extinction, therefore, no sudden decrease of the mass burning rate was observed.

The fuels used were heating oil (a mixture of hydrocarbons ranging from C_{14} to C_{21}) and crude oil (63% Kittiwai, 33% Arabian Light and 4% Oural). Experiments were also conducted with weathered fuels as well as different fuel/water emulsions. Weathering refers to the evaporation of the light components of the fuel that results in significant changes of the fuel properties (density, viscosity, boiling temperature, etc.), in the laboratory it was accomplished by means of a mixer turning at 700 r.p.m. and covering 75% of the horizontal cross section of the container. Mixing was conducted for 3 different periods, 24, 48 and 72 hours. Fuel/water emulsions were obtained by adding fixed quantities of water to the mixer and allowed to emulsify for no more than an hour. Experiments were conducted with crude oil aged for 24 hours and water contents of 10, 20, 30 and 40% in volume.

Equation (20) has been used to calculate the average regression rate (r_T) for crude oil. The average regression rate matches well qualitatively and quantitatively the experimental values (r_E). The regression rate is almost constant for $y_{s,i} > 8\text{mm}$ and decreases dramatically with the fuel layer thickness for $y_{s,i} < 8\text{mm}$. For all data points a constant value of $\chi = 2.9 \times 10^{-3}$ was used. The value of χ was selected to best fit r_T with the experimental data for the constant regression rate zone.

Experimentally obtained average regression rates for heating oil, fresh crude oil, 24 hour weathered crude oil and 24 hour weathered and emulsified crude oil (20% water content) were divided by the calculated regression rate (r_E/r_T) to provide an indication of the error associated with the assumptions used to model the average regression rate. The data was obtained for different pan diameters and is a function of the initial fuel layer thickness ($y_{s,i}$) and is presented in figure 8. The predicted values are in excellent agreement with the theory for initial fuel layer thickness greater than 5 mm, for thinner fuel layers the error increases reaching, in the worst of cases, values close to 50%. This error is justifiable due to the great uncertainty present when conducting experiments with very thin fuel layers and to the average nature of the regression rates presented. The value of χ had to be adjusted to $\chi = 3.9 \times 10^{-3}$ for heating oil, to $\chi = 2.4 \times 10^{-3}$ for 24 hour weathered crude oil and to $\chi = 1.8 \times 10^{-3}$ for 24 hour weathered and emulsified crude oil (20% water content).

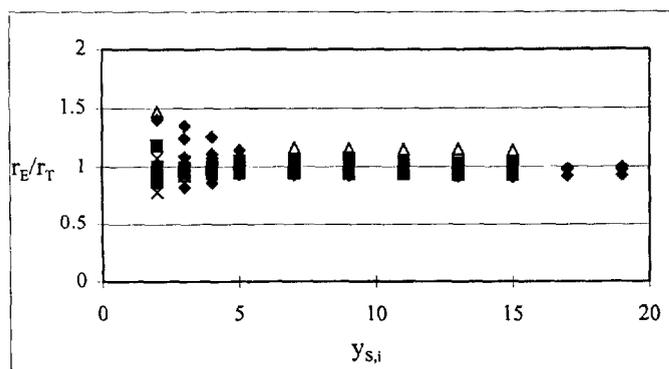


Figure 8. Experimental to Theoretical Regression Rate Ratio for Different Fuels and Pan Diameters

The heat feedback from the flame behaves in a similar way to a pool flame with no water bed, at least in the pre-boilover period. The dependence on the diameter and initial fuel layer thickness is, thus, well described by the heat flux obtained from equation (14). The use of a net heat flux as a boundary condition at the fuel surface seems also to be appropriate. It is important to note that only approximately between 0.18 % and 0.39 % of the energy released by the flame is effectively fed back to the fuel surface. The above values of χ seem comparable to data presented by Arai *et al.* (1990) but no data obtained under similar experimental conditions has been found to verify these magnitudes.

As previously pointed out by many authors (Arai *et al.*, 1990, Garo *et al.*, 1994, 1996, Alramadhan *et al.*, 1990) the water bed acts as a heat sink. The thermal diffusivity of water is significantly larger than that of the fuel thus, as the fuel layer becomes thinner, the overall thermal diffusivity increases. Heat conduction through the fuel and water increases (\dot{q}_c'') and the overall fraction of the total heat flux (\dot{q}_s'') vaporizing the fuel decreases leading to a decrease in the regression rate that can eventually result in extinction ($y_{s,i} < 2$ mm).

The data for weathered and emulsified crude oil is also well described by the predicted regression rate. The same reasoning presented above applies for these cases, being the only difference the regression rate magnitude. Weathering and emulsification alter the thermal properties of the fuel and, thus, the magnitude of the regression rate. By changing the efficiency constant (χ) to fit the experimental data, a practical way is found to incorporate the effect of weathering and emulsification on the fuel properties. The variation of the efficiency constant as a function of the weathering period and the water content is shown on figures 9 and 10, respectively.

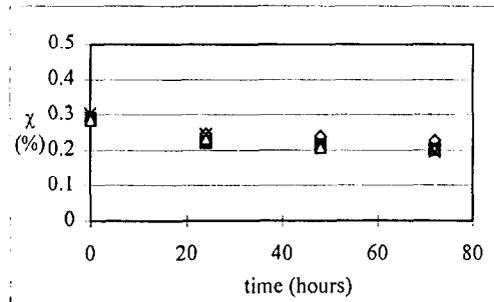


Figure 9. Efficiency Factor as a Function of Weathering Time

Figure 9 shows that initially the efficiency constant (χ) decreases fast, followed by less significant changes till it reaches an almost constant value ($\chi = 2 \times 10^{-3}$). It is well known that the highly volatile hydrocarbons will evaporate very fast, i.e. after less than 24 hours the mass loss of hydrocarbons with boiling points below 500 K ($<C_{11}$) has reached 95% and only reaches total evaporation after 48 hours (Demarquest, 1983). Heavier hydrocarbons ($C_{11}-C_{25}$) tend to evaporate significantly slower reaching 100% mass loss only after more than 10 days. The initial fast change in fuel properties results in an abrupt decrease of χ , followed by an almost negligible change rate.

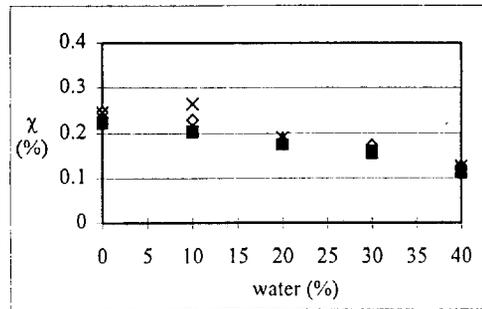


Figure 10. Efficiency Factor as a Function of the Water Content (emulsification)

For emulsified fuels water addition will result in an almost linear decrease of χ (figure 10). The properties of the emulsified fuel change significantly with the water content and, as shown by equation (20), this will have a significant effect on the average regression rate. The way in which emulsification affects the fuel properties is not fully understood but it is well known that properties such as the density vary in an almost linear way with the water content ($\rho(\text{emulsified fuel}) = (1 - \text{water fraction})\rho_f + (\text{water fraction})\rho_w$) and other properties, such as viscosity, increase in a non-linear way.

Although the efficiency constant (χ) does not provide a real explanation to the effects of weathering and emulsification on the average regression rate it serves to quantify the flammability of the fuel independent of the pool size and fuel layer thickness. The relationship between the fuel properties and the efficiency factor goes beyond the heat transfer and evaporation mechanisms controlling the burning rate and a complete explanation will require a comprehensive study that will include the effects of weathering and emulsification on the flame chemistry and radiative feedback. In detail analysis of these relationships go beyond the scope of this work.

4.0 Conclusions

The methodology to assess the burning characteristics of a liquid fuel on a water bed has been presented and verified with different fuels, weathering conditions and water content. Three different and complementary tests are deemed necessary to characterize the three different regimes of the burning process: ignition, flame spread and mass burning.

For ignition; the critical heat flux for ignition as identified in ASTM E-1321 was found to be the parameter that better describes the capability of a fuel to ignite. The critical heat flux for ignition was found to be independent of the geometry and flow conditions and a parameter that could be extrapolated to attempt the characterization of the ignition for more realistic length scales. The minimum heat flux for ignition needs to be accompanied by a minimum heat flux that will lead to boiling of the water bed. The ignition delay time, although a relevant parameter of the ignition process, was found to be dependent of the experimental conditions, and thus, difficult to extrapolate to a different scale.

For flame spread; the minimum external heat flux that will sustain propagation together with the parameter ϕ (function of the fuel properties) will serve to describe the

flame spread characteristics. The complexity of flame spread over liquid fuels makes necessary further validation of this experimental approach.

For mass burning; the efficiency factor, χ , serves as unique parameter to characterize the regression rate during the mass burning process. χ is a property of the fuel that can be extrapolated to different scales and environmental conditions but should be evaluated under conditions where is independent of the flame diameter and the fuel layer thickness. The validity of this approach is constrained to the pre-boilover regime and the assessment of its relevance to the time for the onset of boilover requires further experimentation.

5.0 Acknowledgments

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