

POOL FIRES IN A LOW VENTILATION ENCLOSURE

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A 1.9 m^3 enclosed fire test facility was developed with separate entrained air inlet at floor level and fire product exit at ceiling level. A new ventilation parameter, K_{in} , is proposed as the ratio of the air inlet flow area, A_{in} , divided by a mean enclosure cross sectional area, $V^{2/3}$.

$$K_{in} = A_{in} / V^{2/3}$$

It is shown that this relates to realistic fire scenarios and to investigations by other researchers. A 200mm square 500ml (400g, 18MJ) kerosene pool fire was used as the fire load with three air supply inlet sizes, no air, 0.0017m^2 (K_{in} = 0.11%) and a 0.016m^2 rectangular hole (K_{in} = 1.0%). The range of air inlet area coefficient, K_{in} , investigated here is shown to simulate the situation of a closed room with normal door and window air leaks. This is shown to be smaller than investigated in other enclosed fire studies, which normally represented the situation of an open door or window at the start of the fire. The pool fire load rate of mass consumption was determined using a fire load base mounted on three load cells. The rate of fire load mass loss together with the calorific value for kerosene were used to determine the fire heat release rate, which was corrected for combustion efficiency based on the energy content of the CO and UHC in the fire outlet gases. The heat release rate per unit pool surface area was shown to be comparable with other measurements in larger enclosures with bigger pool diameters. The maximum heat release rate was higher than for a free pool fire of the same pool size due to the additional heat transfer to the pool from the enclosure. The mean near ceiling fire temperature was determined using an array of Type K thermocouples 70mm from the ceiling. The heat release rate, CO, UHC and fire enclosure ceiling layer temperature for the same pool fire size was found to be highly dependent on the ventilation rate. A self extinguishing pool fire was demonstrated for the minimum air supply rate.

Keywords**:** Compartment Fire, Pool Fire, Heat Release Rate, Ventilation Control

INTRODUCTION

Fires in under-ventilated enclosures have heat release rates that are strongly dependent on the fire air consumption rates. Often ventilation controlled fires are investigated in test rigs that have a simulated open doorway or window at the start of a fire (1). This work is concerned with more severe ventilation control and is designed to simulate the fire conditions in an enclosure with doors and windows closed and no failure of the enclosure boundary during the fire. The present work investigates air in leakage rates that are equivalent to natural door, floor and window air leakage gaps. The test facility used was similar to that of Gottuk et al (2,3) but with smaller air inlet flow areas relative to the compartment volume. The same technique of separating the air inlet path and the fire product outlet was used in the present work. A fire product leakage through the ceiling was simulated, but the fire development is not strongly dependent on this location of the exit. Gottuk (2,3) used a fire product exit through a high window that was completely contained in the hot gas ceiling layer.

In the initial stages of a fire there is little need for air entrainment from the outset as there is usually sufficient air in the enclosure. The importance of this stage of the fire depends on the ratio of the mass of air in the enclosure to fire load mass, if the fire load mass is small relative to the enclosure air mass (<1/15 for hydrocarbons) then there may be no need for further air entrainment into the enclosure as the initial air is sufficient to burn all the fire load.

However, this is rarely the case unless the enclosure is very large, such as a hotel atrium, and often the fire load mass is greater than the initial air mass and considerable additional air is required to be entrained into the enclosure to burn the fire load. In the later stages of enclosed fires the heat release rate becomes limited by the air entrainment and for steady state fires the air entrainment can equal the air consumption. A limiting fire case is that of a relatively large enclosure and a small fire, with no external air supply. There is sufficient air to sustain a fire but eventually the oxygen depletion will extinguish the fire if it is a hydrocarbon fire. This situation was investigated in the present work in addition to two air ventilation sizes. Cellulose type fires can continue to smoulder using their fuel bound oxygen.

Heat release rate in enclosed fires were determined in the present work using the determination of the fire load mass loss rate with the fire load resting on load cells. This was preferred to oxygen consumption calorimetry as it does not require the mass flow of the exhaust gases to be determined, as in the case of the cone calorimeter and whole room fire calorimetry (4,5). A method of using oxygen consumption calorimetry is being developed by the authors (6) for the present type of low ventilation fires, by determining the air consumption mass flow rate from the gas analysis based air/fuel mass ratio and the measured fuel mass loss rate.

Pool fires were used in the present work for two main reasons. Firstly, there is a requirement from offshore fire hazard evaluation work to understand more about hydrocarbon pool fires in enclosures and the production of toxic products (7). Secondly, pool fires are fairly reproducible fires and repeat tests at different air ventilation rates or hydrocarbon volatility can readily be undertaken. They represent real fires in enclosures with a time development of the heat release rate and a gradual increase with time of the external air entrainment. These features of real fires cannot be represented in the burner flame fires (4, 8, 9) or forced steady air flow ventilation fires (10,11) used by some investigators of enclosed fires. Pool fires have been used for the study of enclosed fires by several investigators. The work of Gottuk et al (2,3) is very relevant to the present work as they used a similar test facility with hexane pool fires. The test facility of Fleischmann and Parkes (1) was also similar to the present work and they used similar ventilation openings with 200mm diameter heptane pool fires of a very similar size to the present kerosene pool fires. Audouin et al. (12) undertook large scale room fire experiments with a square metre solvent pool fire in under-ventilated conditions and showed that the flame could detach from the pool and move towards the air supply, a phenomena known as a ‘Ghosting Flame’ (13, 14). The work of Chamberlain (7) for large scale diesel pool fires was similar to the present work and intended to measure the heat release rate from the mass consumption of the fuel. Unfortunately in their tests the load cells were damaged in the tests and this could not be done, only an average fire heat release rate was determined. They measured CO, CO_2 and oxygen concentration and wall heat fluxes and gas temperatures in the tests. Restricting the ventilation was shown to markedly increase the CO emissions from 0.5% to 4% at the peak value. Also the reduction in ventilation was shown to increase the peak soot concentration by a factor of three.

AIR IN LEAKAGE FLOW AREAS DUE TO DOOR, WINDOW AND FLOOR GAPS

If a typical small room is considered with size 3 x 5 x 2.5 m the volume is 37.5 m^3 . If this has a 2 x 1m door with a 1mm gap around the sides and top and a 10mm gap at the bottom then the air in-leakage area is 0.015 m^2 . If in addition there is one window that opens and this window is say 0.5m square then there could be a 1mm air leakage gap around the window. This would add 0.002 m^2 to the air leakage area. For a wooden floor further leaks could occur, perhaps equivalent to the door leakage rate. BS 5566 Part 4 gives the leakage area for different doors as 0.01 - 0.06 m^2 and for windows with a 2m perimeter 0.0005 m^2 . This shows that the above estimates of air leakage areas in closed rooms are reasonable. In industrial equipment such as an offshore module air in leakage rates are greater. Industrial equipment often has several pipes or other components passing through walls and they all can introduce additional leakage paths. It is the objective of passive fire protection in compartmentation to minimise these external air leaks. It is reasonable to use the above figures to estimate that a typical natural air in leakage flow area into an otherwise closed room would have an area of between about 0.01 - 0.06 m^2 . If in the above example the window was open or failed in the fire than the air inflow area would be 0.25 m^2 .

The air in leakage area needs to be related to the room volume as it is the air inflow relative the volume of air in the room that is important in fire development. In this paper we propose that an appropriate dimensionless parameter for scaling the air inlet ventilation area would be the air inlet area coefficient defined as

$$K_{in} = A_{in} / V^{2/3} \qquad (1)$$

where A_{in} is the inlet area and V is the volume of the enclosure. Hence $V^{2/3}$ is an enclosure mean cross-sectional area. This parameter is similar to the inverse of the vent coefficient, K_v , as used in explosion venting methodology. K_{in} is an air inlet blockage parameter that is independent of the geometry of the enclosure and the location of the air vent. This will be used in the present work as the air inlet flow as a % of the mean cross sectional area of the enclosure and can then be used to scale or compare this work with other ventilation controlled fires.

In the above example for a room this, air inlet ventilation porosity, K_{in} , would range from 0.09% to 0.6% for the BS 5566 Part 4 range of door leakage areas. For the open widow case the K_{in} value would be 2.25%. In the present work K_{in} values of 0, 0.11% and 1.0% were investigated. Most other workers have investigated ventilation controlled enclosed fires with K_{in} values that are close to the above example of an open window or higher than this. The range of K_{in} investigated for enclosed pool fires by other workers (6) is 1.9% (12), 4.37% (2,3), 9.2% (1), and 7.5 or 21% (7). They are all relatively well ventilated fires compared with the present work and mainly consider the situation of a window or door open at the start of the fire.

EXPERIMENTAL TECHNIQUES

A 1.9 m^3 internal volume enclosed fire test facility, 1.4m x 0.96m x 1.5m external dimensions, was used with separate entrained air inlet at floor level and fire product exit at ceiling level, as shown in Fig.1. This enclosure had an initial air mass content of 2.3 kg at ambient conditions (density 1.2 kg/m^3). The stoichiometric amount of kerosene that could theoretically burn with this air without any requirement for air to flow into the enclosure was 0.155 kg using a 14.7/1 stoichiometric air/fuel ratio by mass. In practice only approximately one third of this mass (50g) could burn due to self inerting of the enclosure atmosphere by the fire products and this will be demonstrated later.. The kerosene pool fire fuel load was a 200mm square pool with between 0.30 and 0.60 kg of kerosene as the initial pool fire load. Higher fuel loads were used for the highest ventilation. This gave an initial enclosure air/fuel ratio of between 7.7/1 and 3.8/1 and air entrainment from outside the enclosure would be required to consume all of the fire load. However, there was sufficient air initially to have a significant fire with no additional air supply and this was one of the fire scenarios investigated. This ratio of the enclosure air mass to fire load mass is one way of assessing the severity of a fire load and the need for external air inflow to sustain the fire.

The fuel load base was at an elevated position of 250mm above the chamber floor. The air inlet and the fire product outlets were arranged to be through long thin slot holes at the periphery of the fire load floor and ceiling level respectively. This was to minimise any fire shape distortion due to the air flow inlet or product exit and hence to make the test situation amenable to modelling. The area of the air ventilation distribution slots surrounding the fuel tray (see Fig. 1) was 0.1 m^2 which was always greater than the air inlet area so that the air flow was controlled at the inlet to the plenum chamber (the largest air inlet was 0.016 m^2). The fire enclosure had a flat steel roof with no insulation and the fire products were drawn off by natural convection at the edges of the roof into a collection volume above the steel roof where they were extracted through a 150mm diameter duct into a dump volume where a large hood and variable speed fan extraction system transported the fire products and discharged them through a 10m chimney. This is a relatively restricted outlet for the fire and was used because it was desired to study the closed enclosure situation. In all fires the air inlet was the flow restriction and there was no case of backflow of fire products through the air inlet in the present work.

Three air supply inlet sizes were investigated, no air (K_{in} = 0), a 46mm diameter hole, 0.0017m^2 (K_{in} = 0.11%) and a 0.016 m^2 rectangular hole (K_{in} = 1.0%) . The aim was to demonstrate the major differences in the fire development due to different air ventilation levels. If the front panel air inlet control was removed and no inlet air restriction was used other than the slot around the periphery of the floor then the air vent area ratio, K_{in} ,would be 6.25% which is still a high ventilation control and is typical of that for a small window in a room, as shown above. For enclosures with a door and/or large window ventilation, higher values of this parameter occur and this is not the case in many fires as they start with initially closed enclosures with only air leakage paths available and this is the situation investigated in this work.

The fire temperature was determined from an array of Type K 3mm diameter mineral insulated exposed junction 1mm diameter bead thermocouples. These were placed 70mm from the ceiling and the average fire ceiling temperature as a function of time was determined for the central region of the ceiling fire. The temperature profiles along the ceiling were quite uniform and the average temperature was used to investigate the influence of K_{in} on the fire ceiling temperature. The peak temperatures were 180C, 300C and 500C for K_{in} = 0, 0.11 and 1% respectively and full details of the fire temperatures and heat losses are given in Ref.15.

The fuel mass loss rate was determined by placing the fuel in a tray on a platform that rested on three load cells in a triangular configuration. The fire platform was supported from the load cells on three long legs protruding through the air supply plenum below the fire enclosure. This enabled the load cells to be mounted on the bottom steel wall of the air plenum and were kept cool by the air in the plenum. There were no problems with the fire temperatures affecting the load cells and a clean steady depletion of the fire load was determined with no random fluctuations due to overheated load cells. The load cells had a 1 gram resolution and a 10 kg maximum capacity. The load cell output, all the thermocouple outputs and the gas composition outputs were all recorded on an IQtech Tempscan 1000 data logger interfaced to a PC and processed using Excel. Thirty two channels of data were scanned simultaneously every second.

The side walls of the fire enclosure were made from insulating fire resistant material, 25mm thick Triton Kaowool 1260 insulation board. One side of the enclosure had a full view high temperature glass window with a door cover made of Triton Kaowool. This window was closed in these experiments with the insulation door shut. The whole enclosure was in a steel box and the corner joints were air sealed as was the front wall window and insulating door, using a high temperature flexible gasket. The only air inlet point was the front inlet port and by changing the front panel the size of this opening could be changed.

A water cooled multi-hole 'X' gas sample probe, with sample holes on centres of equal area, was inserted in the 150mm diameter fire ceiling outlet duct to obtain a mean gas sample from the ceiling layer without any external air entrainment. This has been shown to give very similar gas composition results to that of the ceiling layer, but was a better mixed sample and was a more reliable measurement of the mean gas composition of the ceiling layer. The gas sample was passed through a 2C cooler and filter prior to non-dispersive infra red analysis for carbon monoxide and paramagnetic analysis for oxygen. A heated sample line and heated pump was used to transport the sample without condensation losses to a heated flame ionisation detector for unburnt hydrocarbon (UHC) measurements as ppm methane equivalent.

FUEL MASS CONSUMPTION RATE AND HEAT RELEASE RATE FOR THREE VENTILATION RATES

The fuel mass loss as a function of time is shown in Fig. 2, normalised to the initial pool fire mass, for the three air ventilation conditions. The rate of fire load mass consumption in the fire was used to compute the heat release rate from a mass consumption rate times calorific value computation, as shown in Fig.3 for the largest air ventilation. These results have been corrected for the combustion inefficiency based on the CO and UHC energy content, as detailed in Ref. 6. Any residual CO or hydrocarbon gases in the fire vent gases represents unreleased fuel energy and hence the heat release rate should be corrected for the fire combustion efficiency (6), as shown in Fig.3. The heat release rates corrected for combustion efficiency are shown in Fig. 4 for all three air ventilation conditions. These results clearly show a major influence of the air ventilation on the heat release rates. All the fires had a combustion efficiency greater than 90% so that the maximum correction of the heat release rate for combustion efficiency was 10%.

The results with no external air ventilation show that the flame went out with only 20% of the fuel load burnt and this occurred after about 250s. For both of the ventilated fires all the fuel load was burnt out. The highest ventilation rate (K_{in} =1.0%) had the greatest heat release with a maximum of 70 kW. The lower ventilation rate (K_{in} =0.11%) had all the features of a steady state fire with a near constant heat release rate of about 15 kW from 100s to 900s All three fires had a similar heat release rate in the first 100s, indicating that the air in the enclosure dominated the fire propagation in this period.

The maximum heat release rate of 70 kW converts to 1.75 MW/m^2 of pool surface area or 55.5 kW/m^3 of enclosure volume. For comparison the mean heat release rates in the large scale test of Chamberlain (7) using diesel pool fires were from 0.82 to 1.81 MW/m^2 of pool surface area (using a CV of 43 MJ/kg). On a chamber volume basis the average heat release in the work of Chamberlain was from 36 to 168 kW/ m^3 and the present peak heat release rate is within this range. Audouin (12) also reported the mean heat release rate for enclosed pool fires and this was 0.8 MW/ m^2 of pool area or 8.53 kW/m^3. Thus the maximum heat release per unit pool surface area or per unit enclosure volume in the present work with the highest ventilation rate was similar to that in the much larger experiments of Chamberlain (7) and larger than that in the work of Audouin (12). This indicates that the present relatively small scale tests can yield information applicable to much large fire enclosures. The load cells did not survive the fire in Chamberlain's experiments and the time variation of heat release was not given, nor was this measured by Audouin. The enclosed pool fire experiments of Gottuk (2,3) and Fleischmann and Parkes (1) did not report the heat release rate.

For kerosene pool fires in the open with a pool diameter of 0.2m the heat release rate is 0.86 MW/m^2 but this increases to 1.68 MW/m^2 for 2m pools (16) and for pools in the 20-80m diameter range the heat release is 2.58 MW/m^2 (17). For comparison with the present work using 0.2m pool size the relevant open pool fire heat release rate is 0.86 MW/m^2 (16) and the present maximum heat release is 1.75 MW/m^2 The increase in the heat release rate is due to the radiation feedback to the pool from the enclosure walls. In large diameter open pool fires a similar effect occurs from greater internal radiation from mainly soot in the flame to the pool surface.

The above results show that these fires were ventilation controlled. Haselden et al. (18) gave a criterion for ventilation control that involved a fire load greater than 150 kg/ m^2 of air inlet vent area. In the present work the value of this parameter was 176.5 kg/ m^2 for K_{in} = 0.11% and 37.5 kg/ m^2 for K_{in} = 1.0 % and both experiments were clearly ventilation controlled. For pool fires the load can be increased by increasing the pool depth without increasing the burning rate, which depends mainly on the surface area. Thus the fire load per unit air vent area is not a relevant parameter for pool fires and ventilation control can be achieved at lower values than those recommended by Haselden et al (18). A pool surface area to vent area ratio would be a more appropriate parameter to determine whether a pool fire was ventilation controlled and a criterion that this ratio should be of order unity (>>0.1) or higher for ventilation control. This is because the pool fire mass burn rate is surface area related and the air entrainment is related to the ventilation air inflow area and if the air inlet area is very large relative to the pool area then the pool fire will be closer to an open pool fire with a flat roof above the pool but little restriction on the air inlet. In the present work for K_{in} = 0.11% and 1.0 % this ratio is 25 and 2.5, compared (6) with 1.2 and 0.44 in the work of Chamberlain (7), 6.25 in the work of Audouin et al. (12) and 0.24, 0.57 and 0.84 in the work of Gottuk et al. (2,3). These values demonstrate again that the present work was more ventilation controlled then that of Gottuk and Chamberlain and similar to that of Audouin et al. For ventilation with a door or widow open the pool/vent area ratio would be in the range 0.1-1 and for closed window or door natural ventilation conditions the ratio would be >1.

GAS COMPOSITION RESULTS FOR THREE VENTILATION RATES

The oxygen, carbon dioxide, carbon monoxide and total hydrocarbons volumetric concentrations in the fire enclosure ceiling exit plume are shown as a function of time in Fig. 5 for the three air ventilation cases. These show that the first 200 seconds all the fires were practically the same and hence independent of the ventilation rate and dependent on the air already in the enclosure. All the four gases behave in the same way in this period at the end of which the oxygen has been depleted to 15.5% and the CO is 0.1%. For the case with no air ventilation the pool fire goes out through lack of oxygen. This 15.5% oxygen for extinction is caused by self inerting of the flame using the CO_2 and H_2O products of combustion as inerts and the 15.5% oxygen limit is close to the oxygen flammability limit for CO_2 inerting, which is 14.4% for higher hydrocarbons (19).

For the ventilated fires the oxygen as measured at ceiling level, continued to be depleted, but the fire entrained air at floor level sustains the flame. However, the higher oxygen supply increased the heat release rate which consumed more oxygen. The heat release for the highest air ventilation reached a maximum at 350s into the fire and this was accompanied by a reduction in the oxygen level below that of the intermediate air ventilation case (K_{in}= 0.11%). The CO and UHC emissions suddenly increased in this period of maximum heat release. This indicates that the fire was air starved and would be at its richest equivalence ratio (6). The intermediate air ventilation had a steady heat release rate throughout the fire, after the first 150 seconds, as shown in Fig.4, and the air supply was just sufficient to match that required for the 15 kW heat release. Fires require 3 kg of air per MJ of heat released and hence the 15kW fire requires an air consumption of 0.005 kg/s. A calculation of the air entrainment for mean fire temperature of 200C and a vertical height of 1.5m above the air inlet shows that this is the entrained air flow. This fire did not become highly air starved as the CO emissions did not increase beyond 0.3% and the UHC emissions remained relatively low at below 3000 ppm.

It was shown above that most other investigators of pool fires in enclosures have used higher ventilation rates than in the present work and the higher heat release rates have consumed more oxygen and they have achieved near zero oxygen levels at the peak fire intensity (2,3. 7, 12).. Nevertheless the shapes of the gas concentrations as a function of time are similar to the present work for the highest ventilation, K_{in}= 1.0% (2,3, 21). Gottuk (2,3) found CO levels of 3% at 0% oxygen, Audouin (12) found 1% CO at 0% oxygen and Chamberlain found 4% CO at 0% oxygen. The peak CO in the present work was 1% at 8% oxygen. Chamberlain found for a test with a 280% increased ventilation opening that the CO emissions decreased to 0.5% and the oxygen increased to 5%. The present results together with those in the literature show that CO and oxygen levels are highly dependent on the air ventilation. Under-ventilation (K_{in} < 2.0%) can produce high oxygen and low CO and a trend of increasing CO with increase in air ventilation as in the present work, due to low heat release rates. Over-ventilation (K_{in} > 2.0%) can also reduce the CO and increase the oxygen due to excess air ventilation and a trend of decreasing CO as the air ventilation is increased, as in the work of Chamberlain (7). In between these situations there is a worst case global rich combustion scenario (2,3, 7, 12). The results are also influenced by the pool volatility in that more volatile liquids will give a richer global mixture at the same ventilation rate. Further work is in progress on a wide ranging study of ventilation rates on pool fires.

The high peak CO and UHC levels in Fig.5 for K_{in} = 1.0% represents a significant combustion inefficiency and this was probably due to incomplete mixing in the fire, with the plume richer than stoichiometric, generating locally high levels of CO and UHC which then did not mix adequately with the available oxygen to oxidise them prior to the fire exit. Also the fire temperature in the ceiling layer was too low (500C maximum) to give rapid CO and UHC oxidation. The high level of oxygen shows that the global fire equivalence ratio is very lean and the associated high CO and UHC emissions shows that the recommendation in Ref. 20 that CO emissions can be taken as zero for lean mixtures is not supported in the present work.

The total hydrocarbon emissions in Fig. 5d show that in the early stages of the fire up to 200s the UHC was low for all ventilation rates as there was ample oxygen for a high combustion efficiency. After 200s the UHC trends differed for the three air ventilation rates. After the pool fire went out (at about 200s) for the case with no air ventilation there was considerable vaporisation of the kerosene by the heat in the enclosure and the UHC was at a high level of nearly 8000 ppm. For the case with the highest ventilation there was a similar large increase in UHC even though the heat release was at a maximum, as shown in Fig.4. The peak UHC was in excess of 1% and this could not be resolved by the FID that was being used, a sample dilution system is required to do this. The peak hydrocarbons would be of the order of 2% (by extrapolation of the data). This was accompanied by high CO levels and this indicates that there was a poor combustion efficiency. The smaller ventilation case showed much lower UHC emissions and the main difference was the lower rate of heat release and associated higher residence times. It is the energy content of the CO and UHC emissions that combine to give the combustion efficiency correction to the heat release rates in Figs 3 and 4.

The hydrocarbon levels in Fig.5d were high in the later stages of the fires at low ventilation rates, but the UHC burn out was very rapid for the highest ventilation rate. This was due to the higher temperatures and greater oxidation rate. The vaporised kerosene fumes after the flame went out with no ventilation are very high and could be a toxic problem, although a speciation of the hydrocarbons would be required to access the fire products toxicity. Furthermore, the higher concentration of UHC combined with the high CO levels (which has wide flammability limits) explains the higher risk of backdraught associated with air starved fires.

MEASURED CEILING LAYER TEMPERATURES

The mean central ceiling temperature 70mm form the ceiling was determined using an array of thermocouples. The mean temperature as a function of time is shown in Fig. 6 for the three ventilation conditions. The mean temperature increased markedly as the ventilation rate was increased, from 180C for no ventilation to 280C for K_{in} =0.11% and 500C for K_{in} =1.0%. The peak temperatures measured by Chamberlain (7) were much higher at 1100C for K_{in} =21.2.% and 1200C for K_{in} =7.52.% and the higher ventilation rates had a major influence on these higher temperatures. However, the heat losses would be lower in the larger enclosure of Chamberlain (7) as heat losses as a proportion of the heat release are related to surface to volume ratios which decrease as the enclosure increases in volume. For a cubic enclosure the ratio of surface to volume is 6/x where x is the length of the side of the cube and the Chamberlain test rig was of the order of four times the dimensions of the present rig. However, it is the much higher heat release in the Chamberlain work that governs the higher temperatures. The present results were similar to those of Peatross and Beyler (10) for 0.62 and 0.84m diameter diesel pools in an enclosure much larger (34.2 m^3) than the present with a pool surface area that was 2.69% and 4.94% respectively of the floor area, compared with 3.0% in the present work and 17.3 or 34.6% in the work of Chamberlain (7). For a ventilation rate of 0.28 kg/s Peatross and Beyler (10) found an enclosure temperature of 190 C for the 2.69% pool to floor area ratio, which increased to 260C for the 4.94% pool size and the same air ventilation. These are lower temperatures than for K_{in} =0.11% in the present work.

The air ventilation rate was 0.28 kg/s in the work of Peatross and Beyler (10) compared with 0.04 kg/s in the present work (6). This gives an enclosure volume to air flow rate or residence time of 47s in the present work and 146s in the work of Peatross and Beyler. This much longer residence time would result in larger relative heat losses in their work and hence lower temperatures. In the work of Chamberlain the residence time was 17s and 14s and these tests would be associated with much lower heat losses than in the present work which was a factor in the higher temperatures.

CONCLUSIONS

Enclosed fire tests that simulate fire in rooms with initially closed doors and windows have an air inlet area coefficient, K_{in} , that is less than 0.4%. this increases to typically 4.5% for an open window. Most previous investigations of enclosed fires are for the open window or door situation. The present work was directed at the initially closed room situation with natural air leaks, and three inflow area were investigated with K_{in} values of 0, 0.11% and 1.0%. The heat release rate increased as the air ventilation increased and this was accompanied by a large increase in CO and UHC. This indicated a significant combustion inefficiency despite the high levels of oxygen which would translate lean global equivalence levels, which does not support current recommendations (20) that CO emissions (in these conditions) can be taken as zero.

The relatively small scale tests in the present work yielded pool fire data that was comparable with the much larger enclosed fire tests of Chamberlain (7). The maximum heat release was 1.75 MW/m^2 of pool surface area or 52.2 kW/m^3 of enclosure volume, compared with 0.8 - 1.8 MW/m^2 and 36 - 168 kW/m^3 for the work of Chamberlain. These are very similar values in spite of the higher ventilation in the work of Chamberlain. The main difference was the much higher temperatures in the work of Chamberlain and this was considered to be due to the lower heat losses at the much lower residence times in their work.

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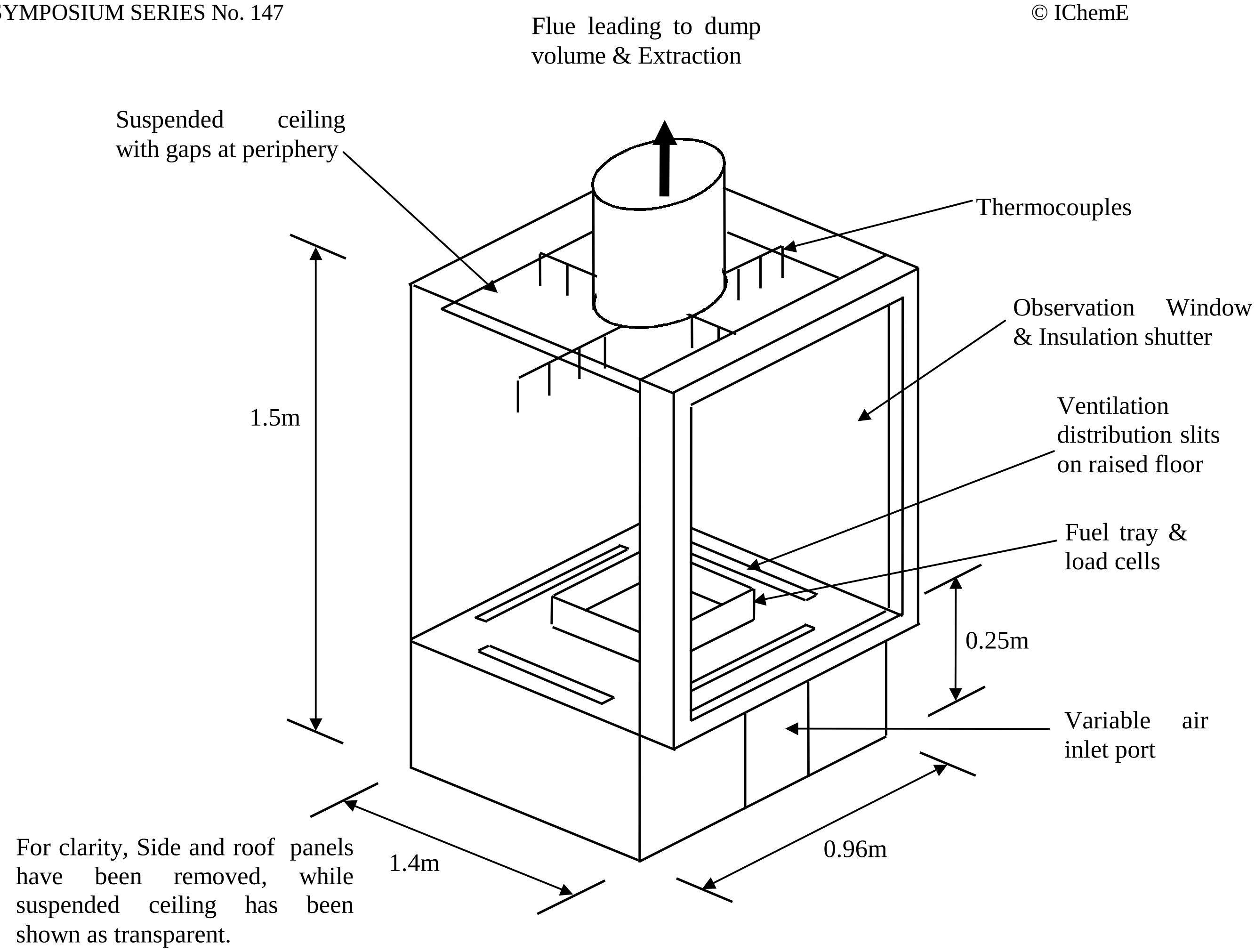

Fig.1. Schematic diagram of experimental investigation rig.

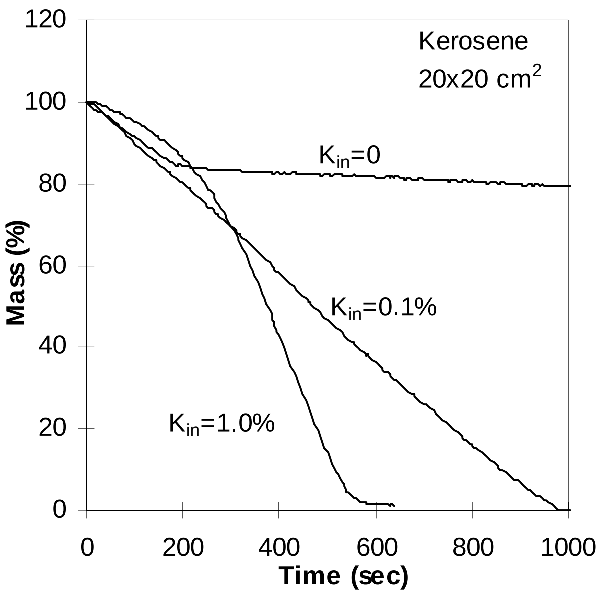

Fig. 2. Remaining fuel mass load (relative to the starting mass) as a function of time for the different air inlet area coefficients (K_{in})

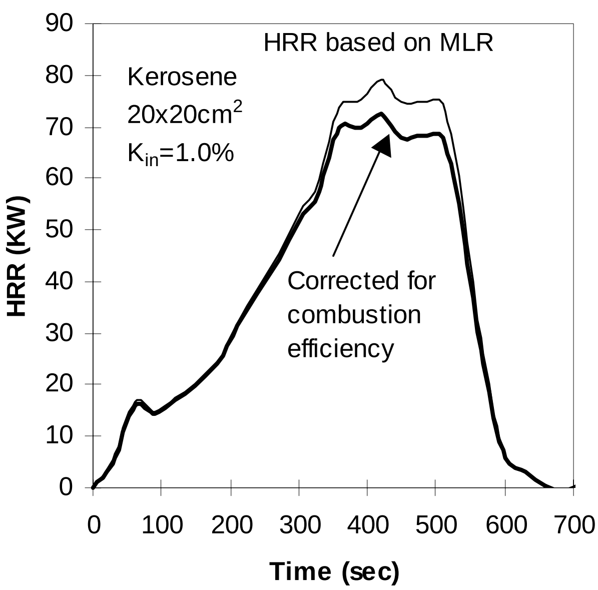

Fig.3. Heat release rate based on the mass-loss rate as a function of time, for the largest air inlet area.

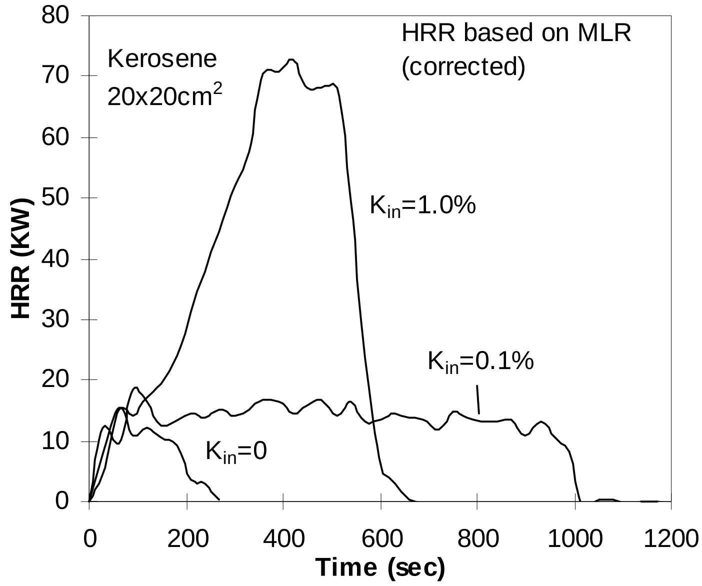

Fig.4. Heat release rate based on the mass-loss rate (corrected for combustion efficiency) as a function of time for the different air inlet area coefficients (K_{in})

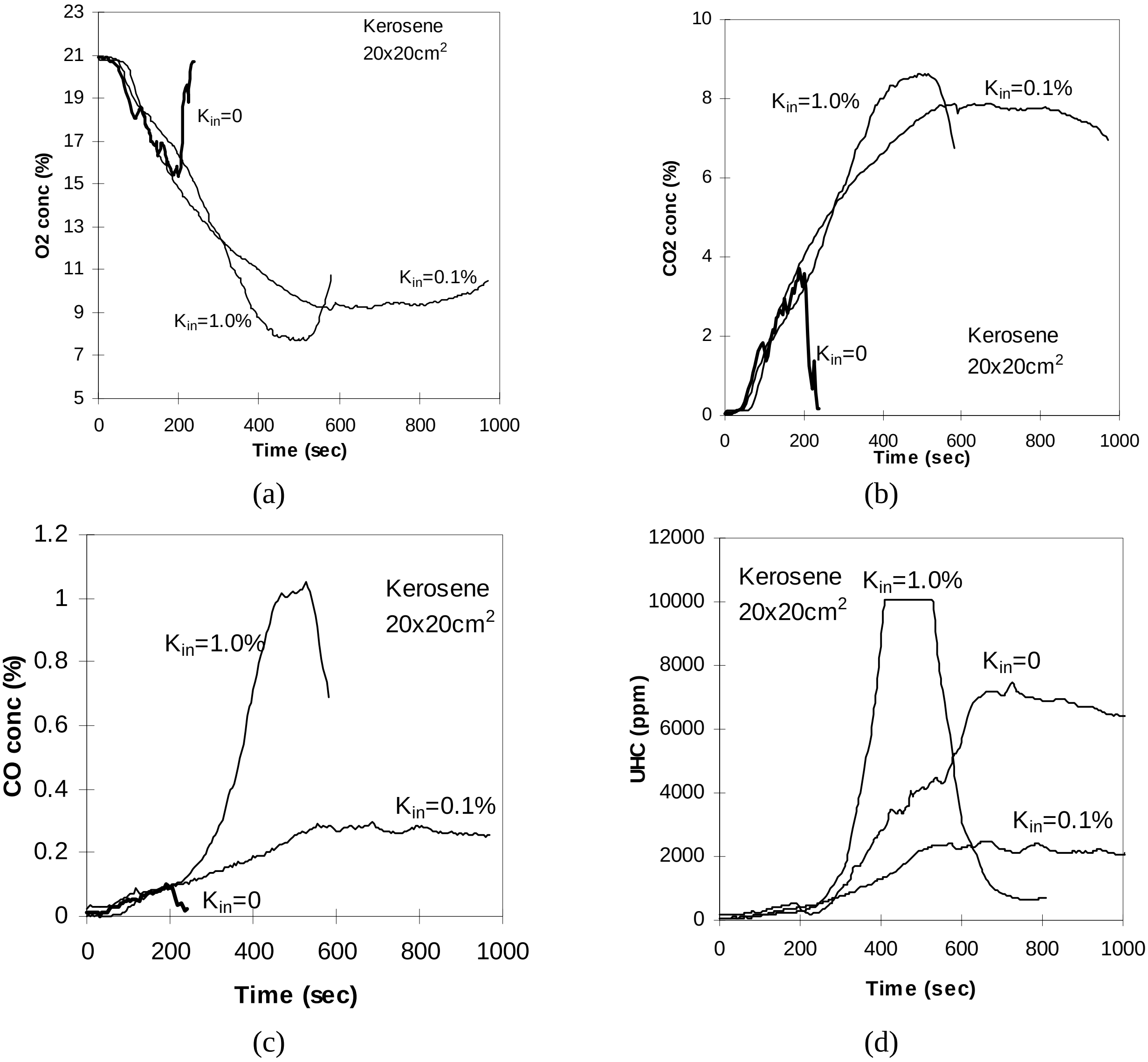

Fig. 5. Fire gas composition analysis as a function of time for the different air inlet area coefficients (K_{in}). (a) Oxygen, (b) carbon dioxide, (c) carbon monoxide, (d) unburnt hydrocarbons

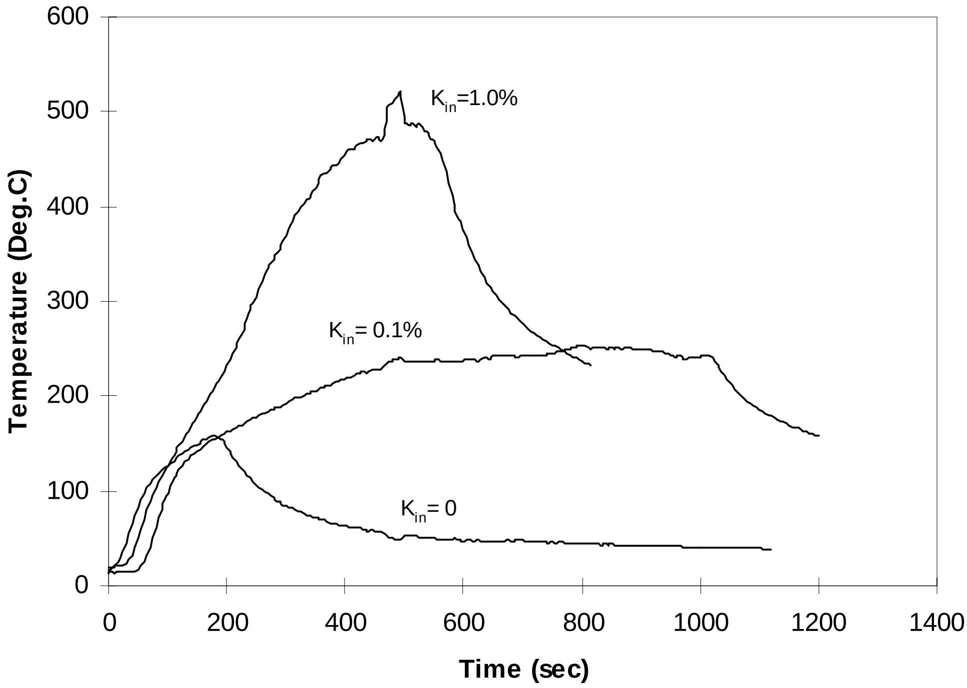

Fig.6. The mean central ceiling temperature 70mm from the ceiling, determined using an array of thermocouples, as a function of time for the three ventilation conditions.