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Experimental and Theoretical Study of Rack Storage Fires

Haukur Ingason

Lund 1996
Experimental and Theoretical Study of Rack Storage Fires

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Keywords: Rack storage fire, fire growth rate, in-rack fire plume, flame height, gas temperature, gas velocity, flame heat transfer, in-rack sprinkler, response time

Abstract: A theoretical and experimental study of rack storage fires and responsiveness of sprinklers is presented. Free-burn tests with non-combustible and combustible material were carried out in reduced scale with verification in large scale. Formulas for in-rack flame height, excess gas temperature, gas velocity and heat flux to storage walls are provided. The formulas include overall heat release rate, vertical flue width, height above the floor, height of virtual origin and sootiness of fuel. They can be used to predict activation times of in-rack sprinklers and it is possible to directly incorporate them into engineering models designed to predict fire growth in storage geometries.

The storage arrangement is important for the initial flame spread and fire growth rate. The reduced scale study shows that the initial fire growth rate decreases with increasing vertical gaps (flues) and that the vertical and lateral flame spread rate increase when the lateral flue height increases. The fire growth rate of rack storages is usually described by a power law dependence on time to the third power. The large scale test shows, however, that the initial fire growth rate is better described by an exponential function.

The present work provides measurements of the heat flux distribution at the surface of four square steel towers representing an idealisation of a rack storage at reduced scale. Three gaseous fuels, carbon monoxide (CO), propane (C₃H₈), and propylene (C₃H₆) were supplied from a circular gas burner at the floor. The fuels were chosen to cover a wide range of sooting tendencies leading to distinctly different flame heat fluxes. The differences are surprisingly large. For the same overall fire heat release rate the peak heat flux from C₃H₈ flames is twice that from CO flames, whereas the peak heat flux from C₃H₆ is 2.8 times greater than from CO flames. The heat fluxes were measured by thermocouples spot-welded onto the backside of the exposed steel tower sheets. The measuring technique was found to be simple, accurate and rugged in addition to being inexpensive.

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Front Page: Maria Andersen
Front page photo: Large scale fire tests with sprinklers carried out in 1989 at the Swedish National Testing and Research Institute (SP) in Borås. The tests were commissioned by the furniture manufacturer IKEA AB.

Layout: Haukur Ingason
Figures/Diagrams: Haukur Ingason

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Nomenclature

$A_1$ defined in equation (16).

$a, b, c$ defined by equation (18) and (19) (m/kW$^{2/5}$)

$bw$ thermal plume width (m)

$b_w$ non-dimensional width

$c_p$ specific heat (kJ/kg K)

$C$ conduction parameter (m$^{1/2}$/s$^{1/2}$)

$C_{b, T, C_u}$ nondimensional constants

$CHP$ Change of Phase Parameter

$D$ effective diameter (m)

$F$ function defined in reference 52

$Fr$ Froude number

$g$ acceleration of gravity (m/s$^2$)

$Gr$ Grashof number

$H$ height of rack storage (m)

$\Delta H_c$ heat of combustion per unit mass of the fuel (J/kg)

$L_f$ mean flame height (m)

$m$ air mass flow rate (kg/s)

$m_e$ total mass-entrainment rate of air below the mean flame tip (kg/s)

$m_f$ fuel mass flow rate (kg/s)

$Q$ chemical heat release rate (kW)

$Q_c$ convective heat release rate (kW)

$Q_{c, 0}$ centreline convective heat release rate (kW)

$Q_{c, \text{exp}}$ convective heat release rate based on measurements (kW)

$Re$ Reynolds number

$RTI$ Response Time Index (m$^{1/2}$s$^{1/2}$)

$r$ stoichiometric mass air-to-fuel ratio
\( t \) time (s)
\( t_0 \) time at t=0 (s)
\( t_{op} \) time of sprinkler activation (s)
\( T_a \) ambient conditions of sprinkler (K)
\( T_g \) gas temperature in wind tunnel (K)

\( T_0 \) centreline gas temperature (K)
\( \Delta T \) temperature rise above ambient (K)
\( \Delta T_{cop} \) temperature difference at sprinkler response (K)
\( \Delta T_g \) temperature difference (K)
\( \Delta T_0 \) centreline excess gas temperature (K)
\( \Delta T_{lf} \) centreline excess temperature at mean flame height (K)
\( u \) velocity (m/s)
\( u_0 \) centreline velocity (m/s)
\( w \) width of a vertical flue (m)
\( z \) height from floor level (m)
\( z_0 \) height of virtual origin (m)

**Greek symbols**

\( \text{entrainment coefficient} \)

\( \text{air-to-fuel stoichiometric fraction} \)
density at ambient temperature

(time constant (s))

convective fraction of
c
chemical heat release rate, \( \frac{Q_c}{Q} \)
nondimensional parameter

defined by equation (22)
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List of papers

This thesis is based on the following papers, published or submitted for publication in international symposiums or fire safety journals.

Paper I  
*Investigation of Thermal Response of Glass Bulb Sprinklers using Plunge and Ramp Tests*
Haukur Ingason
Submitted to Fire Safety Journal (1996)

Paper II  
*Numerical Simulation of the Wind Shadow Effect on the Convective Heat Transfer to Glass Bulb Sprinklers*
Haukur Ingason & Bror Persson

Paper III  
*Two Dimensional Rack Storage Fires*
Haukur Ingason

Paper IV  
*Modelling of a Two Dimensional Rack Storage Fires*
Haukur Ingason
Submitted to Fire Safety Journal (1996)

Paper V  
*CFD Simulation of Fires in Two Dimensional Rack Storage*
Heimo Tuovinen & Haukur Ingason

Paper VI  
*In-Rack Fire Plumes*
Haukur Ingason
Accepted for publication at the Fifth International Symposium of Fire Safety Science, Melbourne, Australia (1997)

Paper VII  
*Flame Heat Transfer in Storage Geometries*
Haukur Ingason & John De Ris
Submitted to Fire Safety Journal (1996)
1 Overview

The research work presented in this thesis has been performed during the time period of 1990-1996 in partial fulfilment of the PhD requirements at the Department of Fire Safety Engineering at Lund University. The first chapter includes a brief overview of the thesis work presented in the enclosed Papers I-VII. In chapter 2 the background and motivation for the research activities and a presentation of other related research activities is given. In chapter 3 a brief introduction to fire protection of rack storages is given and in chapters 4 and 5 an extension of the work presented in the enclosed papers is discussed. In chapter 6 a summary and conclusions is given and in chapter 7 proposals for future works is briefly discussed.

The thesis includes both experimental and theoretical studies of high rack storage fires and sprinkler responsiveness. Prior to the modelling work of the rack storage fire plume, a thorough analysis of the sensitivity of available response models for different heating conditions was conducted.

1.1 Sprinkler response

Papers I and II include an investigation of the responsiveness of glass bulb sprinklers and is based on the experimental and theoretical work given in reference 1. This work was initiated after discussions with Nils-Erik Gustafsson at Industrial Insurance in Finland. Questions had been raised by Gustafsson as to whether a two parameter model using the Response Time Index (RTI) and the Conduction (C) parameters was able to explain all the wind tunnel test data from an extensive test series reported in reference 7. To obtain better results, Gustafsson proposed a third parameter, an actuation parameter called the CHP parameter. This parameter was thoroughly investigated and the results are presented in Paper I. The study did not only consider the three parameter model (RTI, C and CHP) but also the wind shadow effects of the yoke arms on the response characteristic. The work on the wind shadow effects is presented in Paper II.

1.2 In-rack fire plumes

An investigation of the "in-rack fire plume" was carried out and the results are presented in Papers III - VII. The in-rack fire plume is defined here as the buoyant turbulent fire plume created in the vertical flues between the stored goods.
Three different types of configurations were investigated. The work started with a simple geometry and combustion process and progressed to more complex scenarios. Most of the work was carried out in reduced scale (1:3). Since large scale experiments are expensive, only one large scale test was performed for exploratory verification. A schematic figure of the three test configurations is shown in Figure 1. Since real rack storage can vary in size and storage arrangement the scale value of 1:3 should only be considered as approximate.

![Figure 1](image)

**Figure 1.** The three different rack storage configurations used in the experimental study.

The first experimental series included non-combustible two dimensional (2D) rack storage configuration in 1:3 scale where a propane line burner was placed at the bottom of the rack storage and where only horizontal flues had been included. The next test series with non-combustible material in reduced scale included a three dimensional (3D) rack storage configuration where only vertical flues were included. The working name of this test set-up was "the World Trade Centre" (WTC) tower tests because of the similarity in form to the famous building in New York. The WTC test series is described in Paper VII and the results are discussed in section 5.3. The boundary conditions in the WTC tests are akin to those given by the simple physical in-rack fire plume model outlined in Paper VI. An experimental series with free burn tests in a 3D rack storage configuration in two different scales, 1:3 and 1:1, is presented in Paper VI. The commodity used consisted of multiple-wall corrugated paper cartons equally separated in a steel rack with both vertical and horizontal flues. An extended analysis of the results is given in section 5.2.
2 Introduction

It is not an unusual situation that the fire brigade, even when in attendance within a few minutes, is prevented from entering a warehouse building due to rapid fire development in a high racked storage of goods. Consequently, the fire brigade is forced to fight the fire from outside with little hope of reaching the seat of the fire. In order to avoid this situation, high rack storages are often protected with in-rack sprinklers inside and/or on the face of the high rack storage. The efficiency depends on the packaging and storage arrangement as well as the flammability of the commodity. Other important factors are sprinkler orientation and responsiveness of the heat sensitive element and the type of water supply e.g. a dry pipe or wet pipe sprinkler system. Any possible delay of sprinkler activation is very important as this may be critical for control of the fire. A significant amount of flames passing a sprinkler before it activates, may easily ignite combustibles at higher levels, and thus reduce the possibility for the sprinkler water to penetrate to the fire seat. In order to calculate the activation time information on flow conditions close to the sprinkler head is necessary. Thus, it is important to investigate how variation in vertical and lateral flue sizes (gaps created between the stored goods) influence the complex flow field near the in-rack sprinkler.

2.1 Aim

The aim of the work presented here was to establish a simple in-rack fire plume model which could be used to predict the flow conditions and the flame height inside the vertical flues. The practical use is in the prediction of the response time of the first in-rack sprinkler. The prediction of sprinkler activation time in simple room configurations is done routinely using power law correlations of temperature and velocity in plumes and ceiling jets. Similar correlations for complex configurations like high rack storage are, however, not found in the open literature. Large scale experiments are usually expensive to perform and modelling work with Computational Fluid Dynamics (CFD) often extensive and time consuming. Thus, engineering power laws like those presented here are of practical importance. They may reduce the expense of testing and increase our understanding of the important parameters controlling the fire growth rate. These power law correlations can also facilitate future flame spread modelling and suppression studies in high rack storages. In time it is hoped that a fully developed rack storage model (engineering models and/or CFD models) may be used to predict fire growth rate, flame spread, activation times of in-rack and ceiling sprinklers and suppression efficiency of rack storage fires.
2.2 Related work

Large scale experiments\textsuperscript{12-18} have clearly shown that flame spread and fire growth rates are very rapid in high racked storage of goods. Modelling of flame spread and fire growth rate requires knowledge about flame heights and heat flux distribution to the walls of the stored material. The mechanism of flame spread is characterised by a pyrolysis of the exposed material, heat-up of yet unignited material ahead of the pyrolysis zone and combustion of pyrolysed gases which in turn generates heat fluxes to the wall material. Therefore, it is essential to address the heat flux distribution from the in-rack flames. In recent years, great effort has been focused on flame spread modelling over vertical surfaces of combustible materials\textsuperscript{19-22}. The physics and prediction of the fire growth rate is well understood for a single burning vertical wall, whilst the phenomena governing the flame spread in confined configurations like rack storages has not been thoroughly studied.

Despite the number of all large scale experiments performed throughout the world\textsuperscript{12-18}, it appears that very little information is available in the open literature on the in-rack temperatures, velocities and flame heights. This is probably due to the fact that the tests were mainly performed to study the behaviour of sprinkler systems in controlling rack storage fires, rather than to systematically study the in-rack plume flow and flame spread. Conclusions on the effects of different flue (gap) sizes on the flame spread are difficult to make. Recommendations in NFPA 231C\textsuperscript{23} (which are based on full scale experiments carried out at the Factory Mutual Research Test Center, West Clocester, Rhode Island\textsuperscript{23}) on the size of flues (nominal size according to NFPA 231C is 152.4 mm) are probably more related to the passage of water in the rack storage than effects of flame spread.

The Early Suppression Fast Response (ESFR) program conducted by Factory Mutual Research Corporation (FMRC) required numerous large scale tests with ceiling sprinklers as the only rack storage protection\textsuperscript{57}. Previous theoretical and experimental work\textsuperscript{13-15} on rack storage fires have thus been directed to predict what occurs above the rack storage and under the ceiling rather than what occurs inside it. An empirical fire growth rate correlation for 2,3,4 and 5 tier rack storage fires was obtained by Yu and Stavriandidis\textsuperscript{15} where the convective heat release rate is described as a power-law dependence on time to the third power. The fire growth rate was found to be invariant with the number of tiers. Empirical correlations for the maximum excess gas temperature, maximum gas velocity,
temperature profile depth, and velocity profile depth of the transient ceiling flow were established in term of a normalised convective heat release rate of the fire and a normalised radial distance from the fire axis. Variations or effects of different gap sizes was not studied. These correlations can be used to predict the response time of ceiling sprinklers.

Thomas\textsuperscript{12} studied some large scale fire test performed in the United Kingdom\textsuperscript{16, 17} and found no data on heat release rates or mass loss rates. Flue ignition resulted in the most rapid fire propagation and it was observed that the flame propagation was rapid enough to involve boxes on the upper levels before the box contents became involved\textsuperscript{16}. With flue ignition, an increase in box spacing from 75 mm to 150 mm, resulted in a decrease of the time taken for the flame to reach the top of the rack storage. However, for these two flue widths the ignition source was not identical (larger for the 150 mm flue)\textsuperscript{12,16}, which may have effected the initial fire spread. There were visual records available of the position of the rising flame and many video records, but the camera was not consistently positioned to allow the reliable determination of the flame height. Thermocouples were often installed in the gaps between packed goods but they were not useful for any systematic analysis\textsuperscript{12}. Based on his investigation of the UK tests, Thomas\textsuperscript{12} suggested that there is probably a gap size with a highest spread rate, that there is evidence that the width of the vertical gap affects average upward flame speed and that the average flame speed is of the order of 0.1 m/s.

Other related work is presented by Grant and Drysdale\textsuperscript{22} and by Foley and Drysdale\textsuperscript{25}. Grant and Drysdale\textsuperscript{22} modified the engineering approximation method given by Karlsson\textsuperscript{24} who combined a thermal theory of concurrent flame spread\textsuperscript{19} with empirical flame height correlations\textsuperscript{26} and test data obtained from the cone calorimeter\textsuperscript{27}. The numerical solution algorithm given by Grant and Drysdale\textsuperscript{22} is attractive since it permits heat release rate data from cone calorimeter tests to be used directly as input and it also allows burnout and the use of arbitrary flame height correlations. The work by Foley and Drysdale\textsuperscript{25} is important for studies of heat fluxes from the flames in a rack storage geometry. They showed that, for a given heat release rate (line burner at their base) the heat flux is increased when the separation of two parallel walls, is decreased. Blocking the ingress of air at the base of the walls is shown to increase the heat fluxes dramatic.

Recently, Hamer and Beyler\textsuperscript{11}, demonstrated the possibilities of using CFD for the calculation of the flow conditions within a rack storage in order to predict the
activation of in-rack sprinklers. The simulation by Hamer and Beyler\textsuperscript{11} was not compared to any experiments but it clearly shows the possibilities of using CFD for this type of complex configuration. Recently, a CFD submodel to predict flame spread over vertical fuel surfaces was developed by Yan and Holmstedt\textsuperscript{28}. The CFD code includes a simple and efficient pyrolysis model which can be used to predict the flame spread in a rack storage.
3 Rack storage fire protection

The overall fire hazard of stored goods is dependent on the combustibility of the material being stored, including packaging, and on the storage configuration. In warehouses the storage arrangement and type of material stored may vary considerably. The type of protection and determination of the design sprinkler water density and area of sprinkler operation are dependent on the combustibility of the product or mix of products and its packaging including the pallet and the arrangement and height of the storage. Possible storage arrangements include free standing storage, palletised rack storage, post-pallet storage or storage with solid or slatted shelves. Typical examples of storage configurations are shown in Figure 2.

![Figure 2. Typical examples of storage configurations](image)

The present study includes storages of post-pallet (ST3) type and/or palletised rack storage (ST4) type. Storage of high hazard material such as tyres, roll paper stored on end or flammable liquids, are not covered by this study. Rack storage is defined as any combination of vertical, horizontal, and diagonal members that supports stored materials. In general rack storage consists of either "double - and single-row racks" or "multiple-row racks". A common rack storage arrangement in Sweden is a double-row rack, i.e. two single-row racks placed back-to-back, with longitudinal and transverse flue spaces as shown in Figure 3.
Figure 3. Typical double-row (back-to-back) rack arrangement.

The rack storage is often provided with in-rack sprinkler heads located in the longitudinal flue spaces or on the storage face. According to NFPA 231C sprinklers in racks should conform to the ordinary temperature standard response classification with nominal 12.7 mm or 13.5 mm orifice size (dependent on storage height), and be pendent or upright. Sprinklers with 100 °C and 141 °C temperature ratings shall be used near heat sources as required by NFPA 13. The standard also permit the use of Quick response sprinklers in racks. In-rack sprinkler are positioned within the flue spaces based on storage height and commodity classification. Alternatively, the ESFR sprinkler system or the large drop sprinkler system (see NFPA 231C-chapters 9 and 10), which only use ceiling sprinklers, can be installed if the requirement on building height, storage type and commodity type is fulfilled.

Field gives a thorough description of series of large scale tests carried out in the United Kingdom to investigate the problem of protecting high-rack storage with sprinklers. Protection of the stores by a conventional in-rack system incorporating 68 °C glass bulb sprinkler heads was investigated and found to be unsatisfactory. Flames tended to pass a given sprinkler location before its operation and subsequently ignited combustibles at a higher level. Sprinkler heads which were capable of operating at an earlier stage in the fire development were therefore
tested in a new series of tests. Fast responding sprinkler heads were expected to control a fire whilst still in its early stages of development. This would require a sprinkler with a low thermal lag or time constant. Three different sprinkler were used: a glass bulb sprinkler rated at 68 °C (no information is given about the response sensitivity but from a published picture it appears to be an 8 mm diameter bulb), a solder-cup sprinkler rated at 74°C and classified as an ‘intermediate’ response sprinkler, and a solder-link sprinkler rated at 74 °C and classified as a ‘fast’ response. No definition of ‘intermediate’ or ‘fast’ response is given in the paper. The sprinklers were located at the junction of the transverse and longitudinal flues. In addition, the spacing between the boxes, i.e. dimensions of the transverse and longitudinal flues was also varied. The results can be summarised as follows 16:

- in order to operate ahead of an approaching flame, a sprinkler head must have an adequately high sensitivity,

- the glass bulb sprinklers located at all but the highest levels in the rack generally operated after the flame had passed (see Figure 4),

- the intermediate response solder-cup sprinkler used was significantly more sensitive than the glass bulb, frequently although not exclusively, operating before the flame had passed (see Figure 4),

- the fast response solder-link sprinkler consistently operated ahead of the flame (see Figure 4),

- fires ignited within the flue developed rapidly, frequently reaching the top of the rack within 2 minutes. The flame propagation was rapid enough to involve boxes on the upper levels before the box contents (wood wool and polystyrene chips in cardboard boxes) became involved; lateral flame spread at this stage was reported to be minimal,

- fires ignited on the face of boxes initially developed slowly, taking about 7 minutes to ignite boxes on the second level; this appeared to be a critical point in
the fires development. After this critical stage, flames tended to preferentially move towards the central flue and the fire subsequently behaves as if it was ignited in the flue,

- an increase in box spacing, i.e. the dimension of transverse and longitudinal flues, resulted in a decrease in the time taken for the flame to reach the top of the rack for flue ignition,

![Figure 4. Typical sprinkler response to a fire ignited in the vertical flue (based on information given in reference 16).](image)

The simple correlations provided in the present study, i.e. flame height, excess gas temperature, and velocity could be used to predict the response times given in Figure 4 (at least the first sprinkler). Unfortunately, no information about heat release rate as a function of time is available for these large scale tests. Thomas indicated that the fire development could be described by an exponential function of time with a plausible time constant of 10 - 15 seconds. Measured response times and flame heights based on information given by Field are shown in Figure 4.

The observation that the wider flues (150 mm instead of 75 mm) tend to decrease the time for the flames to reach the top is very interesting. The model scale tests presented in Paper VI show some contradictory results. A possible explanation is
that the change in spacing from 75 mm to 150 mm was, according to Thomas\textsuperscript{12}, accompanied by a more powerful ignition procedure. It is known from the model scale tests (Paper VI) that increasing the size of the ignition source will shorten the time for the flames (from ignition) to reach the top of the rack storage. It is, however, desirable to investigate this contradiction further, preferably by large scale tests.
4 Responsiveness of sprinklers

In order to predict the response of a sprinkler head mounted in the ceiling or in a rack storage, parameters which characterise the responsiveness of the sprinkler head are determined from wind tunnel tests. There are a number of different models available\(^2,4,6-8,29-31\). In this thesis these models employing the Response Time Index\(^4,5\) (RTI), the Conduction parameter\(^6,59\) (C) and the Change of Phase Parameter\(^2\) (CHP) were considered. The RTI parameter reflects the thermal time constant of the heat-sensitive element and the C parameter the heat conduction loss to the sprinkler fitting. The CHP parameter is assumed to describe the delay of the sprinkler response shortly before activation.

4.1 Response models

To determine two response parameters at least two different tests are required. To determine three parameters at least three tests are required etc. Thus, by combining a plunge test\(^4\) and a ramp test\(^32\) one would expect to cover most growing fire situation (fast and slow). The method is straightforward but does, however, require equipment which can maintain the boundary conditions assumed by the theoretical models applied to determine the response parameters. The sensitivity of the various thermal response models was investigated by calculating the response parameters from a set of wind tunnel tests followed by prediction of response times from tests performed under other heating conditions. The thermal response models were also evaluated under realistic fire conditions. The heat release rate at sprinkler response was used for comparisons between the thermal response models for different growing fires all of which followed a power law correlation expressed as the square of time. Further, a thermal analysis using a computer program was performed to explain some of the test results.

In Paper I, it is concluded that the experimental wind tunnel results can not be fully explained in the framework of the thermal response models applied. Consistently different C values were obtained for wind tunnel tests with different rates of temperature rise (ramp test). Further, tests with the precondition temperature of the sprinkler close to the operation temperature resulted in higher RTI values compared with cases with normal precondition temperatures. This is treated more thoroughly in Paper I but a brief discussion is given below.
4.1.1 Heat losses to sprinkler fittings

A great deal of work was done in order to find a reasonable explanation to why different C values were consistently obtained. As a result it was possible to show (Paper I and Reference 1) that this was probably due to an uneven heating of the sprinkler frame arms during the different ramp tests. The cooling technique used in the experiments was not efficient enough to cool the entire frame and thus maintain the boundary conditions required by the theoretical model. In retrospect, it is probably nearly impossible to obtain constant boundary conditions for different ramp tests since the sprinkler frame will always be heated somewhat during the heating process. The temperature will always be slightly higher at the contact point exposed to free stream, even when we have a theoretically constant temperature at the sprinkler mount. This was actually shown by the numerical calculations presented in Reference 1. The only way to maintain a constant boundary temperature at both contact points is to cool the entire sprinkler frame, which is not experimentally practicable.

4.1.2 Time delay in glass bulb sprinkles

One of the important conclusions in Paper I is that the higher \( RTI \) values obtained with a preconditioning temperature close to the operation temperature can be attributed to a time delay caused by temperature gradients within the glass bulb. This conclusion is based on a thermal analysis made using a numerical computer program showing a thermal time delay occurring in a plunge test situation. This time delay was found to be of the order of 5-6 seconds (insulated bulb) irrespective of precondition temperature. Thus, the effect of time delay become more important for shorter activation times. Theoretically the time delay implies that the elevated temperature within the glass bulb does not start at the time \( t=0 \) but at the time \( t=t_0 \), where \( t_0 \) is a virtual time origin as proposed by Job et al.\(^{33}\). The time delay appears to be physically justified in the beginning of the heating process and not at the end of the heating process as proposed by Gustafsson\(^{2}\). Thus, to be consistent with the measured activation times, these time delays should be equal. If this is true, it should be possible to determine the time delay from the results of the three parameter model \( (RTI+C+CHP) \). To show this we incorporate a thermal time delay into the original two parameter model \( (RTI+C) \). Integrating equation (5) in Paper I, from \( t=t_0 \) to \( t=t_{op} \) instead of \( t=0 \) to \( t=t_{op} \) where \( t_{op} \) is the measured response time the solution becomes:
\[
RTI = \frac{-(t_{op} - t_0) \left( 1 + \frac{C}{\sqrt{u}} \right) \sqrt{u}}{\ln \left[ 1 - \left( 1 + \frac{C}{\sqrt{u}} \right) \frac{\Delta T_{eop}}{\Delta T_g} \right]}
\]  

(1)

where \( \Delta T_g = T_g - T_a \) (°C), \( \Delta T_{eop} = T_{eop} - T_a \) (°C), \( u \) is the gas velocity (m/s), \( T_g \) is the gas temperature (°C), \( T_{eop} \) is the temperature of the heat-sensitive element at activation (°C) and \( T_a \) is the ambient temperature of the heat-sensitive element (°C). In Table C-6 in reference 1, we find \( RTI \) and \( C \) values calculated for the Ø5 mm sprinkler based on the three parameter model. If we, for example, can use parameters based on two plunge tests (\( T_a = 30 \) °C and \( T_a = 52 \) °C) and a ramp test with \( b = 2 \) °C/min, we obtain \( RTI = 80.05 \) m\(^{1/2}\) s\(^{1/2}\), \( C = 0.541 \) m\(^{1/2}\)/s\(^{1/2}\) and \( CHP = 6.91 \) °C. The measured response time, \( t_{op} \), for a plunge test with \( T_g = 197 \) °C, \( T_a = 30 \) °C and \( u = 2.5 \) m/s is \( t_{op} = 18.5 \) s. Now we can solve for \( t_0 \) in equation (1) using \( RTI = 80.05 \) m\(^{1/2}\) s\(^{1/2}\), \( C = 0.541 \) m\(^{1/2}\)/s\(^{1/2}\), \( t_{op} = 18.5 \) s and \( T_{eop} = 71.6 \) °C, yielding an expected time delay of \( t_0 = 3.15 \) s.

The time delay according to the Gustafsson model \(^2\) for the plunge test situation can be obtained by integrating equation (9) in Paper I:

\[
t_{CHP} = \frac{RTI \cdot CHP}{\sqrt{u}} \left( \Delta T_g - \left( 1 + \frac{C}{\sqrt{u}} \right) \Delta T_{eop} \right)^{-1}
\]

(2)

Using the values given above the time delay, \( t_{CHP} \), according to equation (2) is 3.14 s, which is nearly the same as the value obtained by equation (1), i.e. 3.15 s. Thus, equation (1) appears to be equally good as the three parameter model and it is physically more justified for glass bulb sprinklers exposed to heating conditions such as those in a plunge test.

The time delay effect of a glass bulb sprinkler tested in a plunge test is implicitly incorporated in the two parameter model. Since no time delay exists in a real fire situation, one could expect that the \( RTI \) value used for simulations is too high if not corrected for the time delay. These effects have not been investigated here but Heskestad and Bill \(^6\) have shown a good correlation between predicted response time and measured response times for different type of glass bulb sprinklers in a real fire situation without time delay corrections of the parameters. Hence, the implicit time delay effects in the \( RTI \) and \( C \) parameters are probably of minor practical importance.
4.1.3 Heat release rate at sprinkler response

The calculated heat release rate at sprinkler response in a growing fire condition was also used to evaluate the thermal response models. For the fast, medium and slow fire growth rates the heat release rates at sprinkler response were found to be similar, whether based on calculations with two parameters ($RTI + C$) or with all three parameters ($RTI + C + CHP$). However, when using only the $RTI$ value, the predicted rate of heat release at sprinkler response decreased significantly. Thus, including more than two parameters did not improve the results, at least not of any practical significance. Using the $RTI$ parameter alone appears justified solely in cases where $RTI$ is very large, and where the fire growth rate is fast and the ceiling height is relatively low. In other words $C$ become increasingly less important as $RTI$ increases and the fire growth rate increases. This can also been seen in the calculations given by Heskestad and Bill $^{6,59}$.

4.2 Responsiveness of in-rack sprinklers

The sprinkler deflector of a pendent in-rack sprinkler may delay the activation of the sprinkler by blocking the free stream around the glass bulb. A similar problem, i.e. wind shadow effects of the yoke arms, is discussed in Paper II. One would expect that similar correlations are valid for the in-rack flow situation. In Paper II it is shown that when the free flow around the heat sensitive element is blocked the response time is substantially increased. The RTI theory for calculating the time to activation of sprinklers is shown, however, to be valid irrespective of the orientation of the sprinkler. The calculations show that the influence of the yoke arms upon the convective heat transfer to the bulb is small when the yoke arms are oriented perpendicular to the flow whilst when they are aligned with the flow, the heat transfer is decreased significantly. An important finding is that the Nusselt number for this case is still roughly proportional to the square root of the Reynolds number. This finding is of a practical importance, as the lumped mass theory for calculating the time to operation of sprinklers relies heavily on the assumption of a square root dependence of the Reynolds number.

Usually in-rack sprinkler heads are aligned with the upward flue flow (pendent) and the deflector is headed into the plume flow. Actually, weak evidence for the conjecture that the results in Paper II may also be valid for this type of flow situation can be obtained from the work by Heskestad and Bill $^{6}$. They present measurements of a glass bulb sprinkler (N) showing similar $RTI$ values obtained
either with the yoke arms aligned parallel with the flow or when the deflector is headed into the flow. When the sprinkler yoke arms are aligned perpendicular to the flow the $RTI$ value is about two times less than in the two other situations. These results indicate that the glass bulb is blocked by the deflector which may explain the increase in the $RTI$ value.
5 In-rack fire plumes

A complementary analysis and discussion of the results obtained in Papers III to VII is given in this chapter. We are mainly interested in correlations for prediction of the flame height, the excess centreline gas temperature, the centreline gas velocity and heat fluxes to the fuel surface.

5.1 Two dimensional (2D) tests, non-combustible material

Due to the complexity of rack storage fires it was found appropriate to begin the investigation by using a simple experimental set-up, i.e. two dimensional rack storage consisting of non-combustible material in model scale, see Figure 1. It was believed that the horizontal flues might play an important role for the entrainment. The experimental set-up and analysis is presented in Paper III and in reference 37. The work continued with theoretical modelling of the 2D configuration. In Paper IV, a semi-empirical model is presented to calculate the average in-rack temperatures, velocities and flame heights. In order to obtain reasonable agreement with the measured data, it is necessary to introduce generalised empirical corrections of loss coefficients, stoichiometric requirements, and the distribution of heat release in to the model.

The physical entrainment mechanism in the 2D tests is quite different from that observed in the more realistic three dimensional (3D) rack storage configuration (Papers VI and VII). The 2D configuration created a constrained flow system and the entrainment is governed by the pressure difference between the inside and outside of the rack storage. The 3D system is a self-preserving flow system and the entrainment is more probably governed by turbulent shear forces at the edge of the in-rack fire plume rather than a pressure difference between the inside and outside of the rack storage.

The 2D tests are well suited for comparison with CFD codes. Such a comparison is made in Paper V using the JASMINE code 34-36. At the time of simulation there was no radiation model available in the JASMINE code. Heat losses from gases to the wall boundaries were calculated with a bulk heat transfer coefficient which takes into account both radiative and convective effects. Furthermore, due to the limitations of the physical boundary conditions in the JASMINE version used, the two highest tiers were inactive (no heat transfer). The calculated mass flow rates were found to agree reasonably well with the measured result, while the simulated
gas temperatures were generally overestimated. At lower parts of the plume regions, i.e. near the simulated fire source, the overestimation is largest and increased flue widths yield a larger overestimation. The simulated vertical gas velocities are also overestimated, as can be expected due to the high temperatures. The general difference is that calculated gas velocities are almost constant with height, rather than increasing with height as according to the experimental measurements. The simulation were run in transient mode until 3 minutes when steady state was assumed. In this type of configuration it is essential to predict the heat losses correctly, otherwise unrealistically high gas temperatures will be predicted inside the flues. New simulations using CFD codes which treat the radiative and convective losses to the flue walls more accurately are, therefore, necessary.

In general the following conclusions from the 2D study are made:

- the width of the vertical flue is the governing geometric parameter for the in-rack fire plume flow (Papers III+IV),

- variation of the horizontal flue heights from 50 mm to 100 mm has a negligible effect on the in-rack fire plume flow (Papers III+IV),

- a reasonably good agreement is obtained between the semi-empirical model and the experimental data for the parameters important for calculation of sprinkler response time, i.e. in-rack temperature and velocity at all four tiers. The dominating flow mechanisms are therefore accounted for in the model. The flame height prediction is, however, not as consistent with the experimental data (Paper IV),

- in CFD simulations of rack storage fires, it is necessary to use CFD codes which treat the radiative and convective losses to the flue walls more accurately than given in Paper V.

The total heat losses were measured in a number of pilot tests performed prior to the 2D test series. The test set-up was nearly the same as the one in Paper III except that the instrumentation was kept to a minimum and there were no wings in front of the storage. The storage length was 0.63 m instead of 0.59 m. The chemical and the convective heat release rates were measured above the rack storage using the conventional oxygen consumption calorimetry in the 2 MW range (the NORDTEST Furniture Calorimeter). The convective heat release rate is
a good measure of the steady state conditions during the test. Shortly after ignition and up to 3 minutes the ratio $Q_c/Q$ varies from 0.45 to 0.60 and after 4 minutes the ratio become nearly constant at 0.65 - 0.7. The variation is mainly due to the heat up of the exposed walls during the test period as higher wall temperatures imply less convective heat loss.

It was pointed out in Paper IV (p.14) that the vertical flow profiles (velocity and temperature) might vary both with the flue width and the distance from the burner, i.e. the flow profiles would not be similar in form at all widths and heights. It was thought that this behaviour might explain the poor correlation of the ratio, $Q_{c,exp}/Q$, which was expected to be 0.7. Here, $Q_{c,exp}$ is the convective heat release rate measured at each tier and $Q$ is the total heat output from the line burner. The mass flow rate and the excess temperature difference were calculated from the measured centreline temperatures and velocities assuming a top hat profile at all tiers and widths. For the 100 mm wide flue $Q_{c,exp}/Q$ was approximately 1.2 at the highest tier, which is of course physically impossible, whilst for the 50 mm wide flue, $Q_{c,exp}/Q$, was about 0.9 (see Figure 4 in Paper IV). In order to obtain a better picture of how the temperature and velocity profiles vary in the vertical flue, results from the CFD calculations presented in paper V, were analysed further. In Figures 5 and 6, the non-dimensional excess temperature (\( T_{ex} \)) and the non-dimensional velocity (\( V_{ex} \)) is plotted versus the non-dimensional distance (\( z \)) from the flue centreline for flue widths of 50 mm and 100 mm, respectively, and at the heights $z=0.802$ m (3rd tier) and 0.232 m (1st tier). The vertical flue was divided into 5 volume elements, representing half the flue width (symmetry). On the ordinates in Figures 5 and 6 the calculated values are normalised to the "centreline" value, which is calculated in the centre of the first volume element (5 mm thick for $w=50$ mm and 10 mm for $w=100$ mm) and therefore showing an offset of 0.1 on the abscissa. The last value is calculated in the centre of the last volume element (0.9), i.e. closest to the wall.
The non-dimensional excess temperature plotted as a function of the non-dimensional flue width. The heat output was 18.84 kW.

On the abscissa the distance $x$ from the flue centreline has been normalised relative to the half flue width ($w/2$). Calculation of the ratio of mean to maximum excess temperature was carried out with the information provided in Figure 5. The results indicate that the ratio increases considerably with increasing height and decreases slightly with increasing flue width. Similar results are obtained for the...
velocity profiles (Figure 6). The ratio of the mean to maximum convective heat release rate, $Q_c/Q_{c,0}$, where $Q_{c,0}$ is the centreline (maximum) value and $Q_c$ the integrated value, can be estimated from the following equation:

$$\frac{b_w}{x/w/2} u' = u/\bar{u}, \Delta T' = \Delta T/\Delta T_0 \text{ and } T' = T/T_0.$$  

This method gives an indication of the ratio of the mean to maximum convective heat release rate for different heights and widths. We integrate the curves in Figures 5 and 6 (numerically) at $z/L_f \approx 1$ and obtain that $Q_c/Q_{c,0} = 0.73$ for $w=50 \text{ mm}$ and $Q_c/Q_{c,0} = 0.48$ for $w=100 \text{ mm}$. Thus, we can correct the ratio $Q_{c,exp}/Q$ given in Paper IV to yield $Q_{c,exp}/Q = 0.9 \times 0.73 = 0.66$ for $w=50 \text{ mm}$ and $Q_{c,exp}/Q = 1.2 \times 0.48 = 0.58$ for $w=100 \text{ mm}$. These ratios are much more realistic than those originally obtained in Paper IV (0.9 and 1.2, respectively). Thus, the explanation given in Paper IV concerning the flow profile dependence on height and flue width appears to be rational. This exercise shows how CFD analysis can be used to clarify experimental results which at first sight appear to be inconsistent and difficult to explain.

5.2 Three dimensional (3D) tests, combustible material

In paper VI, a quasi-steady state fire plume law for in-rack centreline gas temperature and gas velocity, is derived from a simple physical model. This exploratory theoretical model suggests that the excess in-rack centreline gas temperature should be proportional to the instantaneous convective heat release rate, $Q_c^{2/3}$ and be inversely proportional to height, $z$. The in-rack gas velocity should be proportional to $Q_c^{1/3}$ and be independent of $z$. The flue width ($w$) dependence is also present in the correlations. The relations have the same functional form as a convective plume flow above a linear fire source. A plot of the measured temperature in the two different scales, appears to give a reasonable good correlation. The scatter is, however, substantial. In order to obtain similar correspondence for the velocity in the two scales it was necessary to use Froude number scaling with storage height as length scale. This does not comply with the results of a linear plume model which indicates constant velocity independent of height $z$. Hence, in order to investigate whether other sets of correlations would
improve the results, the experimental data is plotted using axisymmetric fire plume laws \(^{39,48-50}\), see section 5.2.3. Surprisingly, the axisymmetric correlations yield considerably less scatter for the large scale test performed. This indicates that the experimental data presented here obeys axisymmetric power laws better than the line plume power laws as suggested by the linear plume model in Paper VI. Plotting the available flame height data indicate the same tendency, see section 5.2.2 and Paper VII. For axisymmetric power law correlations, the flame height should be proportional to the chemical heat release rate, \(Q^{2/5}\), the excess centreline gas temperature should be proportional to the instantaneous convective heat release rate, \(Q_c^{2/3}\), and inversely proportional to \(z^{5/3}\). The centreline gas velocity should be proportional to \(Q_c^{1/3}\) and inversely proportional to \(z^{1/3}\).

There is no apparent physical explanation at hand why the axisymmetric power laws yields better correlation of the experimental data. The line plume model in Paper VI has some physical justification, in particular the geometric consideration of the entrainment rate per unit height. It is evident from the experiments presented in Papers VI and VII, however, that a better overall correspondence is obtained using axisymmetric power law correlations. In the simple line plume model the horizontal entrainment velocity at the edge of the fire plume was assumed to be proportional to the centreline vertical plume velocity. This is akin to the assumption given for free axisymmetric and line plume fires \(^{39,40,48-50}\). The largest difference between the axisymmetric plume and the in-rack fire plume is the physical blockage of air entrained into the plume per unit height. The stored boxes block a substantial part of the thermal plume perimeter compared to axisymmetric plumes. In order to obtain agreement between the theory and the experiments the right hand term in the continuity equation in Paper VI (equation (1)) needs to be modified. It is not clear at the moment how the right hand term could be modified regarding the blockage but simulations with CFD would be helpful in investigating this. It is clear, however, that the buoyant in-rack fire plume is quite different from the free buoyant turbulent plume flow in terms of geometry, turbulence, flow profiles, friction, pyrolysis of the fuel and heat losses. The simple assumptions given in Paper VI are probably not adequate to describe the dominating mechanism of air entrainment into the in-rack fire plume. Ignition at the centre of two parallel boxes, i.e. half way into the transverse flue, would perhaps yield results which correlate better with the linear power law model. To conclude the discussion the most important findings in Paper VI are summarised below:
- plot of excess gas temperature for both model scale and large scale tests appear to correlate with the theoretically obtained power law relation but there is a considerable scatter in the results,

- in order to obtain similar correspondence for the velocity in the two scales it was necessary to use Froude number scaling with storage height as length scale,

- the initial convective fire growth rate was found to be more accurately described by an exponential function than by a power-law dependence on time to the third power,

- the initial convective fire growth rate appears to decrease with increasing flue width,

- the horizontal flues clearly affect the average upward flame speed which was found to be in the range of 0.05 to 0.13 m/s.

- the horizontal flues affect the horizontal flame spread rate.

5.2.1 Modelling

Using model scale experiments is an effective way to obtain approximate answers concerning growth and spread of fires. The obvious advantages are low cost and ease of operation relative to large scale alternatives. However, the problem with such an approach is the lack of complete dynamic similarity between the large scale test and the model. The number of dimensionless groups that should be preserved is quite large as the forces relating to buoyancy, inertia and viscosity are all involved. Dimensionless groups such as the Froude number (Fr), the Reynolds number (Re) and the Grashof number (Gr) should be preserved from scale to scale. For geometrically similar enclosure fires, it is possible to maintain all these groups constant if the gas temperature is kept constant and the pressure, \( p \), is increased and the characteristic scale length, \( L \), reduced such that the product \( p^2L^3 \) is conserved from scale to scale. In the study presented here both the large scale test and the model scale tests were performed at atmospheric pressure, i.e. \( p^2L^3 \) was not preserved. If temperature and pressure are constant from scale to scale, constancy of both Froude number and Reynolds number, as required for strict modelling, cannot be satisfied simultaneously. Hence, Froude modelling is only possible for situations in which viscous forces are relatively unimportant.
Froude number modelling requires that velocities are scaled relative to the square root of the characteristic length, i.e. $u/L^{1/2}$ must be maintained constant. If the geometric similarity is conserved the heat release rate, $Q$, must be scaled relative to the $5/2$ power of the characteristic length $45-47$, i.e. $Q/L^{5/2}$ must be a constant from scale to scale.

Here, the geometrical similarity is maintained if the non-dimensional height, $H/w$, is constant from scale to scale, where $H$ is the total height of the rack storage and $w$ is the vertical flue width. The non-dimensional height, $H/w$, for $w=50$ mm (model scale), varies from 28.4 to 32.4 ($H=1.42$ m for $h=50$ mm, $H=1.52$ m for $h=75$ mm and $H=1.62$ m for $h=100$ mm), for $w=75$ mm $H/w$ is equal to 18.9, for $w=100$ mm $H/w$ is equal to 14.2. $H/w$ is equal to 36.8 for the large scale test ($H=5.52$ m, $w=50$ mm, $h=300$ mm). Thus, the requirement of geometrical similarity is not fully satisfied here and the results should be regarded as approximate. However, it is of interest to compare the results between these two scales. Thus, a comparison was made between the measured values of the chemical heat release rate, $Q$, the excess centreline gas temperature, $\Delta T_0$, and the centreline gas velocity, $u$, for model scale tests with $w=50$ mm and the values for the large scale test at the highest measuring point, i.e. $1/3$ of the box height below the top of the rack storage at the time when the flame height reached the top of the rack storage ($H=H_f$). In Table 1, calculated values according to the scaling laws discussed are given. This means that the heat release rate, $Q$, should be preserved on the linear scale, $H$, to the $5/2$ power, the excess temperature, $\Delta T_0$, should be the same in both scales and the velocity should be preserved on the linear scale, $H$, to the $1/2$ power.

| $w=50$ mm, $h=50$ mm | 33.2 | 29.8 | 1.98 | 1.97 | 1.20 |
| $w=50$ mm, $h=75$ mm | 26.7 | 25.1 | 1.61 | 1.90 | 0.98 |
| $w=50$ mm, $h=100$ mm | 33.5 | 21.4 | 1.68 | 1.84 | 1.03 |

The overall correspondence between the model scale and the large scale tests is reasonably good with one exception, i.e. where $\frac{Q_F}{Q_M}=33.5$ and $\left(\frac{H_F}{H_M}\right)^{5/2}=21.4$. 

Table 1 Froude number scaling of the measured values at the time when $L_f=H$. The heat release rate at $L_f=H$ for the large scale test was 1253 kW. The index F refer to large scale tests and index M to the model scale tests.
The reason for this discrepancy is not known but the results obtained in the model scale tests should give reasonable answers concerning the fire behaviour in large scale tests. Heskestad \(^{46}\) discuss the need that the thermal properties of the walls must be modelled for proper response. This was not considered here but since the excess temperature appears to scale reasonable well this is probably not important here.

In the following section the in-rack flame height, centreline excess gas temperature and centreline gas velocity is plotted using the concept of a virtual origin and assuming that the data can be plotted according to the axisymmetric power law correlations.

### 5.2.2 In-rack flame height

In Figure 7, the flame height data is plotted for the experiments with paper cartons in two different scales. A description of the experiments is given in Paper VI. The flame height data was obtained from video recordings by comparing the ‘average’ fluctuation of the flame tip to a ruler painted on the boxes. This method, although subjective, was found to be reasonably accurate (+/- 50 mm by estimation) as the fluctuations of the flames were substantially less than those observed in free axisymmetric plumes. A more objective method would be to define the flame height as a mean temperature but that would require either significantly more instrumentation or more comprehensive analysis. For the purpose of the present study the method used here is, however, considered satisfactory. A least square fit to the flame height data (model and large scale tests) shown in Figure 7 yields the following equation:

\[
L_f = -3.73w + 0.343Q^{2/5} \tag{3}
\]

The linear correlation coefficient, \(R\), is equal to 0.993. Equation (3) is valid for \(L_f \leq H\). The flame height for axisymmetric fire plumes according to Heskestad \(^8\) is also plotted in Figure 7 for comparison:

\[
L_f = -1.02D + 0.23Q^{2/5} \tag{4}
\]

where \(D\) is the effective diameter of the fire source in meters. In order to compare Heskestad's flame height correlation with the in-rack flame height correlation we replace \(D\) with \(w\) in equation (4) and put \(w\) equal to 0.15 m. The in-rack flame height diverges from the axisymmetric flame height as \(Q\) increases. The increase
is small for low $Q$, and increases asymptotically to $0.343/0.23 = 1.49$, i.e. nearly one and a half times the axisymmetric flame height for high $Q$. The effective diameter of the fire source, $D$, is assumed to be equal to $w$ in all cases. If $D$ increases as $Q$ increases the flame height will become lower and hence, the ratio $L_{f,r}/L_{f,a}$ may easily exceed 1.5. The index $r$ relates to rack storage and index $a$ to axisymmetric.

To demonstrate the effects of the presence of rack storage on the flame height we can use equations (3) and (4). A 500 kW fire within a rack storage with a flue width $w$ equal to 0.15 m yield a flame height of 3.56 m, whilst from a free burning fire source with a diameter of 0.15 m the flame height will be 2.61 m. This corresponds to an increase in height by 36%. If we double $Q$ corresponding flame heights will be 4.88 m and 3.49 m, respectively, i.e. 40 % increase. The reason for the higher flame heights in the rack storage is that less air is entrained into the fire plume per unit height. The visible flame volume contains the combustion zone and if less air is entrained the flame tip is ‘raised’ up to a height where enough air is entrained to complete the combustion of the fuel pyrolysed at lower levels.

Figure 7. The non-dimensional flame height ($L_f/w$) plotted as a function of $Q^{2/5}/w$ for the model scale tests and the large scale test. The flame height for axisymmetric fire plumes according to Heskestad is plotted for comparison assuming $w=D=0.15$ m.
The flame height data for the model scale experiments with $w=50$ mm, $h=75$ mm and $w=50$ mm, $h=100$ mm were excluded in Figure 7 and equation (3) because they clearly diverged from the other flame height data. The excluded data is plotted instead in Figure 8 where it can be observed that as the height $h$ is increased the flame height tends to increase for corresponding $Q$ and $w$. One possible explanation for this divergence is that the entrainment rate is reduced when $h$ increases. A plot of the experimental flow data show, however, that $h$ does not have any noticeable effect on the in-rack centreline temperature and velocity, see section 5.2.3. This seems inconsistent with what one would expect, i.e. if the entrainment rate is reduced the cooling of the plume should also change. In general, there appears to be some inconsistency in the data when $h$ is increased. For example Table 1 in Paper VI shows that the flame spread much faster when the horizontal flue is at $h=100$ mm compared to $h=50$ mm while Figure 3 in Paper VI shows that the fire with $h=100$ mm develops slower than the fires with $h=50$ mm and $h=75$ mm. A further investigation of these results is necessary. An interesting observation in Figure 8 is that the slope of the curve becomes identical to the slope of equation (3) when $Q^{2/5}/w > 55$.

![Figure 8](image-url)  

*Figure 8. The non-dimensional flame height plotted as a function of $Q^{2/5}/w$ for tests diverging from equation (3).*

The non-dimensional flame height, $L/w$, was plotted as a function of $Q^{2/5}/w$ for the steel tower tests presented in Paper VII. The horizontal flues were excluded in the
steel tower tests. The air entrained was therefore only through the vertical flues. The fuel type and flue width was varied. Correlation of the flame height data yields the following equation (Paper VII):

\begin{equation}
\text{(5)}
\end{equation}

This correlation shows a remarkably good correspondence with equation (3). This indicates that the horizontal flues do not have any significant influence on the flow dynamics within the vertical flues. Further, the type of fuel supply appears not to have any significant effect on the in-rack flame height. The linear correlation coefficient, R, for equation (5) was 0.986, which is slightly less than the one obtained for equation (3), R=0.993. The flame heights used to derive equation (5) were obtained visually during the experiments and not from video recordings.

One way to demonstrate that the simple line plume model developed in Paper VI underestimates the air entrainment for the 3D experimental set-up used here is to plot the non-dimensional flame height data according to a flame height correlation obtained from this simple line plume model. The total entrainment at the flame tip, \(m_e\), can be obtained by integrating the local entrainment rate \(dm/dz\) from the virtual origin, \(z_0\), up to the flame tip, \(z=L_f\). Thus, by differentiating equation (9) in Paper VI and integrating we get:

\begin{equation}
\int_{z_0}^{L_f} \frac{dm}{dz} dz = \left[ \left(4 \alpha \rho_\infty \right)^{2/3} \left( \frac{g}{c_p T_\infty} \right)^{1/3} \left( w^2 Q_c \right)^{1/3} \right] dz \quad \text{(6)}
\end{equation}

We make the same fundamental assumptions as Heskestad \(^41\) for free burning buoyancy-controlled turbulent flames. The chemical kinetics is assumed to be instantaneous and the in-rack flame extends to a height where \(m_e\) is just sufficient to complete the combustion reactions. The air requirement from the surroundings is proportional to the stoichiometric requirement of the pyrolysis gases, i.e.:

\[ m \propto m_f r \quad \text{(7)} \]
where $m_f$ is the fuel flow rate and $r$ is the stochiometric mass air-to-fuel ratio. This relationship defines the air-to-fuel stoichiometric fraction $^{42,43}$:

$$\phi = \frac{m}{m_f r}$$  \hspace{1cm} (8)

where the fuel flow rate is determined using:

$$m_f = \frac{Q}{\Delta H_c}$$  \hspace{1cm} (9)

and where $Q$ is the chemical heat release rate and $\Delta H_c$ is the heat of combustion per unit mass of the fuel. The ratio of the convective energy is defined as $\chi = Q/Q$. With aid of equations (6) - (9) and assume $\phi = \phi_e$ at $m = m_e$ we obtain the following relationship for the in-rack flame height:

$$L_f = z_0 + \phi_e r \frac{\Delta H_c}{\Delta H_c} \left( \frac{c_p T_m}{4 \alpha \rho_s} \right)^{1/3} \chi g \left( \frac{Q}{w} \right)^{2/3}$$  \hspace{1cm} (10)

These results suggest that we should plot $L_f$ as a function of $(Q/w)^{2/3}$. To be consistent with the other plots we plot the non-dimensional flame height $(L/w)$ as a function of $(Q/w^{5/2})^{2/3}$. 

![Graph showing flame height as a function of heat release rate per unit mass squared and convective energy ratio]
Figure 9. Non-dimensional flame height plotted as a function of \((Q/w^{5/2})^{2/3}\).

If we compare Figure 9 to Figure 7 we see clearly that the experimental flame height data correlates better with \(Q^{2/3}\) than \(Q^{2/3}\). If the experimental data would correlate with \(Q^{2/3}\) the data should fall onto a straight line. As the flame height is controlled mainly by the amount of air entrained, this indicates that the simple line plume model overestimates the flame height and consequently underestimates the air entrainment.

5.2.3 In-rack temperature and velocity

Turbulent buoyant axisymmetric fire plumes with a large density defect or temperature rise relative to the surrounding are known as strong plumes, while plumes with a small density defect or temperature rise are known as weak plumes. Numerous studies can be found in the literature on weak plumes and strong plumes. Weak plume theories have been extended to accommodate for large density deficiencies (strong plumes) and, to account for area sources a virtual origin, \(z_0\), is introduced. The virtual origin is most conveniently determined from temperature data above the flames along the plume axis. This method is, however, quite sensitive and the task is difficult in practice. Slight inaccuracies in the determination of centreline temperatures have large effects on the value obtained for the virtual origin.

Above axisymmetric buoyant turbulent diffusion flames, the plume radius and centreline values of excess temperature and velocity obey the following relationships:

\[
b_{\Delta T} = C_b \left(\frac{T_0}{T_\infty}\right)^{1/2}(z-z_0) \tag{11}
\]

\[
\frac{\Delta T_0}{T_\infty} = C_T \left[\frac{1}{g c_p \rho_\infty T_\infty^2} \right]^{1/3} \frac{Q_c^{2/3}}{(z-z_0)^{5/3}} \tag{12}
\]

\[
u_0 = C_u \left[\frac{g}{c_p \rho_\infty T_\infty} \right]^{1/3} \frac{Q_c^{1/3}}{(z-z_0)^{1/3}} \tag{13}
\]
where $b_{\Delta T}$ is the plume radius to the point where the temperature rise has declined to $1/2 \Delta T_0$ and $T_0$ is the centreline temperature. These relations are known as strong plume relationships $^{8,39}$. The values of the nondimensional constants $C_b$, $C_T$ and $C_u$ are according to Heskestad $^8$: 0.12, 9.1 and 3.4, respectively.

Heskestad $^{52}$ has presented a theoretical method to determine the virtual origin of fire plumes. The method is based on the assumption that the distance of the virtual origin below the level of the mean flame height will scale proportionally with the local width of the flow field at the elevation of the mean flame height, or expressed mathematically:

$$z_0 = L_f - Eb_{L_f}$$  \hspace{1cm} (14)

where $z_0$ is the elevation of the virtual origin above the top of the combustible, $L_f$ is the mean flame height, $b_{L_f}$ is the radius of the fire plume at the mean flame height, specifically the radius where the mean velocity is equal to $1/2$ the centreline value and $E$ is a nondimensional constant. Combining the equations for the mean flame height $^{53}$ and plume width, one can obtain for $z_0$:

$$z_0 = -1.02D + FQ^{2/5}$$  \hspace{1cm} (15)

where $F$ includes variables of environment at ambient conditions, fuel and nondimensional constants. Heskestad $^{52}$ plotted virtual origins from a number of investigations using a rearranged form of equation (15), i.e. $\frac{z_0}{D}$ versus $\frac{Q^{2/5}}{D}$. Equation (15) was found to be well represented for common fuels and normal atmospheric conditions with the coefficient $F$, i.e. the slope of the curve, equal to $0.083 \text{ m/kW}^{2/5}$ and $\frac{z_0}{D}$ intercept at -1.02. Data points deviating from this representation appeared not to be correlatable with the fuel type$^{52}$. Equation (15) is valid for porous fuel arrays, provided any in-depth combustion is not substantial, i.e., provided most of the volatiles released undergo combustion above the array of the crib $^{52,56}$.

Another method to determine, $z_0$, is to apply the measured centreline temperature data. You and Kung $^{13}$ used a technique based on equation (12) for strong fire plumes above a burning rack storage and Kung and Stavrianidis $^{51}$ used it for
strong fire plumes above pool fires. Equation (12) can be rearranged into a form as follows:

\[ z = A_1^{3/5} \left[ \frac{\Delta T_0}{Q_{c}^{2/3}} \right]^{-3/5} + z_0 \]  

(16)

where \( A_1 = C_T \left[ \frac{1}{g c_p \rho \frac{T_{\infty}^2}{T_w^2}} \right]^{1/3} \) and \( \Delta \tilde{T}_0 = \Delta T_0 / T_\infty \). Thus, the plume height, \( z \), varies with the variable, \( \left[ \frac{\Delta T_0}{Q_{c}^{2/3}} \right]^{-3/5} \), linearly with a slope of \( A_1^{3/5} \).

You and Kung \(^{13}\) did as follows: For each test, the centreline temperatures at three selected levels were used to determine the slope, \( A_1^{3/5} \), before flames persistently touched the lowest selected level (4 m above the rack storage). For each test, an averaged value of \( A_1^{3/5} \) was obtained from the determined values of \( A_1^{3/5} \) at about five time instants (a growing fire). The average value of \( A_1^{3/5} \) for all tests was obtained \((A_1^{3/5} = 0.256 \text{ m/kW}^{2/5} \) and \( C_T = 11)\) and used in equation (16) to finally determine the virtual origin. The virtual origin was then plotted against \( Q_{c}^{2/5} \) and a best representation of the data was found to be:

\[ z_0 = -2.4 + 0.095Q_{c}^{2/5} \]  

(17)

for three, four and five-tier storage heights. Slightly different representation was given for two-tier storage height.

To derive, \( z_0 \), for the experiments presented in this study, a combination of these two methods was applied. This was done by rearranging equation (12) and apply the excess centreline temperature, \( \Delta T_{L_f} \), i.e. the centreline flame temperature at the measured mean flame height, \( L_f \). The flame height data for the combustible rack storage experiments (model and large scale) was found to be best represented by equation (3), which can be written in more general form as:

\[ L_f = a + bQ^{2/5} \]  

(18)

where \( a = -3.73w \text{ m} \) and \( b = 0.343 \text{ m/kW}^{2/5} \). If we assume that the mean flame height, \( L_f \), has a corresponding excess gas temperature we can with aid of equation (12) express the flame height using \( z = L_f \) and \( \Delta T_0(L_f) = \Delta T_{L_f} \) as:
\[ L_f = z_0 + cQ^{2/5} \]  \hspace{1cm} (19)

where \( c = \left( \frac{A_1 T_\infty}{\Delta T_{r_f}} \right)^{3/5} \chi^{2/5} \), \( \chi = \frac{Q_c}{Q} \) and \( A_1 = C_T \left[ \frac{1}{g c_{p}^{2} \rho_{\infty}^{2} T_{\infty}^{2}} \right]^{1/3} \). The ratio \( \chi \) is usually in the range of 0.6 - 0.7 for free burning flames, whilst it is in the range of 0.4 - 0.7 for rack storage fires. It is possible to combine equations (18) and (19) such that:

\[ z_0 = a + (b - c)Q^{2/5} \]  \hspace{1cm} (20)

Thus, \((b - c)\) equals the coefficient \( F \) given by Heskestad 52. The coefficients \( b \) and \( c \) can be obtained from the experiments and thus it should be possible to obtain an expression for \( z_0 \), which is consistent with the gas temperature and the flame height data, and with an intercept, \( a \). We have already determined that \( b = 0.343 \) m/kW\(^{2/5}\), and \( c \) will be determined from the excess centreline temperature data. But first we need to confirm that this method yields reasonable results. Heskestad 52 predicted theoretically that \( c / \chi^{2/5} = 0.15 \) m/kW\(^{2/5}\) for strong fire plumes by using variables of common fuels and environment at ambient conditions. This is in accordance with data from Kung and Stavrianidis 51 which showed that \( c / \chi^{2/5} \) is in the range of 0.10 - 0.16 m/kW\(^{2/5}\) for strong fire plumes. Using variables of fuels and environment at atmospheric conditions and, \( C_T = 9.1 \) as in equation (12)

we can, with aid of equation (19), calculate . Thus,

\[ g = 9.81 \text{ m/s}^2 \]

\[ \rho_{\infty} = 1.2 \text{ kg/m}^3 \]

\[ c_{p} = 1.01 \text{ kJ/kg K} \]

\[ T_{\infty} = 293 \text{ K} \]

\[ T_{r_f} = 500 \text{ K (ref. 8,52),} \]

\[ A_1 = 0.085 \]

\[ Q_c = 0.7, \]

\[ Q = 0.165 \text{ m/kW}^{2/5} \]

obtain \( A_1 = 0.085 \) and consequently \( Q_c = 0.7 \), which should be compared to 0.15 m/kW\(^{2/5}\). We also find that \( c = 0.143 \) m/kW\(^{2/5}\). If \( b \) = 0.23
m/kW^{2/5}$, see equation (4), then $b-c=0.087$ m/kW^{2/5} which should be compared to $F=0.083$ m/kW^{2/5} in equation (15). Similar exercise as shown above has been made by Heskstad $^{56}$.

The constants $A_1$ and $b-c$ were determined in the following way for each experiments carried out in this study. An average value of was determined through a curve fit (method of least squares) of a plot of versus $z/L_f$. The value of was then determined at $z/L_f=1$. The value of for each test was then determined with aid of equation (16). The values of were calculated for the time period when the mean flame reached the base of the second tier ($Q_c=48$ kW) until the flames reached up to the top of the rack storage ($Q_c=738$ kW). The results are shown in Table 2. The values are average values during the time period considered.
Table 2  Determination of (b-c) in equation (20).

<table>
<thead>
<tr>
<th></th>
<th>(m/kW^{2/5})</th>
<th>(K)</th>
<th>(m/kW^{2/5})</th>
<th>b-c</th>
</tr>
</thead>
<tbody>
<tr>
<td>model w=50 mm,</td>
<td>0.405</td>
<td>469</td>
<td>0.533</td>
<td>0.234</td>
</tr>
<tr>
<td>h=50 mm</td>
<td></td>
<td></td>
<td></td>
<td>0.109</td>
</tr>
<tr>
<td>model w=75 mm,</td>
<td>0.683</td>
<td>593</td>
<td>0.576</td>
<td>0.354</td>
</tr>
<tr>
<td>h=50 mm</td>
<td></td>
<td></td>
<td></td>
<td>-0.011</td>
</tr>
<tr>
<td>model w=100 mm,</td>
<td>0.189</td>
<td>210</td>
<td>0.595</td>
<td>0.186</td>
</tr>
<tr>
<td>h=50 mm</td>
<td></td>
<td></td>
<td></td>
<td>0.157</td>
</tr>
<tr>
<td>large scale w=150</td>
<td>0.453</td>
<td>518</td>
<td>0.500</td>
<td>0.240</td>
</tr>
<tr>
<td>mm, h=300 mm</td>
<td></td>
<td></td>
<td></td>
<td>0.104</td>
</tr>
</tbody>
</table>

\( T_\infty = 287 \text{ K for model scale tests and } T_\infty = 284 \text{ K for large scale test.} \)

There is a great variation in the results. The variation could possible be explained by asymmetry of the flames. The tests were performed in SP’s large test hall (18 m by 20 m and 15 m high (average)) under a hood system without any arrangement to avoid draft near the experimental test set-up. This could have affected the temperature and velocity measurements but probably not the observed flame heights. Apparently, there is a great need for further research and large scale testing in order to establish more reliable values of \((b-c)\). Meantime, we use a value which appears to give a reasonably good representation of the model scale and the large scale tests. The value obtained by Heskestad \(^{52}\), i.e. \( F=(b-c)=0.083 \text{ m/kW}^{2/5} \) for axisymmetric turbulent fire plumes could be used since it is reasonably close to many of the values obtained in Table 2.

Now we will plot the temperature data and velocity data using the power law correlations given by equations (12) and (13) and with aid of equation (20) where \( a=-3.73w \) and \( (b-c)=0.083 \text{ m/kW}^{2/5} \). In Figure 10, a plot of the excess centreline temperature, \( \Delta T_0 \), versus the ratio \( (z-z_0)/Q_c^{2/5} \) is shown. It was found necessary to adjust \( z_0 \) for the test with \( w=100 \text{ mm and } h=50 \text{ mm} \), in order to obtain a correspondence with the other data. The data for this test is plotted with \( b-c=0.083 \text{ m/kW}^{2/5} \) and \( a=-0.8 \text{ m} \). Overall, there is reasonable correspondence, except in the
model scale tests at high values of \((z-z_0)/Q_c^{2/5}\) where the gas temperature diverge from the large scale data. These low temperatures are related to data at high elevations with low heat release rates. There is still considerable scatter in the model scale data but the data for the large scale tests fall nicely into a single curve. This indicate that the large scale test was a well behaved experiment yielding reasonably good results. To obtain a better representation of the data in Figure 10 it is plotted such that every 3rd data point is shown for the model scale tests and every 4th data point for the large scale test. For comparison the centreline temperature for the large scale test is plotted with every 2nd data point in Figure 11.

\[
\Delta T_0 = 28 \left[ \frac{T_\infty}{g c_p^2 \rho_\infty^2} \right]^{1/3} \frac{Q_c^{2/3}}{(z-z_0)^{5/3}}
\]

(21)

For \((z-z_0)/Q_c^{2/5} >0.20\), the excess centreline temperature is inversely proportional to \((z-z_0)/Q_c^{2/5})^{5/3}\). The scatter is considerably less for the large scale test compared to when plotted according to the simple linear plume correlation presented in Paper VI. A curve fit of the large scale test using the least square method for \((z-z_0)/Q_c^{2/5} >0.20\) yields the following relation of the gas temperature (R=0.973):
where $c_p = 1.01 \text{ kJ/kg K}$, $T_\infty = 284 \text{ K}$, $\rho_\infty = 1.238 \text{ kg/m}^3$ and $g = 9.81 \text{ m/s}^2$ were used to determine the non-dimensional constant, $C_T = 28$. Thus, the coefficient $C_T$ is increased by nearly a factor 3 compared to axisymmetric free burning fires.

\[ C_T = 28 \]

At $(z-z_0)/Q_c^{2/5} \leq 0.20$ the average excess temperature is equal to 836 K (std=46 K).

If we replace $z$ with $L_f$ and put $\Delta T_0 = 518 \text{ K}$ in equation (21) and use the values of $c_p, T_\infty, \rho_\infty$ and $g$ given above and $\chi = 0.50$ we obtain that $C$ in equation (19) is equal to 0.24 m/kW$^{2/5}$, which is the same value as in Table 2 for the large scale test.

A parameter, $\xi$, relating plume the centreline temperature with the centreline velocity was postulated by Heskestad \(^{41}\) to be a constant and confirmed by a number of fire tests:

\[ \xi = \frac{T_\infty^{2/5} (c_p \rho_\infty)^{1/5}}{g^{2/5} (\Delta T_0 Q_c)^{1/5}} \frac{u}{(\Delta T_0 Q_c)^{1/5}} \]  

\[ (22) \]
The parameter \( \frac{T_m^{2/5} (c_p \rho_m)^{1/5}}{g^{2/5}} \) is considered to be a constant here as we only consider the large scale test in order to determine, \( \xi \). A plot of \( \frac{u}{(\Delta T_0 Q_c)^{1/5}} \) as a function of \( Q_c \) for \( 48 < Q_c < 738 \) kW \( (L_f < H) \) yield a reasonably constant value of \( \xi = 2.71 \) (std=0.37). Here we used the ambient conditions given earlier for the parameter \( \frac{T_m^{2/5} (c_p \rho_m)^{1/5}}{g^{2/5}} \). The value of \( \xi \) obtained here should be compared to \( \xi = 2.628 \) given by You and Kung, \( \xi = 2.4 \) given by Kung and Stavrianidis, and \( \xi = 2.2 \) given by Heskestad.

Now we can use the Froude scaling laws discussed earlier to plot the velocity data. In Figure 12, the normalised centreline velocity is plotted against \( Q_c^{1/3} / (z - z_0)^{1/3} / H^{1/2} \). In Figure 12, every 3rd data point is plotted for model scale tests and every 4th for the large scale test. The scatter for the model scale tests is still considerable, but the large scale test appears to be better correlated compared to Paper VI, specifically at the lower levels.

![Figure 12](image.png)

*Figure 12.* The normalised velocity as a function of \( Q_c^{1/3} / (z - z_0)^{1/3} / H^{1/2} \) for the model scale tests and the large scale test.
Fitting a curve (least squares) to the large scale test for \( Q_c^{1/3} / (z - z_0)^{1/3} < 3.4 \) gives the following equation:

\[
 u = 3.54 \left[ \frac{g}{c_p \rho \infty T \infty} \right]^{1/3} \left( \frac{Q_c}{(z - z_0)} \right)^{0.45}
\]  

where the same ambient conditions as earlier were used. It is interesting to observe that the coefficient \( C_u = 3.54 \) is nearly the same as for axisymmetric plumes \( C_u = 3.4 \) although the power dependence has been changed from 1/3 to 0.45. This is of course based on just one test but it gives an indication of the type of power law might be expected for in-rack gas velocity.

The results presented in section 5.2 are very encouraging but it must be emphasised that there is a great need for further large scale testing in order to establish more credible general predictions for this type of fires.

### 5.3 Three dimensional (3D) tests, non-combustible material

The work presented in Paper VII provides measurements of the heat flux distribution at the surface of four equally separated rectangular towers exposed to flames from a circular gas burner at the floor. Three gaseous fuels carbon monoxide (CO), propane \( (C_3H_8) \), and propylene \( (C_3H_6) \) were used. The fuels were chosen to cover a wide range of sooting tendencies leading to distinctly different flame heat fluxes. The differences were surprisingly large. For the same overall fire heat release rate the peak heat flux from \( C_3H_8 \) flames was twice that from CO flames, whereas the peak heat flux from \( C_3H_6 \) was 2.8 times greater than from CO flames. The heat fluxes to the tower walls were measured by thermocouples spot-welded onto the backside of the exposed steel tower sheets. The measuring technique was found to be simple, accurate and rugged in addition to being inexpensive. The maximum error was estimated to be less than 5%.

The important questions for the study were how: (a) the flame dimensions and (b) wall heat fluxes depend on the: (1) fire heat release rate, (2) fuel type and (3) wall separation distance. The study addresses these questions by providing formulas for the flame heat flux distribution in terms of the overall fire heat release rate, fuel sootiness and separation distance. The results can be directly incorporated into engineering models designed to predict fire growth in storage geometries. The paper also provides additional data needed for the development of more general
CFD models capable of predicting fire growth of other geometries. In this regard correlations of the velocities and temperatures of the flame gases in the central flue, the gas temperature distributions in the horizontal direction, the width of the thermal plume and measurements of flame radiation are provided. In the following a comparison of some of these measurements to the large scale test is made. This comparison is shown in Figures 13 - 15.

![Graph showing non-dimensionless flame height versus \( Q^{2/5}/w \) in model scale (WTC) and large scale.]

Figure 13. The non-dimensionless flame height versus \( Q^{2/5}/w \) in model scale (WTC) and large scale.
Figure 14. The centreline excess gas temperature versus \((z-z_0)/Q_c^{2/5}\) in model scale (WTC) and large scale.

Figure 15. The normalized velocity versus \(Q_c^{1/3} / (z-z_0)^{1/3} / H^{1/2}\) in model scale (WTC) and large scale.
The height of the virtual origin, $z_0$, for the WTC tests is assumed to be equal to zero. The results are surprisingly similar, despite the fact the model scale tests were carried out with non-combustible material and without presence of horizontal flues. The variation in centreline temperature and velocity (Figure 14 and 15) depending on the flue width $w$ is surprisingly small in the WTC experiments. This does not comply with the fact that the flame height was found to be dependent on the flue width $w$. A more thorough analysis of the WTC tests needs to be done regarding this observation.
6 Summary and Concluding Remarks

This thesis includes both experimental and theoretical studies of high rack storage fires and sprinkler responsiveness. Prior to the modelling work of the rack storage fire plume, a thorough analysis was carried out in order to investigate the sensitivity of available response models to different heating conditions.

Important finding from Paper I is that the prediction of sprinkler response in realistic fire scenarios is generally well represented by the two parameter model, i.e. using the $RTI$ and $C$ parameter. Including more than two parameters does not improve the results, at least not to any practical significance.

In Paper II, a theoretical calculation of the convective heat transfer to a glass bulb sprinkler is carried out. The results show that for the type of sprinkler investigated, the heat transfer coefficient was more than twice as large for the best orientation (yoke arms orientated perpendicular to the flow) as for the worst orientation (yoke arms aligned with the bulb and the flow). The result will be a substantial increase of the time to operation of the sprinkler. It is also shown that when the free flow around the heat sensitive element is blocked by the yoke arm, the $RTI$ theory for calculating the time to activation of sprinklers, is valid irrespective of the orientation of the sprinkler. This finding is of a practical importance as the $RTI$ theory relies heavily on the assumption of a square root dependence of the Reynolds number. This finding will not change any practical engineering procedures in use today, but it verifies the validity of the traditional $RTI$ lumped mass heat transfer for these flow conditions.

Free-burn tests were carried out in reduced scale with tentative verification in large scale (Papers III-VII). In-rack flame heights, centreline excess temperatures and velocities were plotted using quasi-steady power law correlations. The correlations include chemical and convective heat release rate, vertical flue width, height above the floor and height of virtual fire source. It appears that using ordinary axisymmetric power law correlations to plot the 3D experimental data (ignition at centre of fuel array) yields a better representation of the mechanisms governing the in-rack plume flow than using linear power law correlations.

This thesis does not profess to give any final answers about the flame propagation and fire growth rate in large scale high rack storage fires; but, as the results appear to correlate reasonably well from model scale to large scale it is possible to give some tentative conclusions. The most practical and important result is the
presentation of simple engineering power law correlations for in-rack plume flow and in-rack flame height. As a consequence, it should be possible to calculate the activation time of the first in-rack sprinkler in a similar 3D rack storage configuration as presented here. The required input parameters are the fire growth rate, the vertical flue width and the height from floor. Further, as the heat flux distribution to the storage walls is given as a function of the flame height and fuel type, it should be possible to apply these correlations as input to other flame spread models. Thus, it should be possible to predict flame spread in rack storage fires quite accurately within the near future. The results presented here are very encouraging but in order to establish more credible predictions there is still a great need for additional research, both in reduced scale and in large scale.

The heat flux to the tower walls at a given location of the non-dimensional flame height ($z/L_f$) was not significantly affected within the flue width range tested (63 - 150 mm). Thus, for the same non-dimensional flame height and fuel type approximately the same heat flux to the wall is obtained. However, since the flame height ($L_f$) is dependent on the flue width ($w$), the heat flux at a given location will vary slightly with the flue width. This might explain why the initial fire growth rate is affected by the flue width. The model scale study shows that the initial fire growth rate and vertical flame spread decreases when vertical gaps (flues) increases in size. This result is in contradictory to observations from large scale tests performed in the UK. Unfortunately, in these large scale tests the increase in gap size was accompanied by a more powerful ignition procedure. The model scale study also indicates the importance of the horizontal flues on the flame spread, although they do not appear to affect the in-rack plume flow to any appreciable extent. For a given vertical gap size the flame spread both in the vertical and lateral gaps was increased when the lateral flue height was increased. In the case with varying horizontal flue heights, there appear to be some inconsistency in the in-rack flow data and the flame spread data so any general conclusions are difficult to draw. However, based on the model scale results a tentative recommendation of storage arrangement would be to keep the vertical gaps wide and the horizontal gaps narrow. The fire growth rate of rack storages is usually described by a power law dependence on time to the third power. The large scale test shows, however, that the initial fire growth rate is more accurately described by an exponential function.

The work presented in this thesis provide a sound basis for future studies of flame spread and fire growth modelling of high rack storages. Other fields of application include the prediction of activation times of in-rack sprinklers and
recommendations of critical distances between the stored goods. Further, the results may increase our understanding of parameters governing fire growth rates in different types of storage configurations. The thesis shows that it is possible to reproduce flow conditions and flame heights in different scales which is of practical importance since all large scale testing is very expensive. Another field where this work is useful is in future suppression studies of rack storage fires. The in-rack fire plume flow could be coupled to evaporation models of the droplets travelling down through the vertical flues and the heat flux measuring technique presented in Paper VII could be used to measure the cooling effects on the storage walls. Computation Fluid Dynamics (CFD) has not been widely used for the simulation of high rack storages. The work here provides extensive experimental data for comparison with CFD and possible semi-empirical models developed at some future date for rack storage configurations.
7 Future Work

There is a great need for further large scale testing in order to investigate the validity and limitations of the simple correlations presented here (e.g. high and low rack storages). In the 3D study the fire was ignited at the centre of the flue where the air is able to entrain from four sides (symmetric). In a single-row rack storage there is usually a certain blockage of the entrained air due to the neighbouring stored boxes (i.e. perpendicular to the direction of loading). It is important to investigate how this might affect the results. Such a study could presumably be made in an intermediate scale (1:2) or small scale (1:3) with verification in large scale. Additional investigation on the effects of other ignition locations than used here is desirable. In any future test series in this area it is crucial to have control (by measurements) of the symmetry of the flames.

There are many solid and liquid fuels, particularly aromatic fuels, which are much more sooty than propylene; so one can anticipate even greater heat fluxes for aromatic fuels despite their lower completeness of combustion. It is important to investigate if the trend of increasing heat fluxes to the storage walls with increasing sootiness of the fuel continues for fuels with $S \geq 5.6$ (Paper VII).

Some important questions for future flame spread studies in high rack storages are: i) how fast does the pyrolysis front (area) move in comparison to the flame height? ii) is the production rate of pyrolysis gases at the lower tiers sufficient to extend the flames all the way up prior to ignition of packaging material at the higher tiers? iii) how significant role does the horizontal flues play in rate of growth of the fire?, and iv) what controls the flame spread in the horizontal flues? These questions and many others should be addressed in future flame spread studies. The material presented in this thesis does not provide these answers but it will hopefully be able to provide a sound basis for their determination.

Well defined response tests with in-rack sprinklers are desirable. Special attention should be given to the wind shadow effects created by the sprinkler deflector. This can be done by comparing measured and calculated response times in rack storage tests using response parameters determined in a wind tunnel tests with the sprinkler deflector directed against the flow stream.
8 Acknowledgements

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The project was sponsored by the Swedish Fire Research Board (BRANDFORSK) which is gratefully acknowledged.

There is one person who I would like to dedicate this work to, namely my former section manager Mr Anders Ryderman, who died in 1995. He, more than anyone else, made it possible for me to start this work and his enthusiasm and encouragement were crucial to me during this time period.

Finally, my special thanks to my wife Gunny who supported and encouraged me all the time and to our 4 1/2 year old daughter Linda, who really tried to understand why I was not always home. Linda I can tell you now, I’m coming home.
9 References


3. Private communication with N. E. Gustafsson, Industrial Mutual Insurance Company, Finland.


Paper I

Investigation of Thermal Response of Glass Bulb Sprinklers using
Plunge and Ramp Tests

Haukur Ingason

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Investigation of Thermal Response of Glass Bulb Sprinklers using Plunge and Ramp Tests

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Abstract

The thermal behaviour of glass bulb sprinklers was investigated in a heated wind tunnel under various test conditions. The response parameters considered were the Response Time Index (RTI), the Conduction parameter (C) and the Change of Phase Parameter (CHP). Combinations of one, two and three response parameters were used to predict the response times of various wind tunnel test conditions, and for slow, medium and fast growth rate fires. These calculations indicate that the wind tunnel results cannot be fully explained in the framework of the thermal response models applied. Tests with a high precondition temperature of the sprinkler resulted in higher RTI values compared with cases with normal precondition temperatures. A plausible explanation for the higher RTI values could be a time delay caused by temperature gradients within the glass bulb. This indicates that the CHP parameter previously explained in terms of work needed to shatter the glass bulb, more likely reflects an affect of thermal time delay.

Calculated heat release rates at sprinkler response under a growing fire condition were used to evaluate the thermal response models. For the fast, medium and slow fire growth rates the heat release rates at sprinkler response were found to be similar, whether based on calculations with two parameters (RTI and C) or with all three parameters (RTI, C, CHP). When using the RTI value only, however, the predicted rate of heat release at sprinkler response decreased significantly.

Notation

A = surface area of the heat-sensitive element (m²)
c_p = specific heat of heat-sensitive element (J/kg K)
C = Conduction parameter (m/s)^{1/2}
C' = conductance (W/K)
CHP = Change of Phase Parameter (°C)
g = acceleration due to gravity (m/s²)
D = characteristic length; diameter of a cylinder or a glass bulb (m)
h = convective heat transfer coefficient (W/m² K)
H = ceiling height (m)
k_f = thermal conductivity of gas at film temperature; T_f = (T_g + T_b)/2 (W/m K)
k = thermal conductivity (W/m K)
m = mass of the heat-sensitive element (kg)
q'' = heat transfer to the heat-sensitive element (W/m²)
Q = Heat Release Rate (kW)
\( r = \text{radius (m)} \)
\( \text{RTI} = (mc/hA)u^{1/2} = \tau u^{1/2} = \text{Response Time Index (m}^{1/2}\text{s}^{1/2}) \)
\( t = \text{time (s)} \)
\( t_{\text{top}} = \text{response time (s)} \)
\( T_b = \text{surface temperature of heat-sensitive element (K)} \)
\( T_e = \text{average temperature of heat-sensitive element (K)} \)
\( T_{\text{eop}} = \text{average temperature of heat-sensitive element at sprinkler response (K)} \)
\( T_f = \text{the sprinkler fitting temperature (K)} \)
\( T_g = \text{gas temperature (K)} \)
\( T_w = \text{wall temperature (K)} \)
\( T_0 = \text{ambient temperature (K)} \)
\( T_1 = \text{liquid temperature at the interface of the liquid and glass of the bulb (K)} \)
\( T_2 = \text{liquid temperature at centre of the bulb (K)} \)
\( \Delta T_g = T_g - T_0 = \text{gas temperature difference (K)} \)
\( \Delta T_e = T_e - T_0 = \text{element temperature difference (K)} \)
\( \Delta T_{\text{eop}} = T_{\text{eop}} - T_0 = \text{temperature difference at sprinkler response (K)} \)
\( u = \text{gas velocity (m/s)} \)
\( z = \text{axial coordinate (m)} \)
\( \alpha = \text{fire growth rate coefficient (kW/s}^2) \)
\( \beta = \text{rate of temperature rise (°C/s)} \)
\( \beta T = \text{thermal expansion coefficient (K}^{-1}) \)
\( d = \text{characteristic length (m)} \)
\( \varepsilon = \text{emissivity} \)
\( \nu_f = \text{kinematic viscosity of gas at film temperature (m}^2\text{/s)} \)
\( \nu = \text{kinematic viscosity (m}^2\text{/s)} \)
\( \mu = \text{dynamic viscosity (kg/m s)} \)
\( \rho = \text{density of the material (kg/m}^3) \)
\( \sigma = \text{Stefan-Boltzman constant, } 5.67 \times 10^{-8} \text{ (W/m}^2\text{K}^4) \)
\( t = \text{time constant (s)} \)

**Introduction**

The potential of sprinklers to control or extinguish a fire depends to a large extent on the rapidity with which the sprinklers respond. For characterising the thermal responsiveness of automatic sprinklers it is necessary to know the response parameters of the sprinkler. The most frequently used response parameters are the Response Time Index (RTI) \(^{1,21}\) and the Conduction parameter (C) \(^{2,22}\). The RTI parameter reflects the thermal time constant of the heat-sensitive element and the C parameter the heat conduction loss to the sprinkler fitting.

It has been shown that the RTI and C gives a very good prediction of sprinkler response when tested in realistic fire scenarios \(^{2,22}\). Questions have, however, been raised by Gustafsson \(^{3,4}\) as to whether the two parameter model (RTI and C) is able to explain all the wind tunnel test data from an extensive test series reported in reference 5. The test series included twelve different sprinklers of six different designs and two different temperature ratings. Plunge tests \(^1\) with different precondition temperatures were used in the test series together with ramp \(^6\) tests. Gustafsson showed that the test results could not be explained using a
constant RTI parameter, nor was the addition of the C parameter able to fully explain the results. Thus, to obtain better results, Gustafsson 3 proposed an actuation parameter called the CHP parameter. The CHP parameter is assumed to describe the delay of the sprinkler response shortly before activation. CHP is the ratio $E/mcp$ defined by Evans and Madrzykowski 7 with $m$ and $c_p$ being the mass and specific heat of the heat-sensitive element, respectively, and $E$ being the heat absorbed by the link at the activation temperature. For solder type sprinklers the absorbed energy is assumed to arise from the heat of fusion. It is difficult to know exactly what physical processes are taking place in a glass bulb immediately prior to rupture. Gustafsson 4 proposes that a certain amount of mechanical energy must be absorbed by the bulb, and that a certain amount of work has to be done, before the bulb shatters. 

In the present study, a systematic investigation of these three parameters was carried out. The RTI, C and CHP parameters were determined from test data from ramp tests and plunge tests reported in reference 8. Computer programs written by Gustafsson 3 were used for the determination of the parameters. Three different types of glass bulb sprinklers were used: fast response (Ø 3 mm), medium response (Ø 5 mm) and slow response (Ø 12 mm). They were tested in a plunge test (constant gas temperature) with three preheating temperatures and in a ramp test (linearly increasing gas temperature) with two rates of temperature rise.

The sensitivity of the various thermal response models was investigated by calculating the response parameters from a set of wind tunnel tests followed by prediction of response times from tests performed under other heating conditions. The thermal response models were also evaluated under realistic fire conditions. The heat release rate at sprinkler response was used for comparisons between the thermal response models for different growing fires all of which followed a power law correlation expressed as the square of time.

Further, a thermal analysis using a computer program was performed to explain some of the test results. In this study no attempt was made to explain the test results using the approaches described in references 9-11.

**General theory of response models**

In the following, a brief discussion of the basic differential equations for one-, two- and three-parameter models 1-3 is given.

**One-parameter model**

The basic heat balance equation for the one-parameter model 1 is given by

$$m c_p \frac{d(T_e)}{dt} = h A (T_g - T_e)$$  \hspace{1cm} (1)

where $T_g$ is the gas temperature (°C), $T_e$ the temperature of the heat-sensitive element (°C), $m$ the mass of the heat-sensitive element(kg), $c_p$ specific heat of heat-sensitive element (J/kg °C), $h$ is the convective heat transfer coefficient (W/m² °C) and $A$ is the surface area of the heat-sensitive element (m²). Equation (1) assumes a lumped heat capacity system with no heat losses to the sprinkler fitting. Assuming that the convective heat transfer coefficient correlates to the
square root of the free air flow velocity \(^1\), and expressing temperatures relative to the initial (ambient) temperature, \(T_o\), equation (1) becomes:

\[
\frac{d(\Delta T_e)}{dt} = \frac{\sqrt{u}}{RTI}(\Delta T_e - \Delta T_c)
\]

(2)

where \(\Delta T_g = T_g - T_o\) (°C), \(\Delta T_e = T_e - T_o\) (°C), \(u\) is the gas velocity (m/s), \(RTI = (mc/hA)u^{1/2} = \tau u^{1/2}\) = Response Time Index as defined by Heskestad and Smith \(^{21}\) (m\(^{1/2}\)s\(^{1/2}\)) and \(\tau\) is time constant (s). \(RTI\) is the only unknown parameter and can be determined through a single plunge test. For the ramp test situation, the following substitution is made in equation (2)

\[
\Delta T_g = \beta t
\]

(3)

where \(\beta\) is the rate of temperature rise (°C/s).

**Two-parameter model**

The basic heat balance equation for the two-parameter model \(^{2,22}\) is given by

\[
m c_p \frac{d(T_e)}{dt} = h A (T_g - T_e) - C'(T_e - T_o)
\]

(4)

where \(C'\) is the conductance (W/°C). Equation (4) assumes, as equation (1), a uniform temperature distribution within the heat-sensitive element. Additionally, heat loss to the sprinkler fitting is assumed. At the sprinkler fitting, \(T_o\) is assumed constant. Expressing temperatures relative to \(T_o\), and rearranging, equation (4) becomes:

\[
\frac{d(\Delta T_e)}{dt} = \frac{\sqrt{u}}{RTI} [\Delta T_g - (1 + C) \frac{\Delta T_e}{\sqrt{u}}]
\]

(5)

where \(C\) is defined according to Heskestad and Bill \(^{2,22}\) as

\[
C = C' RTI/mc_p \quad \text{(m}^{1/2}\text{s}^{1/2})
\]

(6)

For the ramp test situation, \(\Delta T_g\) is substituted by \(\beta t\) in equation (5). Both \(RTI\) and \(C\) are unknown parameters. At least two plunge tests, or a plunge test and a ramp test, are required to determine these two parameters according to the method used in this study \(^3\), although other methods are available to determine the \(C\) parameter \(^{2,12}\). If the temperature rise of the sprinkler fitting is known, \(C'(T_e - T_0)\) in equation (4) can be replaced by \(C'(T_e - T_0) - C'(T_f - T_0)\) where \(T_f\) is the sprinkler fitting temperature \(^2\).

**Three parameter model**
The temperature increase in the heat-sensitive element is described here by two differential equations at different time intervals. According to the three-parameter model defined by Gustafsson 3, equation (5) is valid until the phase transition or work to shatter the glass bulb is initiated at time $t_1$. From time $t_1$ to the operation time, $t_{op}$, the rate of convective energy transfer ($\Delta E$) to the sensing element is given by equation (7),

$$\frac{d(\Delta E)}{dt} = h A (T_g - T_{cop}) - C'(T_{cop} - T_o)$$ \hspace{1cm} (7)

By dividing equation (7) by $mc_p$ and using the relationships $RTI = (mc/hA)u^{1/2}$ and $C = C' RTI/mc_p$ and expressing temperatures relative to $T_o$, the following equation can be derived:

$$\frac{1}{mc_p} \frac{d(\Delta E)}{dt} = \sqrt{\frac{u}{RTI}} [\Delta T_g - (1 + \frac{C}{\sqrt{u}})\Delta T_{cop}]$$ \hspace{1cm} (8)

Equation (8) can be solved by integrating from $t_1$ to $t_{op}$. Gustafsson 3 has defined the $CHP$ parameter to be equal to the parameter presented by Evans and Madrzykowski 7 where they considered $\Delta E/mc_p$ to be "a parameter for the linkage". The equation for $CHP$, which has the dimension ($^\circ$C), will be

$$CHP = \frac{\Delta E}{mc_p} = \frac{\sqrt{u}}{RTI} \left[ t_{op} \int_{t_1}^{t_{op}} \Delta T_g - \left(1 + \frac{C}{\sqrt{u}}\right) \Delta T_{cop} \right] dt$$ \hspace{1cm} (9)

For the ramp test situation, $\Delta T_g$ is substituted by $\beta t$ in equation (9). $RTI$, $C$ and $t_1$ are unknown. At least three plunge tests under different conditions or two plunge tests and a ramp test, are required to determine the unknown parameters.

**Solution**

The values of the $RTI$, $C$ and $CHP$ parameters have been calculated with computer programs developed by Gustafsson 3: TESTPLOT and SPRISENS. TESTPLOT determine the values of the response parameters in the two-parameter model while SPRISENS determine the values of the response parameters in the three-parameter model. A more thorough description on the solution of the differential equations presented here and the determination of the thermal response parameters, can be found in reference 8.

**Experimental set-up**

The wind tunnel used in the present study was built in accordance with the Fire Research Station (FRS) wind tunnel 6 in the United Kingdom. The wind tunnel was used for both plunge 1,21 and ramp 6 tests. The performance of the wind tunnel used here and the test procedure is more thoroughly described in reference 8.
A removable plywood sheet, to which a sprinkler head and a water cooling system were attached, was located at the top of the working section of the wind tunnel. The water cooling system was used to maintain the temperature of the sprinkler fitting at a relatively constant temperature, i.e. at the preconditioning temperatures given in Table 1. The sprinkler head and the steel pipe assembly of the water cooling system was preheated in a small furnace for a minimum of 10 minutes before each test. Corrections were made for possible reduction of the bulb temperature during transport of the assembly from the furnace to the working station.

The response time of the sprinkler was measured with a stop-watch. In the plunge test the timing started when the plywood sheet completely covered the working section, whereas in the ramp test the timing started when the control unit program started. The gas temperature in the plunge test was kept at 197 °C, except for one test series where the temperature was 147 °C, and the gas velocity was kept at 2.5 m/s. In the ramp test the gas temperature was increased linearly at two different temperature rates ($\beta$): 2 and 10 °C/min. The temperature tolerance was less than ±1 °C. The velocity was maintained at 1.0 m/s at the beginning of the test but was reduced very soon after the start of the test probably due to variations in the supply voltage, caused by switching on the heater. This was taken into account when calculating the $RTI$, $C$ and $CHP$ parameters. The water temperature in the water cooling system was maintained at 30 °C during the ramp test. It was observed that the water temperature increased slightly during the test at a rate of 2 °C/min.

The measured operation temperatures from sprinkler bath tests were as follows: Ø3 mm; $T_{eop}=69.1$ °C, Ø5 mm; $T_{eop}=71.6$ °C and Ø12 mm; $T_{eop}=71.8$ °C. The standard deviation of the bath tests were as follows: Ø3 mm; 0.8 °C, Ø5 mm; 1.4 °C and Ø12 mm; 2.1°C. This information was obtained from the sprinkler manufacturer. To reduce costs, the sprinklers were delivered from the manufacturer as separate glass bulbs and sprinkler frames. The sprinklers were carefully mounted on site before each test.

Test results

The results of the test series are summarised in Table 1. A more thorough data analysis is given in reference 8.

Analysis of the test results

One of the objectives of the present study was to investigate the validity of the three-parameter model. One way of investigating the applicability of a response model is to determine the thermal response parameters from a set of wind tunnel tests and use them to predict response times in wind tunnel tests performed under other conditions. This has been done using the test results presented in Table 1. Thus, different combinations of plunge tests and ramp tests were used to calculate the $RTI$, $C$ and $CHP$ parameters.

Table 1. Measured response times from plunge and ramp tests. The sprinklers were oriented perpendicular to the air flow in the working section of the wind tunnel. The sprinklers were preconditioned in a furnace for at least 10 minutes.
<table>
<thead>
<tr>
<th>Type of test</th>
<th>Number of tests</th>
<th>Preconditioning temperature (°C)</th>
<th>Gas temperature (°C)</th>
<th>Velocity (at start in a ramp test) (m/s)</th>
<th>Rate of temperature rise, $\beta$ (°C/min)</th>
<th>Average response time (s)</th>
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### Prediction of wind tunnel tests

In Figure 1, a predicted response time versus measured response time for the Ø 3 mm sprinkler in different plunge test situations is shown. In this graph, gas temperatures and preconditioning temperatures are given as $T_g$ and $T_o$ for each test situation. The gas velocity was 2.5 m/s in all tests. Calculated response times are based on a number of different combinations of $RTI$, $C$ and $CHP$ whereas measured are based on one average value for each test conditions.

The graph shows that the three-parameter model ($RTI + C + CHP$) gives improved predictions of response times and a reduced spread in the results compared to the two-parameter model ($RTI + C$). The same trend was obtained with the Ø 5 mm and the Ø 12 mm sprinklers. These findings are in accordance to the results obtained earlier by Gustafsson when he analysed the wind tunnel tests given in reference 5. The points showing the largest discrepancy between predicted and measured response times, when using the two-parameter model, are all related to response parameters calculated from tests with the highest preconditioning temperature, i.e. $T_o=52$ °C. In other cases, the prediction of the two-parameter model is reasonably good when using a combination of tests with more normal preconditioning temperatures (22 °C and 30 °C). Therefore, one could expect that this behaviour is related to some physical phenomenon occurring in a plunge test at high preconditioning temperatures and therefore it is motivated to add the third parameter, $CHP$, to improve the results. Since high preconditioning temperatures are from a practical point of view less interesting than ambient temperatures, these results show that the $CHP$ parameter is more of an academic than practical concern.
Similar comparisons were carried out for the ramp test situation. These calculations show that the correlation between prediction and observation is not substantially improved by the use of the three-parameter model.

**Prediction under real fire conditions**

The response models have also been evaluated under realistic fire conditions. A fire growth rate following a power law correlation as the square of time was used. According to Heskestad and Bill, the Heat Release Rate (HRR) at response of a sprinkler for growing fires, is a more meaningful measure of sprinkler responsiveness than conventional response time. Therefore, the thermal response parameters determined from the different test conditions, have been used to calculate the HRR value at sprinkler response.

The growth phase of many fires can be characterised by t²-type growth behaviour, as expressed below:

\[ Q = \alpha t^2 \]  

where \( \alpha \) is a constant characterising the increase in fire growth. Equation (10) is often a good representation of flaming and radial spreading fires in low fuel piles. The NFPA 72E standard lists three fires which have t²-type growth behaviour: a slow growth rate fire (\( \alpha = 0.00293 \text{ kW/s}^2 \)), a medium growth rate fire (\( \alpha = 0.0117 \text{ kW/s}^2 \)) and a fast growth rate fire (\( \alpha = 0.0469 \text{ kW/s}^2 \)).

From experiments and the use of dimensional arguments, Heskestad and Delichatsios obtained correlations for maximum temperature rise and velocity for unconfined ceiling layer flows as a function of the radial distance from the
plume axis. These correlations have been used in the calculation of the sprinkler response for slow, medium and fast growth rate fires. The calculations were performed using a computer program specially designed to solve the basic differential equations for a sprinkler response, i.e. equations (2), (5) and (8). A first-order Runge-Kutta scheme was applied to solve the equations numerically.

The calculations were carried out for the tested sprinklers at two ceiling heights, $H=3$ m and 6 m. The ambient temperature of the sprinkler was $T_0=22$ °C, and the radial distance from the fire axis was 2.12 m. The results are presented in the form of a normalised HRR in Figure 2. Calculations of HRR at sprinkler response with one- ($RTI$), two- ($RTI + C$) and three-parameter ($RTI + C + CHP$) models, were made where the HRR values have been normalised to the lowest HRR value, obtained from a calculation using the one parameter model.

In all the cases, the slow fire growth rate gave the lowest values of HRR at the sprinkler response. Figure 2 demonstrates the effect of using two or three parameters compared with the use of one parameter. Relatively small changes in the calculated HRR at sprinkler response are observed between the three-parameter model and the two-parameter model. Thus, the three-parameter model is questionable from a practical point of view.

![Figure 2](image)

**Figure 2.** Normalised HRR at sprinkler response of a Ø 3 mm sprinkler at a height of 3 m (slow fire growth rate). The lowest HRR value obtained was used to normalise the other values.

The same conclusion can be drawn as Heskestad and Bill in their sensitivity analysis of the one- and two-parameter model, i.e. the effect of $C$ is quite large at low $RTI$ (Ø 3 mm sprinkler) and less significant at higher values of $RTI$ (Ø 12 mm sprinkler). The effect of $C$ for the Ø 12 mm sprinkler is almost negligible for the fast growth rate fire while there is a slight difference for the slow growth rate fire.

In Figure 2 a distinction is made between calculations with parameters determined with ramp tests based on $\beta=10$ °C/min and $\beta=2$ °C/min. This is done in order to clarify the effects of different rates of temperature rise, $\beta$, on the calculated HRR value at sprinkler response. Higher $C$ values were obtained with tests based on $\beta=10$ °C/min than with tests based on $\beta=2$ °C/min. This is reflected here because higher HRR values at sprinkler response are obtained with...
parameters determined from ramp tests with $\beta=10$ °C/min than from tests with $\beta=2$ °C/min. The difference is greater for fast response sprinklers (Ø 3 mm) than for slow response sprinklers (Ø 12 mm). The difference in the C values causes up to a 19 % increase in HRR values at sprinkler response (Ø 3 mm, slow fire growth rate, $H=3$ m). The corresponding increase in the C value is 44 %.

**Thermal analysis of the glass bulb**

To investigate the discrepancies revealed earlier, a thermal analysis using computer simulation was carried out. A computer program based on the Finite Element Method (FEM) was used for the analysis.

**Heat transfer theory for a glass bulb**

A correlation for convective heating of a single cylinder in a cross flow is given by equation (11), expressed using the Nusselt number (Nu), the Reynolds number (Re) and the Prandtl number (Pr).

$$Nu = C_0 (Re)^n Pr^i$$

(11)

where $C_0$ and $n$ can be found as tabulated values in Holman and the Reynolds number, Nusselt number and Prandtl number are defined as:

$$Re = \frac{u D}{\nu_f} ; \quad Nu = \frac{h D}{k_f} ; \quad Pr = \frac{C_p \mu}{k}$$

where $h$ is a heat transfer coefficient (W/m² K), $D$ is a characteristic length; diameter of the cylinder (m), $k_f$ is the thermal conductivity of gas at film temperature; $T_f = (T_g + T_b)/2$ (W/m K), $u$ is the air velocity (m/s), $\nu_f$ is the kinematic viscosity of gas at film temperature (m²/s), $\mu$ is the dynamic viscosity (kg/m s), $T_g$ is the gas temperature (K) and $T_b$ is the surface temperature of the glass bulb (K). Equation (11) can be found in a similar form in McAdams. The convective heat transfer coefficient, $h$, can be calculated from the definition of the Nusselt number. For all the simulated test situations the Reynolds number was between 40 and 4000. According to Holman, the values of $C_0$ and $n$ can be put equal to 0.683 and 0.466, respectively.

It was assumed that pure heat conduction is the only heat transfer mechanism in the liquid. It can be shown that this is a realistic assumption by estimating the Grashof number ($Gr$) for free convection within the glass bulb. At very low Grashof numbers, there are very minute free-convection currents and the heat transfer occurs mainly by conduction across the fluid layer. In Holman, the limit of the pure conduction regime is defined by the requirement that $Ra < 10^3$ where $Ra$ (Rayleigh number) is defined as the product of $Gr Pr$. According to Holman this is valid for a fluid contained between two vertical plates separated by the distance, d. The Grashof number is given by the following equation:
Gr = \frac{g\beta T(T_1 - T_2)\delta^3}{\nu^2} \tag{12}

where $\beta T$ is the thermal expansion coefficient (K$^{-1}$), $d$ is a characteristic length (m), $\nu$ is the kinematic viscosity (m$^2$/s), $T_1$ is the temperature at surface (°C) and $T_2$ is the temperature at centre of the bulb (°C). The temperatures $T_1 = 83$ °C and $T_2 = 65$ °C were obtained from computer simulation for the Ø 5 mm glass bulb. The acceleration due to gravity, $g$, is 9.81 m/s$^2$ and the following data$^{16}$ were used for glycerine at 50 °C: $\beta T = 5.0 \times 10^{-4}$ (K$^{-1}$), $\nu = 1.5 \times 10^{-4}$ (m$^2$/s) and $Pr = 1.63$. The characteristic length, $d$, was assumed to be equal to the radius of the glass bulb, 0.0025 m. The Rayleigh number obtained was 0.1 which is much less than 10$^3$, for the Ø 5 mm glass bulb. Thus, it is realistic to assume that the heat transfer in the Ø 5 mm glass bulb can be simulated by pure heat conduction.

In the computer simulation the glass bulb was assumed to be axisymmetric. Assuming that the material properties do not vary with temperature, the axisymmetric temperature distribution can be determined by the following differential equation:

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial z^2} = \frac{\rho c_T}{k} \frac{\partial T}{\partial t} \tag{13}$$

where $k$ is the heat conductivity (W/m K), $\rho$ is the density of the material (kg/m$^3$), $r$ is the radius (m) and $z$ is the axial coordinate (m). The boundary conditions were varied in accordance with the test situation. The convective heat flow and radiation at the boundary were determined using the following equation:

$$q'' = h(T_g - T_b) + \varepsilon\sigma(T_w^4 - T_b^4) \tag{14}$$

where $q''$ is the heat flux to the boundary (W/m$^2$), $T_w$ is the surrounding temperature (for example the wall temperature in the wind tunnel) (K), $\varepsilon$ is the glass emissivity and $\sigma$ is the Stefan-Boltzman constant, $5.67 \times 10^{-8}$ (W/m$^2$ K$^4$). Equation (13) was solved numerically by the computer program SUPER -TASEF$^{18}$ (Temperature Analysis of Structures Exposed to Fire) which is based on the finite element method and can solve the transient non-linear heat flow equation for both axisymmetric and planar geometries. The boundary conditions can be chosen arbitrarily depending on the test situation. To reduce the computation time, only one quarter of the glass bulb was simulated (see Figure 3). This was possible due to the symmetry of the bulb.

It is difficult to determine the thermal properties of the liquid in the glass bulbs as the manufacturer do not declare the composition of the contents, although it is reasonable to assume an alcohol-based liquid such as glycerine. Comparison between weighed and calculated liquid masses for the three different bulbs gave best agreement for the Ø 5 mm glass bulb. Therefore, it was decided to use the Ø 5 mm glass bulb for a complete computer simulation. The following thermal properties were obtained from tabulated data in Holman for glass and glycerine at 30°C: $k_{glass} = 0.78$ W/m °C, $\rho_{glass} = 2700$ kg/m$^3$, $c_{p,glass} = 840$ J/kg °C, $k_{liquid} = 0.286$ W/m °C, $\rho_{liquid} = 1258$ kg/m$^3$ and $c_{p,liquid} = 2445$ J/kg °C. The
temperature dependence of the thermal properties $k$, $\rho$ and $c_p$ of glycerine is small and therefore has been neglected in this analysis.

In Figure 3 the Ø 5 mm sprinkler and the dimensions of the idealised glass bulb used in the simulation are shown.

![Diagram of sprinkler fitting and glass bulb dimensions](image)

**Figure 3.** The shape of the Ø 5 mm glass bulb is simplified to a rectangular form to facilitate the calculations. One quarter of the bulb (axisymmetric) was used in the simulation.

**Comparison between the calculated and measured temperature in a free glass bulb**

To investigate whether it is reasonable to assume pure heat conduction in the liquid and whether the assumed thermal properties of the liquid were reasonable, comparison between calculated and measured liquid temperatures within a glass bulb exposed to two different boundary conditions was carried out. Temperatures were compared for a glass bulb cooled by free convection and heated by forced convection in a plunge test, respectively.

In Figure 4 the temperature reduction in a Ø 5 mm glass bulb, suddenly removed from a furnace at 55.4 °C, is shown. The glass bulb was not mounted in the sprinkler frame. Two identical experiments were performed. The thermocouple used to measure the liquid temperature was of type K with a wire diameter of 50 µm. The thermocouple wires were twined together along the axial centreline. The measured temperature is expected to be the average temperature along the axial length of the glass bulb. The heat transfer coefficient due to free convection was calculated using the definition of the Nusselt number.
Figure 4. Comparison of the measured liquid temperature in a Ø 5 mm glass bulb and computer simulation. The glass bulb have been cooled by free convection.

Figure 5. A glass bulb in a plunge test situation with; \( T_g =197 \, ^\circ\text{C}, \, u=2.5 \, \text{m/s} \) and \( T_0=27 \, ^\circ\text{C} \). The response time is based on experiments without a sprinkler frame.

The following values were used in the calculation: \( T_f=312 \, \text{K}, \, \beta_T = 1/T = 3.2 \times 10^{-3} \, (\text{K}^{-1}), \, \nu_f = 16.6 \times 10^{-6} \, (\text{m}^2/\text{s}), \, k_f = 27.1 \times 10^{-3} \, (\text{W/m K}) \) and \( D= 0.005 \, \text{m} \). The Nusselt number obtained was 3.55 which yielded a heat transfer coefficient of
The heat losses from the glass bulb were calculated with the boundary conditions defined in equation (14). The temperature within a glass bulb (without frame) in a plunge test situation was also measured. The plunge test conditions were as follows: $T_g=197 \, ^{\circ}C$, $u=2.5 \, m/s$ and $T_0=27 \, ^{\circ}C$. The results of the measurements are presented in Figure 5. For the computer simulation $h=72 \, W/m^2 \, K$ was used. The following values were used to calculate $h$: $T_f=385 \, K$, $\nu_f=24.36 \times 10^{-6} \, (m^2/s)$, $k_f=32.6 \times 10^{-3} \, (W/m \, K)$, $D=0.005 \, m$ and $Pr=0.7$ which yielded $Re=513$ and $Nu=11.1$.

The measured temperatures are in good agreement with the calculated temperatures. The good agreement obtained in Figures 4 and 5, confirms that the computer simulation gives reasonable results for the selected thermal properties and boundary conditions.

**Computer simulation of a Ø 5 mm sprinkler bulb mounted in the frame**

The next step was to carry out a computer simulation of the experimental tests with the Ø 5 mm sprinkler bulb mounted in a steel frame. The glass bulb was assumed to be heated by forced convection and cooled by heat losses to the sprinkler fittings. The heat transfer coefficient, $h$, was calculated according to equation (11) and the definition of the Nusselt number. The heat loss coefficient, $h_1$, defined here as the heat transfer coefficient at the ends of the glass bulb (heat loss to the sprinkler frame), was determined by fitting the calculated response time to the measured response time in a ramp test with $\beta=2 \, ^{\circ}C/min$. The best fit was found for $h_1=270 \, W/m^2 \, K$. The results of the calculations are given for both ramp and plunge tests in Table 2, where $T_f$ corresponds to the water temperature at the sprinkler fitting and $T_0$ corresponds to the preconditioning temperature of the glass bulb.

**Table 2. Results from calculations with the computer program SUPER-TASEF.**

<table>
<thead>
<tr>
<th>Test</th>
<th>$T_0$ (°C)</th>
<th>$T_f$ (°C)</th>
<th>$T_g$ (°C)</th>
<th>$h_1$ (W/m² K)</th>
<th>$h$ (W/m² K)</th>
<th>$\text{top-exp}_\text{TASEF}$ (s)</th>
<th>$\text{top-exp}_{\text{SUPER-TASEF}}$ (s)</th>
<th>Difference (s)</th>
<th>Remarks</th>
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</thead>
<tbody>
<tr>
<td>Plunge</td>
<td>30</td>
<td>197</td>
<td>72</td>
<td>17.7</td>
<td>17.6</td>
<td>0.1</td>
<td>no frame</td>
<td></td>
<td></td>
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<tr>
<td>&quot;</td>
<td>22</td>
<td>197</td>
<td>72</td>
<td>22.4</td>
<td>22.3</td>
<td>-0.1</td>
<td>frame</td>
<td></td>
<td></td>
</tr>
<tr>
<td>&quot;</td>
<td>22</td>
<td>197</td>
<td>72</td>
<td>22.4</td>
<td>22.5</td>
<td>-1.1</td>
<td>insulated frame</td>
<td></td>
<td></td>
</tr>
<tr>
<td>&quot;</td>
<td>30</td>
<td>197</td>
<td>72</td>
<td>18.5</td>
<td>19.1</td>
<td>-1.3</td>
<td>frame</td>
<td></td>
<td></td>
</tr>
<tr>
<td>&quot;</td>
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<td>197</td>
<td>72</td>
<td>18.5</td>
<td>19.8</td>
<td>-1.3</td>
<td>frame</td>
<td></td>
<td></td>
</tr>
<tr>
<td>&quot;</td>
<td>52</td>
<td>197</td>
<td>72</td>
<td>10.5</td>
<td>11.8</td>
<td>-1.3</td>
<td>frame</td>
<td></td>
<td></td>
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<tr>
<td>&quot;</td>
<td>52</td>
<td>197</td>
<td>72</td>
<td>10.5</td>
<td>11.8</td>
<td>-1.3</td>
<td>frame</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Ramp</td>
<td>30</td>
<td>30</td>
<td>$\beta=2$</td>
<td>270</td>
<td>2075</td>
<td>2075*</td>
<td>-</td>
<td>frame</td>
<td></td>
</tr>
<tr>
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<td>30</td>
<td>$\beta=10$</td>
<td>270</td>
<td>519</td>
<td>447</td>
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<tr>
<td>&quot;</td>
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<td>30</td>
<td>$\beta=2$</td>
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<td>$\beta=10$</td>
<td>-</td>
<td>47</td>
<td>329</td>
<td>-</td>
<td>insulated frame</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

* $h_1 = 270 \, W/m^2 \, K$ was found to fit the data for $\beta=2 \, ^{\circ}C/min$. 
For simplicity, the response time, \( t_{\text{topS-TASEF}} \), (based on calculations using SUPER-TASEF) was determined by calculating the average temperature from 18 node points in the liquid of the glass bulb (total of 95 node points in the liquid) and interpolating the time at which the average temperature was equal to the operation temperature, 71.6 °C. Here, it is assumed that the average temperature of 71.6 °C corresponds to the pressure, caused by thermal expansion, at which the bulb will break. Using all 95 nodes available in the liquid, instead of 18 nodes, resulted in a 0.5 % difference in the average temperature in the plunge test situation. A more correct method for the plunge test situation (a steep gradient across the bulb) may have been to use a weighted average using the radial distance to each point. Rough calculations indicate that the difference is probably less than 1 °C.

In Table 2, the heat transfer coefficient, \( h=47 \, \text{W/m}^2\, \text{K} \), for the ramp test, is an average value calculated by using the measured gas temperature and velocity during the test. This value was relatively constant for both the rates of temperature rise used in the ramp test. The value of \( h_1=190 \, \text{W/m}^2\, \text{K} \) was a "first guess" value and it was obtained by using equation (15):

\[
h_1 = \frac{C'}{A} = \frac{C \, mc_p}{RTI \, A}
\]

where \( A \) is the contact area between the frame and the glass bulb (m²). The correlation between \( h_1 \) and the parameter \( C \) was obtained from the definition of \( C \), equation (6). The following values were used to calculate the first value of \( h_1 \); \( C=0.69 \, \text{m}^{1/2}\,\text{s}^{1/2}, \, RTI=97.4 \, \text{m}^{1/2}\,\text{s}^{1/2}, \, mc_p=1 \, \text{J/°C}, \, A=3.8 \times 10^{-5} \, \text{m}^2 \) (the area of both ends of the bulb).

**Discussion of the results obtained for the plunge test**

The correspondence between calculated and measured operation times is very good for the plunge test, see Table 2. The heat transfer parameters used were calculated from equations for a cylinder in a cross flow without any correlation to the test results. The plunge tests with sprinklers at high preconditioning temperatures (52 °C) are of special interest here since Gustafsson has attributed the behaviour of glass bulb sprinklers tested at high preconditioning temperatures to a time delay caused by the work necessary to shatter the bulb (the temperature within the bulb does not increase during this time). The simulations presented in this paper show a relatively small difference between measured and calculated response times at high preconditioning temperatures. The calculations indicate, therefore, that the heating process within the glass bulb is sufficiently described by pure heat conduction up to the time of operation of the bulb.

The calculations also show that the temperature gradient is relatively steep across the glass bulb whereas it is rather weak along the axial length of the glass bulb. The weak temperature gradients indicate small heat losses to the sprinkler fitting. This is actually reflected in the calculation since the effects of changing \( h_1 \) from 190 to 270 W/m² K has little effect on the operation time as seen in Table 2. Further investigation of the effects of non-uniform temperature distribution across the glass bulb on the constancy of the \( RTI \) parameter are presented later in this paper.
Discussion of the results obtained for the ramp test

It was observed that the value of the parameter $C$ was higher when determined from ramp tests with $\beta=10 \, ^\circ\text{C/min}$, than ramp tests with $\beta=2 \, ^\circ\text{C/min}$. For one of the tested sprinkler types, the value of the parameter $C$ increased by 43 % when using a rate of temperature rise of 10 $^\circ\text{C/min}$ as opposed to 2 $^\circ\text{C/min}$. This is reflected in the simulation since the value of the heat loss coefficient, $h_1$, must be considerably higher than 270 W/m$^2$ K to obtain the same response time as was measured in the tests with $\beta=10 \, ^\circ\text{C/min}$. According to equation (15), a higher value of $h_1$ corresponds to a higher $C$ value as all the other parameters are constant. One conjecture concerning the difference in the $C$ values is that the cooling efficiency was not the same for $\beta=10 \, ^\circ\text{C/min}$ and $\beta=2 \, ^\circ\text{C/min}$ tests.

![Figure 6. Measured surface temperatures on the sprinkler frame during a ramp test with $\beta=2 \, ^\circ\text{C/min}$.](image)

To further investigate this conjecture, the cooling effects of the water in the sprinkler fitting on the frame were measured by attaching two Ø 0.12 mm, type K, thermocouples with highly conductive glue to the sprinkler frame. One was placed close to the sprinkler fitting ($T_A$) and one close to the sprinkler deflector ($T_B$). Ramp tests were carried out with $\beta=2 \, ^\circ\text{C/min}$ and $\beta=10 \, ^\circ\text{C/min}$. The results with $\beta=2 \, ^\circ\text{C/min}$ are shown in Figure 6. At sprinkler response, $T_B$, is about 10 $^\circ\text{C}$
higher than $T_A$. The temperature, $T_B$, was found to be about 5 °C higher at sprinkler response for $\beta=2$ °C/min than for $\beta=10$ °C/min, whilst $T_A$ was about the same. This indicates that the heat loss may have been larger at $\beta=10$ °C/min, i.e. a higher $C$ value. Further, new simulations with SUPER-TASEF indicate that this temperature difference is sufficient to yield comparable difference in the $C$ values as obtained by SPRISENS and TESTPLOT.

Apparently, it is difficult to obtain the boundary conditions described by equation (4) with the procedure applied here. Thus, the use of different ramp tests to determine the coefficient $C$ is questionable. The cooling method at the sprinkler fittings must be described accurately and a better method to ensure the boundary conditions is to use the procedure described by Heskestad and Bill $^2$, $^{22}$.

**Investigation of effects of time delay**

The assumption of a uniform temperature distribution within a glass bulb has been discussed by Job et al. $^{20}$ They showed that an initial delay in the temperature rise within the glass bulb occurred in a plunge test situation. This resulted in a delay in the response time of the sprinkler. As a consequence different RTI values for different heating conditions will be obtained and the glass bulbs will not respond as predicted. They concluded that for a wide range of heating conditions the time delay would be constant for each type of glass bulb and introduction of the time delay as a third parameter was proposed. They also concluded that the time delay could be neglected during heating conditions such as those in plunge tests with low temperatures or velocities, in ramp tests and particularly in fire situations.

The effects of time delay were investigated here by using SUPER-TASEF. The heat loss to the sprinkler frame was neglected to enhance the effects of the temperature gradients across the glass bulb. This condition corresponds to the one-parameter model. Similar plunge test conditions to those used by Job et al. $^{20}$ were used. The calculations show that the assumption of a uniform temperature distribution will result in different RTI values depending on the plunge test situation. The deviation between the highest and the lowest value obtained for the simulated cases was 25 % ($T_g=100^\circ$C+$u=0.65$ m/s and $T_g=197^\circ$C + $u=2.5$ m/s, respectively). The discrepancies in the RTI values are effects of the time delay. The relative effects of the time delay increase with more severe plunge test situations (shorter response times) subsequently leading to higher RTI values. This means that increasing gas temperature and increasing gas velocity will increase the RTI value for a certain glass bulb (assuming no heat losses). The time delay was found to be nearly constant for all the simulated plunge situations (average of 5.8 s). Simulation in a ramp test with $\beta=10$ °C/min resulted in no time delay.

To obtain a constant RTI value for all the test conditions, the response time must be corrected. This can be done by subtract the time delay for sprinklers tested from the measured response times and then calculate a new corrected RTI value $^8$, $^{20}$. The effects of heat losses to the sprinkler frame on the results have not been studied.

The RTI value is expected to be higher for higher preconditioning temperatures because of pronounced effects of the time delay. To show this a calculation with $T_0=52$ °C and $T_0=30$ °C using $T_g=197$ °C and $u=2.5$ m/s was performed. The obtained RTI value for $T_0=52$ °C was 127 while that for $T_0=30$ °C was 105.5. This is exactly the same tendency as was observed for the tested sprinklers. The higher
The RTI value for a preconditioning temperature of 52 °C is, therefore, more likely to be caused by effects of the time delay on the calculated RTI value rather than on any work required to shatter the bulb shortly before activation. This can explain the difficulties in obtaining agreement between predicted and observed response times using the two-parameter model when response times based on tests with preconditioning temperature of 52 °C were used to calculate the RTI and C parameters.
Effects of radiation

An investigation of radiation was also carried out. The effects of radiation on the thermal response parameters were found to be greater in a ramp test than in a plunge test. The systematic error caused by radiation exchange in a ramp test will, however, have a negligible effect on the calculated response time in a fire situation. The maximum increase in predicted heat release rate at response of the sprinklers was found to be 3%.

Conclusions

The thermal behaviour of glass bulb sprinklers was investigated in a heated wind tunnel under various test conditions. Special attention was given to the three parameter model proposed by Gustafsson. The results can be summarised as follows:

1. The wind tunnel tests can not be fully explained within the framework of the thermal response models applied in the analysis. Firstly, consistently different $C$ values were obtained for ramp tests with different rates of temperature rise. This tendency was observed for all three sprinkler types tested. Secondly, tests with a preconditioning temperature close to the operation temperature resulted in higher $RTI$ values than in cases with normal preconditioning temperatures.

2. Measurements of the frame temperatures and a thermal analysis with a finite element computer program shows that a plausible reason for the different $C$ values is an uneven heating of the sprinkler frame arms during different ramp tests. Calculations showed that to obtain a constant $C$ parameter from different ramp tests, the average temperature at the two contact points between the glass bulb ends and the frame (at the button and near the deflector, respectively), must be equal at sprinkler response. This was probably not the case in the experiments.

3. Thermal analysis showed that the higher $RTI$ values obtained with a preconditioning temperature close to the operation temperature could be attributed to a time delay caused by temperature gradients within the glass bulb. This indicates that the $CHP$ parameter deemed to be necessary to explain some earlier test data in terms of heat of fusion of a link or work to shatter a glass bulb, is more probably an effect of the thermal time delay caused by the non-uniform temperature in the glass bulb.

4. Thermal analysis showed that the time delay will yield different $RTI$ values in a plunge test for different heating conditions and consequently the glass bulbs will not respond as predicted. It was also found that the relative effects of the time delay on the $RTI$ value will increase with shorter response times. The time delay was found to be relatively constant for the plunge test conditions, while for time varying heating conditions, such as those in ramp tests, no time delay was found.

5. Estimates of heat release rates at sprinkler response for fast, medium and slow fire growth rates were found to be almost identical, whether based on calculations with two parameters ($RTI$ and $C$) or with all three parameters ($RTI$, $C$, $CHP$). However, when using two parameters instead of one ($RTI$), the calculated rate of heat release at
sprinkler response was found to increase significantly. This is probably due to the relatively large effect of the parameter $C$ (the heat losses to the sprinkler fittings) on the total time before sprinkler response. The small difference in the calculated heat release rates at sprinkler response when using two- or three-parameter models under a real fire conditions shows that the gain in using three parameters instead of two is negligible in practice.

6. The effects of radiation on the thermal response parameters were found to be greater in a ramp test than in a plunge tests. However, the systematic error caused by radiation exchange in a ramp test will have negligible effects on the calculated response time in a fire situation. The maximum increase in predicted heat release rate at response of the sprinklers was found to be 3%.

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19. Private communication with S. Olsson, Swedish National Testing and Research Institute, Borås.


Paper II

Numerical Simulation of the Wind Shadow Effect on the Convective Heat Transfer to Glass Bulb Sprinklers

Haukur Ingason & Bror Persson

Paper III

Two Dimensional Rack Storage Fires

Haukur Ingason

Paper IV

Modelling of a Two Dimensional Rack Storage Fires

Haukur Ingason

Submitted to Fire Safety Journal (1996)
Modelling of a Two Dimensional Rack Storage Fire

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ABSTRACT

A theoretical model to predict mass flow rate, gas temperature, gas velocity and flame height in a two dimensional rack storage fire has been developed. Experiments have been carried out using inert boxes and a propane diffusion flame on a line burner located at the bottom of the rack. Reasonably good correlation is found between the theoretical model and the experiments.

The model calculates the flow within the rack with the aid of the equations of continuity, and conservation of momentum and energy. The model distributes fractions of the total convective heat from the burner along a calculated flame height in accordance with an empirical relationship obtained from measured convective heat. The flame height is calculated by comparing the amount of available air flow and fuel at each height for different heat release rates and widths of regular spaces between non-combustible boxes.

The theoretical results were compared to experimental results based on an approximately two dimensional, 4 boxes high rack storage with a total height ranging from 1.14 m to 1.34 m and with a 0.59 m long diffusion line burner placed at the bottom of the vertical flue. The size of the rack is about 1/3 that of real rack storage.

NOTATION

\( b \) width of the box (m)
\( C_p \) specific heat (kJ/kg K)
\( D_{H1} \) the hydraulic diameter of the horizontal flue (m)
\( D_{H2} \) the hydraulic diameter of the vertical flue (m)
\( f_D \) the Darcy friction factor
\( g \) acceleration of gravity (m/s²)
\( h \) height of the horizontal flues (m)
\( \Delta H_c \) heat of combustion per unit mass of the fuel (J/kg)

\( \text{head loss at the entrance of the lowest horizontal inlet} \)
head loss due to friction inside the lowest horizontal inlet
\( (N \text{ m/kg}) \)

head loss due to the burner and change in direction of the flow \( (Nm/kg) \)

head loss due to a converging flow at the \( i \)th intersection \( (Nm/kg) \)

\( k_{\text{non-c}} \) correction factor due to use of other cross sections than circular

loss coefficient at the tee-intersection where the burner is located

loss coefficient for the converging flow at the \( i \)th intersection

\( l \) longitudinal length of the rack (m)

\( L_f \) flame height (m)

\( m_i \) mass flow rate of air in a vertical flue at the \( i \)th tier \((i=1, 2, \ldots, N)\) \((kg/s)\)

\( m_{0i} \) mass flow rate of air in a horizontal flue at the top of the \( i \)th tier \((kg/s)\)

\( m_f \) fuel flow rate \((kg/s)\)

\( m_e \) total mass-entrainment rate of air below the level of the flame tip \((kg/s)\)

\( P_s \) perimeter of a vertical flue (m)

\( p_0 \) hydrostatic pressure \(1/2h\) above floor level \((N/m^2)\)

\( p_{1,b} \) static pressure at the base of the first tier \((N/m^2)\)

\( p_{N,a} \) static pressure at the top of the rack storage, i.e the \( N \)th tier \((Pa)\)

\( p_{N,b} \) static pressure at the base of the \( N \)th tier \((Pa)\)

\( p_{i,a} \) static pressure at the top of the \( i \)th tier \((Pa)\)

\( p_{i,b} \) static pressure at the base of the \( i \)th tier \((Pa)\)

\( p_{0i'} \) hydrostatic pressure at the top of the rack storage \((Pa)\)

\( p_{0i'} \) hydrostatic pressure outside the rack at the \( i \)th tier \((Pa)\)

\( p_{i'} \) static pressure in the horizontal flue at the \( i \)th tier \((Pa)\)
\( p_i'' \)  static pressure at the top of the box at the \( i \)'th tier (Pa)
\( Q \)  total heat release rate from the burner (kW)
\( Q_c \)  convective heat release rate from the burner (kW)
\( Q_{c,exp} \)  convective heat release rate based on single point measurements within the rack (kW)
\( Q_{c,i} \)  accumulated convective heat release rate at the top of the \( i \)'th tier (kW)
\( Q_{c,b} \)  accumulated convective heat release rate at the base of the \( i \)'th tier (kW)
\( Re \)  Reynolds number
\( r \)  stoichiometric mass air-to-fuel ratio
\( s \)  height of the boxes (m)
\( T_{i,a} \)  temperature at the top of the \( i \)'th tier (K)
\( T_{i,b} \)  temperature at the base of the \( i \)'th tier (K)
\( T_0 \)  ambient temperature (K)
\( \Delta T \)  temperature difference (K)
\( u_{00} \)  velocity in the lowest horizontal flue (m/s)
\( u_{1,b} \)  velocity at the base of the first box (m/s)
\( u_{i,a} \)  velocity at the top of the \( i \)'th tier (m/s)
\( u_{i,b} \)  velocity at the base of the \( i \)'th tier (m/s)
\( V_i \)  volume flow in the vertical flue at the \( i \)'th tier (m\(^3\)/s)
\( V_{i-1} \)  volume flow in the vertical flue at the \( i-1 \)'th tier (m\(^3\)/s)
\( w \)  width of a vertical flue (m)
\( z \)  height from floor level (m)

**Greek symbols**

\( \rho_o \)  density at ambient temperature (kg/m\(^3\))
\( \rho \)  density within the rack storage (kg/m\(^3\))
\( \phi \)  air-to-fuel stoichiometric fraction

**INTRODUCTION**

Fire growth rate and flame propagation are dependent on how the burning material is stored. In rack storage fires this becomes extremely important as the upward flame propagation is usually very rapid. The vertical shafts, or flues, created between adjacent pallets of stored goods tend to work as chimneys and subsequently they enhance acceleration of flames up to the ceiling. Rack storage fire protection usually consists of in-rack sprinklers which are placed at different elevations in the vertical flues or on the faces of the rack storage. The efficiency of such protection depends on the geometry of the stored goods, their height, the flue dimensions and intervals of occurrence as well as the flammability of the stored goods.

The delay prior to in-rack sprinkler initiation is extremely important as this may be critical for control of the fire. To calculate the response time, a knowledge of the flow conditions close to the sprinkler is necessary. A theoretical model which can predict the flow conditions is therefore of interest. Such a model could benefit the optimisation of geometric parameters in large rack storage configurations with regard to sprinkler location. However, no simple theoretical model is presently available to predict the flow conditions. A fully developed rack
storage model could be used to predict fire growth rates and response times for in-rack sprinklers using different rack storage configurations. To establish a model like this, several steps must be undertaken, starting with simple geometries and simple combustion processes and progressing to more complex scenarios. In the present study the first step is taken by creating a model which can be used to calculate the flow conditions in a non-combustible two dimensional rack storage.

BACKGROUND

Models for buoyancy driven flows (created by a fire) in vertical shafts can be found in various studies\(^2,3\). The present work differs from these studies in that a rack storage usually consist of a multi-level (or tier) system and not of a single level. In this study the flow within each level is governed by the similar equations as those previously applied to single level problems. The boundary conditions at the ends of each level (i.e. at the intersections) are however different to those used in the single level case.

Heskestad and Yu\(^2\) determined the convective flow in a combustible circular stack with a simple model. The model is assumed to be valid until the material on the wall ignites. The model assumes that the inflowing cold air at the base of the stack is heated by a fire (which is located just below the base) over a small distance just inside the stack, raising the gas temperature from ambient to a constant bulk temperature. The air velocity at the inlet of the stack, created by the draft in the stack, is determined from a relationship developed by Delichatsios\(^3\). The velocity is derived from the momentum equation, assuming turbulent conditions in the stack. The static pressure at the top of the stack is assumed to be equal to the atmospheric pressure at the same elevation.

Block\(^4\) worked with a theoretical model for the burning rate of wood cribs. Although a wood crib is, by its construction, comparable to a rack storage, Block\(^4\) made an assumption which has not been applied in the present study, i.e., that all the air entering the vertical shafts enters at the bottom of the wood crib. A simple model presented by Ingason\(^1\) shows that this assumption is justified for two dimensional rack storage when the mass flow rate at the top of the rack is the parameter of interest.

Experimental study carried out by Ingason\(^1,5\) show that the width of the vertical flue in a 1/3 scale two dimensional rack storage configuration is the predominant geometrical parameter controlling the flue flow. The mass flow rate was found to increase nearly linearly with the width where the effect of the horizontal flue heights was small. For a given flue width, the mass flow rate in the vertical flue was nearly constant, independent of the heat output (\(>6\) kW).

Karlsson \textit{et al.}\(^6\) made similar experiments on 1/3 scale tests to examine the geometrical effects on the flame height. Correlations presented by Karlsson \textit{et al.}\(^6\) for flame heights clearly show that the effects of the horizontal flue heights are small.

In the present study no consideration is given to the problem of heat transfer from the flame to the surrounding walls which is of importance for flame spread studies. Foley and Drysdale\(^7\) recently published a thorough study of this problem where they used a similar configuration as in present study, i.e. two parallel walls
(open at both ends) and a propane line burner at the floor. They showed that the heat fluxes increase as the separation between the walls is reduced.

THEORY AND APPROACH

The theoretical model is based on simple relationships from fluid dynamics. It can be used to calculate the fire parameters in an inert environment in two dimensional (2-D) rack storage configurations. The experimental configuration is shown in Fig. 1 in three views.

The model predicts the mass flow rate of air, temperature, velocity and flame height in the flues. Gas temperature, gas velocity and mass flow rate are represented by a mean value across the cross section of the flue. The input parameters used in the model are:

- the longitudinal length of the rack (l)
- the height (s) and the width (b) of the boxes
- the width of the vertical flue (w)
- the height of the horizontal flues (h)
- the convective heat energy released from the propane line burner on the floor ($Q_c$).

![Diagram of rack storage configuration](image)

**Figure 1.** Three different views of the rack storage are shown. The wings shown in the birds-eye view were used to obtain a uniform flow of air into the horizontal flues ($l=0.59$ m, $s=0.235$ m, $b=0.22$ m, $w=0.05$, 0.075 and 0.1 m, $h=0.05$, 0.075 and 0.1 m).

In their work on combustible stacks, Heskestad and Yu \(^2\) assumed a point source just inside the inlet of the stack. This approach has been modified here as the flames usually extend over several boxes in height. It is known from measurements carried out by Tamanini \(^8\) on buoyancy-controlled open flames that the cumulative fraction of the total heat release rate within the flame increases as a function of height. Tamanini’s measurements show that above about half of the flame height (for a propane fire) most of the chemical heat energy has been
released. Based on this knowledge it was justified to establish a method to describe the heat energy release along the flame height. Figure 2 outlines how the convective heat energy is released along the flame height.

The following assumptions have been made:

1) No convective heat losses are assumed in the vertical flue. This means that the heat transferred to the walls by convection from the hot gases is neglected in the model (the radiant component is assumed lost through the radiative fraction, usually assumed to be 30% of the total heat release rate).

2) For simplicity, the gas temperature, $T_{i,b}$, at the base of each box, is assumed to be raised over a short distance to a bulk temperature, $T_{i,a}$, which is constant over the box height.

3) The static pressure, $p_{N,a}$, at the top of the rack storage, is equal to the hydrostatic pressure, $p_{0N}$, outside the rack (see Fig. 3).

4) The static pressure, $p_{i,a}$, at the centreline of the vertical flue is equal to the static pressure, $p_{i''}$, at the top of each intermediate box in the intersecting horizontal flue (see Fig. 3).

5) The air is assumed to be drawn through the horizontal flues due to pressure difference between the inside and outside of the rack. No air is assumed to "leak" out through the horizontal flues.

6) The cross sectional velocity and temperature distribution in the vertical flue is assumed to correspond to a "top hat" profile.

7) Steady-state turbulent flow is assumed within the flues.

8) Due to excess of air in the flue flow, the molecular weight of the mixture of entrained air and combustion products is approximated by the molecular weight of air.
Figure 2. The convective heat release rate \( Q_c \) from the burner is released as a function of height as illustrated by the diagram beside the rack.

To determine the mass flow rates of air inside the rack, various pressure drops are calculated. The mass flow rates, in turn, give the corresponding gas temperatures (densities) and velocities. Based on assumption (5) a lower pressure is created inside the rack. The pressure drops inside the rack are calculated by subtracting the pressure head changes due to entrance losses, obstructions (such as those due to the line burner), changes in upward momentum of the heated air, wall friction, minor losses at intersections of horizontal and vertical flues and changes in density.

A computer program, written in Fortran 77, was developed to calculate the pressures and flows within the rack. The program starts the calculations by "guessing" the mass flow rates followed by a calculation of the pressure conditions. An iteration process is established at each tier until an overall pressure balance is obtained between the atmospheric pressure and the pressure inside the rack. The equations of continuity and conservation of energy and momentum are used to calculate this flow.

**The continuity equation**

Using the continuity equation (conservation of mass) and assumption (5), we find that

\[
\begin{align*}
m_1 &= 2m_{00} \\
&m_i = m_{i-1} + 2m_{0i} \\
&m_N = m_{N-1} + 2m_{0N}
\end{align*}
\]
where \( m_i \) is the mass flow rate of air at the \( i \)’th tier and \( m_N \) is the mass flow rate of air at the highest tier. Ensuing equations will be written in general form where the index \( i \) has integral values ranging from \( i = 1 \) ... \( N \) where \( N \) is the number of the highest tier.

**Conservation of energy**

When the mass flow rate in the vertical flue at each tier is known (the mass flow rate is constant between the base and the top of each box) the temperature at corresponding height can be calculated using:

\[
Q_{c,i,a} = m_i C_p (T_{i,a} - T_0) \quad (2a)
\]

\[
Q_{c,i,b} = m_i C_p (T_{i,b} - T_0) \quad (2b)
\]

where \( Q_{c,i,a} \) and \( Q_{c,i,b} \) corresponds to the released convective energy at each height (see Fig 2). The program determines the convective heat energy at the top (\( Q_{c,i,a} \)) and the base (\( Q_{c,i,b} \)) of each box from an empirical distribution correlation based on measurements of the convective heat at each tier. Accordingly, the temperatures at corresponding locations can be calculated:

\[
T_{i,a} = T_0 + \frac{Q_{c,i,a}}{m_i C_p} \quad (3a)
\]

\[
T_{i,b} = T_0 + \frac{Q_{c,i,b}}{m_i C_p} \quad (3b)
\]

Using the relationship for the mass flow rate (\( m = \rho u w l \)) and the ideal gas law (\( \rho T = \rho_0 T_0 \)) where influences of pressure changes and gas composition are neglected, the velocity at the top and the base of each tier can be calculated using:

\[
u_{i,a} = \frac{m_i T_{i,a}}{\rho_0 T_0 w l} \quad (4a)
\]

\[
u_{i,b} = \frac{T_{i,b}}{T_{i,a}} \nu_{i,a} \quad (4b)
\]

**Calculation of pressure drop**

Definitions of pressures and geometric parameters are given in Fig. 3. The pressure index (\( 1a, 1b, 2a, \ldots \)) indicate the location of each point. For example, \( p_{i,a} \) corresponds to the static pressure at top of the \( i \)’th box while \( p_{i,b} \) correspond to that at the base of the \( i \)’th box. Similarly \( p_{i}' \) is the pressure at the middle of the \( i \)’th horizontal flue while \( p_{i}'' \) is the pressure at the base of the \( i \)’th horizontal flue (see Fig. 3).
Figure 3. The definition of the pressures inside and outside the rack is shown. The pressure is balanced at each tier by iteration and when the program has run once through all tiers, a new calculation round is started by calculating the pressure at the lowest tier again.

Based on assumption (4) it is possible to set up an iteration process to establish a pressure balance at each intersection. As an example, the pressure at the top of the $i'$th tier, $p_{i,a}$, is assumed to be equal to the pressure at the bottom of the $i'$th horizontal flue, $p_{i'}$, and this pressure is then 'forced' by iteration to be equal to that at the middle of the $i'$th horizontal flue plus the hydrostatic pressure difference between these points.

The pressure $p_{i'}$ is obtained by calculating the head losses in the horizontal flue. In each iteration step the value of $p_{i'}$ is compared to the value of $p_{i'}$ plus the hydrostatic pressure difference. Depending on whether the pressure $p_{i''}$ is too high or too low, the value of the mass flow rate $m_i$ is decreased or increased. When a pressure balance is obtained the program continues to the next tier.

After a pressure balance is obtained at the top of the rack, after the first calculation round, the procedure is repeated for all tiers. This procedure is carried out until an overall pressure balance for the rack storage is obtained.

The pressure at the top of each tier can be obtained from a force balance on the gas volume (control volume) between the base and the top of the tier:

\[ p_{i,b}wl - p_{i,a}wl = m_i(u_{i,a} - u_{i,b}) + \frac{fD}{4} \rho_{i,a}P_s \frac{u_{i,a}^2}{2} + \rho_{i,a}gwl \]  

The first term on the right hand side of equation (5) is due to change in momentum of the heated gas volume, the second term is due to wall friction and the last term is due to gravity. For simplicity we use $\rho_{i,a}$ and $u_{i,a}$ for calculating
the friction and gravity term. One could use average values of $\rho$ and $u$ but calculations indicate that the variation is small. Thus, using the ideal gas law

$$\rho_{i,a} = \rho_0 \frac{T_{i,a}}{T_0},$$

the mass flow rate ($m_i = \rho_{i,a}u_{i,a}W_I$) and continuity equation

$$u_{i,b} = \frac{T_0}{T_{i,a}} u_{i,a}$$

we find that:

$$P_{i,b} = \frac{p_{i,b}}{\rho_0} = \frac{p_{i,b}}{p_0} - \frac{T_0}{T_{i,a}} \left( 1 - \frac{T_{i,a}}{T_0} \right) u_{i,a}^2 - f_D \frac{T_0}{D_{H2}} \frac{s}{2} - \frac{T_0}{T_{i,a}} g s$$

(6)

The pressure at the base of each tier (except the lowest) is obtained by using the modified Bernoulli equation:

$$p_{i,b} = p_{(i-1),a} + \frac{T_0}{T_{(i-1),a}} \frac{u_{(i-1),a}^2}{2} - \frac{T_0}{T_{i,b}} \frac{u_{i,b}^2}{2} - \frac{T_0}{T_{(i-1),a}} \left( \frac{1}{T_{(i-1),a}} + \frac{1}{T_{i,b}} \right) gh - h_e$$

(7)

where

$$h_e = K_{li} \frac{T_0}{T_{i,b}} \frac{u_{i,b}^2}{2}$$

The second and the third term on the right hand side of equation (7) originate from the "dynamic" head change, the fourth term relate to the average weight of the gas volume and the fifth term is due to the head loss at the intersection where horizontal inflow collides with upward moving flow in the vertical flue. The coefficient $K_{li}$ can be obtained from Table B-(6-36) in the ASHRAE handbook 10 for converging flows at an intersection. The coefficient depends on the ratio of the volumetric flows ($m^3/s$), $V_i$ and $V_{i-1}$, between the tiers. The loss coefficient, $K_{li}$, was determined from a regression formula made from the tabulated data (table 6-36) in the ASHRE handbook 10 ($K_{li} = 1.22 - 1.027(V_i / V_{i-1})^{2.015}$).

The pressure at the top of the highest tier, $p_{N,a}$, can be determined using eqn. (6). The pressure, $p_{N,a}$, can also be calculated using assumption (3), i.e. $p_{N,a}$ is assumed to be equal to the static pressure $p_{0N}$ outside the rack at the appropriate height. This means that $p_{N,a}$ from eqn. (6) can be balanced to following equation:

$$\frac{p_{N,a}}{\rho_0} = \frac{p_0}{\rho_0} - g (N(s + h) - \frac{1}{2}h)$$

(8)

Calculation of $p_i'$ (see Fig. 3) assumes that the program knows the mass flow rate $m_{i+1}$ in the vertical flue. Thus, in the first round of calculation the program increases the value of $m_i$ by 30% and sets that value to $m_{i+1}$. The mass flow rate $m_{0i}$ in the horizontal flue is, by definition, half the difference between $m_{i+1}$ and $m_i$. The velocity in the horizontal flue can be determined from $m_{0i}$. The static pressure $p_i'$ can be calculated using two methods. Let us call these methods A and
B and the corresponding pressures $p_{i,A}'$ and $p_{i,B}'$. The pressure $p_{i,A}'$ can be determined using:

$$\frac{p_{i,A}'}{\rho_0} = \frac{p_{i,a}}{\rho_0} - \frac{1}{2} gh$$

(9)

while the pressure $p_{i,B}'$ is obtained using:

$$\frac{p_{i,B}'}{\rho_0} = p_0 - 1.5 \frac{u_{00}^2}{2} - f_D \frac{b}{D_{fl}} \frac{u_{00}^2}{2} - gi(s + h)$$

(10)

The pressures are balanced such that the difference between the pressures $p_{i,A}'$ and $p_{i,B}'$ is equal to or less than 0.005 Pa before the program continues to the next tier. This maximum allowable pressure difference was an arbitrarily chosen value. It was found to correspond to between 1 and 5 % of the pressure difference over the horizontal flue, $(p_{i}' - p_{0i}')$. The pressure differences $(p_{i}' - p_{i}''')$ and $(p_{i}' - p_{0i}')$ are of the same order of magnitude.

It was found that the pressure $p_{i,B}'$ was not very sensitive as the air velocity is usually low (<0.4 m/s) in the horizontal flues. Consequently, the pressure calculations converged relatively quickly as $p_{i,B}'$ did not change very much.

The static pressure $p_{i,B}$ at the base of the first tier (box) can be obtained by using the modified Bernoulli equation between the base of the first box (just above the line gas burner) and the ambient air outside the rack as below:

$$\frac{p_{i,B}}{\rho_0} = \frac{p_0}{\rho_0} - \frac{u_{1,b}^2}{2} - h_{e_b} - h_{f_{ao}} - h_{e_s}$$

(11)

where the head loss per unit mass are defined as:

$$h_{e_b} = \frac{1}{2} \frac{u_{00}^2}{2}$$

$$h_{f_{ao}} = f_D \frac{b}{D_{fl}} \frac{u_{00}^2}{2}$$

$$h_{e_s} = K_{i,b} \frac{u_{1,b}^2}{2}$$

The loss coefficient, $K_{i,b}$, at the tee-intersection where the burner is located is very difficult to estimate and can only be determined accurately through laboratory tests. The value of $K_{i,b}$ depends on the geometric shape and location of the burner and the variation of $w$ and $h$. Appropriate laboratory tests have not been performed and thus the coefficient was determined partly based on experiments and partly based on handbook formulas. The value of $K_{i,b}$ was assumed to consist
of an experimentally determined constant, $K_0$, multiplied with a function $f\left(\frac{w}{h}\right)$ describing the influence of the cross-sectional ratio, $\frac{w}{h}$. Fried and Idelchik 19, present a relationship for a loss coefficient in a merging flow situation of a symmetrical (equilateral) wye with a sharp 90° turn without partition. Using this relationship and a volume flow ratio of 1:2, the following relationship is obtained for $K_{lb}$:

$$K_{lb} = K_0 \left(1 + \frac{1}{4}\left(\frac{w}{h}\right)^2\right)$$  \hspace{1cm} (12)

$K_0$ in eqn (12) was experimentally determined and the best fit for all the cases was found to be 6.5.

The friction factor $f_D$ in eqns (6), (10) and (11) was calculated from the following expression:

$$k_{non-c}$$

(13)

where $k_{non-c}$ is a correction factor due to the use of other cross sections than circular. A value of 1.08, see Fried and Idelchik 19, was found to be the average value valid within the experimental range. The other part of eqn (13) i.e., $0.3164\, \text{Re}^{-0.25}$, is the friction factor for a smooth circular tube in a Reynolds number range of 4000 to $10^5$. Due to a lack of exact knowledge about the surface roughness, the surface was assumed to be smooth. The Reynolds number was determined according to the following equation:

$$f_D = f_{smooth}$$

(14)

The denominator of eqn (14) is a regression formula for kinematic viscosity based on tabulated data from Ref. 9. Determination of the friction factor usually requires that the film temperature (in our case only in the vertical flue) is used for calculations of the Reynolds number. The film temperature is the average of the wall and the gas temperature. As the wall temperature is not known the calculated gas temperature was used. Calculations using wall temperatures varying from ambient to the gas temperature show that this assumption does not affect the results significantly. Thus, for simplicity, the computer program uses the
calculated gas temperatures, $T_{i,a}$, in the vertical flues instead of the film temperature.

When the program has run once through all tiers, a new calculation round is started by calculating the pressure at the lowest tier again (see Fig. 3). The program applies the calculated mass flow rates from the previous round. Iterations for individual tiers are repeated until an overall pressure balance is obtained. At this stage all mass flow rates, temperatures and velocities in the flues have been calculated.

**The flame height**

A simple model is proposed to calculate the flame height in the vertical flue. This is necessary in order to determine the fractions of the total convective heat release rate at the top and the base of each box, respectively. The flame height is determined by simply comparing the amount of available air and fuel at each tier. It is assumed that the flame tip will extend to a height where the total flux of air entrained is sufficient to complete the combustion reactions. For open buoyancy-controlled diffusion flames Heskestad \(^1^1\) assumed the air demand from the surroundings was proportional to the stoichiometric requirements of the pyrolysis gases, i.e.,

\[ \text{(15)} \]

\[
\text{where } m_e \text{ is the total mass-entrainment rate of air below the level of the flame tip, } \\
m_f \text{ is the fuel flow rate and } r \text{ is the stoichiometric mass air-to-fuel ratio. The proportionality constant is greater than unity because much of the air entrained in the vertical flues never takes part in the combustion reactions. The following equation defines the air-to-fuel stoichiometric fraction }^{12,13} : \\
\text{(16)} \]

\[
\text{where fuel flow rate is determined using: } \\
m_f = \frac{Q}{\Delta H_c} \hspace{1cm} \text{(17)}
\]

Q is the total heat release rate from the burner and $\Delta H_c$ is the heat of combustion per unit mass of the fuel. For open diffusion flames, as in the case of natural fires, complete combustion usually cannot be achieved at $\phi=1$ \(^1^3\). For turbulent diffusion flames in rack storage the value of $f$ has been empirically determined in the first phase of the project, as described in Ref. 1. From regression curves on the
experimental data the value for $f$ was found to be 6 at the flame tip for $w=50$ mm, 7.6 for $w=75$ mm and 10 for $w=100$ mm. The value of $f$ was determined from the experimental data $^{1,5}$ by using eqns (16) and (17). Regression of all data points yield $\phi=7.5$ at the flame tip $^{1,5}$. According to Delichatsios $^{14}$, $\phi$ can be as much as 10 for free-burning buoyant turbulent diffusion flames (based on measurements presented in Refs. 15,16) and Heskestad $^{20}$ has found this value to be equal to 12. We assume, as for open flames $^{11,14}$, that there is a direct relationship between entrained air and flame heights.

In the model presented here there are two steps taken to determine the flame height. First we determine to which tier (flame height interval) the flames ($L_f$) will extend by comparing the calculated mass flow rates of air and the fuel flow rates. For example if $m_f$ is greater than the mass flow rate $m_1$ divided by $r\phi$ but less than the mass flow rate $m_2$ divided by $r\phi$, the flame height will be greater that $(1.5h+s)$ and less than $(2.5h+2s)$. The next step is to determine how far the flames will extend within this interval. We simply assume that the flames will extend a distance that is proportional to the ratio of available fuel and air. This assumption can be justified as it was found that the ratio $Q/w$ tended to be linearly proportional to the flame height, $L_f$ $^{1,5}$. Thus, for the $i$th tier the following equation is used to calculate the flame extension within this interval:

$$\text{(18)}$$

The flame height is not calculated above the rack storage. Should it extend beyond the rack storage then it is simply put equal to the rack storage height.

**Distribution of the convective heat release rate**

It is known from measurements of free-burning diffusion flames $^8$ that the fraction of the total heat released increases with the flame height. Thus, we assume that close to the burner no energy has been released and at the flame top all energy has been released. The distribution of the convective heat energy between the flame tip and the burner needs to be determined from experiments.

This has been done by calculating the experimental convective heat release rate $Q_{c,exp}$ with the aid of the measurements presented in Ref. 1. Assuming $C_p=1000$ J/kg °C, the $Q_{c,exp}$ was determined from eqns. (2).
Figure 4. A plot of $Q_{c, exp}/Q$ versus $z/L_f$ is shown where $Q_{c, exp}$ is the experimental value at the height $z$, $Q$ is the total heat release rate from the burner and $L_f$ is the observed flame height (averaged value of the fluctuating visible flame).

The mass flow rate was calculated on the basis of a single point measurement at the flue centreline (the only experimental determination available) and not on the basis of an integrated value over the cross section (averaged value) which would have been better. The single measured value was used and the temperature and velocity were assumed to be uniform over the cross section. In Fig. 4, a plot of $z/L_f$ versus $Q_{c, exp}/Q$ is shown, where $Q_{c, exp}$ is the calculated value at the height $z$, $Q$ is the total heat release rate from the burner and $L_f$ is the observed flame height $^1$ (i.e. the averaged value of the fluctuating visible flame tip height).

One could expect the ratio $Q_{c, exp}/Q$ at the flame tip ($z/L_f=1$) to be close to 0.7 (see assumption (1)). In Fig. 4 it can be observed that this is not the case. One possible reason is that the vertical flow profiles (temperature and velocity) varies both with the flue width and the distance from the burner i.e., the flow profiles are not similar of form at all widths and heights. This may affect the calculated convective heat release rate $Q_{c, exp}$. There may as well be some three dimensional effects along the length of the rack which are not considered in the calculation of $Q_{c, exp}$. Simulations with Computational Fluid Dynamics (CFD) would be an appropriate method to investigate these effects.

The shape of the dependence of $Q_{c, exp}/Q$ on $z/L_f$ was determined based on the empirical results by setting $Q_{c, exp}/Q =1$ when $z/L_f=1$. This causes only a vertical adjustment of the relationship:
Thus, the $Q_c$ at the base and the top of each box is determined in accordance to the following equation:

$$Q_c = 0.7Qr(z/L_f)$$  \hspace{1cm} (20)$$

where $0.7Q$ is assumed to be the total convective heat release rate from the burner and $r(z/L_f)$ is obtained from eqn (19). When $z/L_f<0.1285$ we assume $r(z/L_f)=0$.

**EXPERIMENTAL SET-UP**

A brief description of the experimental set-up is given here. A more thorough description can be found in Ref. \(1\). The following combinations of parameters were used:

- $w=50$ mm, $75$ mm, $100$ mm with $h=50$ mm
- $w=50$ mm with $h=75$ mm
- $w=50$ mm with $h=100$ mm

where $w$ is the width of the vertical flue and $h$ the height of the horizontal flue.

The following four heat release rates were used for each of the combinations above:

- $Q=18.84$ kW, $24.8$ kW, $34.7$ kW and $44.5$ kW

The rack storage consisted of rectangular Navilite N boxes ($l=0.59$ m, $s=0.235$ m and $b=0.22$ m) held up by two narrow steel columns. The rack storage was two boxes wide and four boxes high (see Fig. 1). Walls were put at each end of the boxes to create two dimensional conditions. Specially made wings were mounted at each wall in order to minimise the boundary effects of the air flowing in through the horizontal flues (see the bird's view in Fig. 1). The Navilite N plates used to build the boxes were $9.5$ mm thick with the following thermal data \(17\):

- heat conductivity $0.17$ W/m $\degree$C, specific heat $950$ J/kg $\degree$C and density $810$ kg/m$^3$.

The length of the gas burner was $590$ mm and other dimensions of the gas burner are given in Ref. \(1\) and \(5\).

Prior to the tests, the line burner was calibrated for the heat outputs used in the test series. During the test series the different heat outputs were manually adjusted using a rotameter.

The instrumentation consisted of type K thermocouples (0.25 mm) and bi-directional probes (16 mm in diameter) located along the centreline in the vertical flue.

The relationship between pressure and velocity includes corrections for variation in the Reynolds number according to calibration curves reported by McCaffrey and Heskestad \(18\). The results were averaged over the time interval during which the measured gas temperatures and velocities were reasonably steady. It was found that this time was about four minutes, i.e. the last four of the six minutes of the test. In the beginning of a test, the walls were relatively cold,
causing large heat losses. After about 2-3 minutes the gas temperature in the vertical flue was found to increase moderately until the end of the test. The measured gas temperatures were corrected due to radiation effects.

RESULTS AND DISCUSSION

In the following presentation, calculations are compared to experimental results. First, calculations of mass flow rates at different tiers and for different flue widths are compared to experimental results. Further, experimental flame height, gas temperature and velocity measurements are compared to calculated values.

In the presented calculation results, considerations have been given to possible influences of high aspect ratios ($l/w$) on the $K_{li}$ values ($K_{li} = 1.22 - 1.027(V_i / V_{i-1})^{0.15}$). The aspect ratios are found to be in the range of 6 to 12 in our case. In Fried and Idelchik the influence of high aspect ratios are shown for rectangular elbows (p. 168-169). The loss coefficient will decrease by about 25-30% for aspect ratios 5-8 compared to an aspect ratio equal to one. If we assume that a similar relationship exists for converging flows at an intersection we can reduce the calculated aspect ratios by a certain value. It is found that 25% reduction yields reasonably good agreement with the experimental data.

In general it is found that the calculated mass flow rates agrees reasonably well with the experimental data except at the highest tier where the mass flow rate is notably lower than the experimental data (see Fig. 5 to 7). Further, Fig. 5 and 6 show that the model correctly predicts the moderate influences of the horizontal flue height on the vertical flue flow.

It is interesting to observe that the mass flow rate at each tier tends to be nearly constant for heat outputs greater than 15-20 kW. This is the same results as was obtained theoretically in Ref. 1 and 5 for a single stack, and is clearly valid for the individual tiers as well.

A comparison between calculated and observed flame heights is shown in Fig. 8. The agreement is reasonably good for the lower tiers but not for higher tiers. Further, the slope of the theoretically obtained curve differs considerably from that of the experimental curve, especially for the higher tiers. The slope of the curve may change slightly if the mass flow rates are increased at the higher tiers. Using a constant value of $\phi$ may affect the results as well. An average value of 7.5 for all three flue widths was used in the calculations whereas $\phi$ was found to vary from 6 - 10 depending on the flue width. The results can be improved by using different $\phi$. 


Figure 5. Comparison of calculated and measured mass flow rates at different tiers for \textit{w}=50 \textit{mm} and \textit{h}=50 \textit{mm}. Observe that calculated mass flow rates are only shown for those cases where the calculated flame height is lower than the height of the rack storage.

Figure 6. Comparison of calculated and measured mass flow rates at different tiers for \textit{w}=50 \textit{mm} and \textit{h}=100 \textit{mm}.
Figure 7. Comparison between calculated and measured mass flow rates at different tiers for \(w=100\) mm and \(h=50\) mm.

Figure 8. Comparison between observed and calculated flame heights. The agreement is less good at higher tiers. The mass flow rates of air at the higher tiers are underestimated and consequently the flame heights become higher.
Figure 9. Comparison between measured centreline temperature and the calculated (average) temperature.

Figure 10. Comparison between measured centreline velocity at the second, third and fourth tiers and corresponding calculated (average) velocity.
In Fig. 9 the calculated average temperatures are compared to the measured centreline temperatures where the temperature is plotted as a function of $z/(Q/w)$. This correlation can be justified as it was found that the ratio $Q/w$ tended to be linearly proportional to the flame height, $L_f$. As can be observed in Fig. 9 calculated temperature is found to be slightly less than the measured. The reader should have in mind that the calculated temperatures are average values whereas the measured are maximum values. In Fig. 10 the calculated velocities (average) at the second, third and the fourth tiers are compared to the corresponding measured centreline velocities. In Fig. 11 the calculated velocity (average) at the first tier is compared to the corresponding measured centreline velocity. The agreement is reasonably good in both cases.

**CONCLUSIONS**

A theoretical model to predict mass flow rate of air, temperature, velocity and flame height in a two dimensional rack storage was developed. It was necessary to introduce generalised empirical corrections of loss coefficients, stoichiometric requirements, and distribution of heat release in the model. Comparison of calculated and measured values show that the dominating flow mechanisms are accounted for in the model.

The model shows that the air flows into the horizontal flues due to the pressure difference between the inside and outside of the rack storage. According to the results the leakage, if any, out through the horizontal flues is probably very small. It is found that about one third to one fourth of the air enters through the lowest flue. The model predicts very well what was observed in Ref. 1, i.e. that the height of the horizontal flues have insignificant effects on the flow inside the rack storage, whereas the width of the vertical flue dominates the flow.
The model is found to predict the flame heights poorly in the upper part of the stack. The flame height model needs to be improved. The mass flow rates of air at higher levels can be increased, and the slope of the curve reduced by adding heat losses to the model. Furthermore, using different $\phi$’s will also improve the results as the flame height model is found to be very sensitive to the value of $\phi$.

The most important parameters for calculation of sprinkler response are the gas temperature and gas velocity. Therefore it is of importance that the model predicts these parameters well. The results presented in this study show that the calculated temperatures and velocities at all heights in the rack storage agrees reasonably well with the experimental data.

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Paper V

CFD Simulation of Fires in Two Dimensional Rack Storage

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CFD SIMULATION OF FIRES IN TWO DIMENSIONAL RACK STORAGE

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ABSTRACT

Two dimensional Computational Fluid Dynamics (CFD) calculations have been carried out to simulate a fire in a rack storage. Parameters varied in the simulations were the heat release rate and both vertical and horizontal flue dimensions. Heat release rates of 18.8 and 34.7 kW were used in combination with vertical and horizontal flue widths of 50, 75, and 100 mm. The calculation results were compared with experimental measurements. This work show that CFD simulations are useful for fire problems in rack storage configurations.

INTRODUCTION

Upward propagation of fire in a rack storage is very rapid. Manual fire fighting with hose streams alone has been found to be an extremely doubtful method to bring fires in rack storage under control. Fire protection in rack storage must be automatic in-rack sprinklers placed at several levels in the storage. As it is very important to know the activation time of in-rack sprinklers, a knowledge of the gas temperature and flow velocity field at the sprinkler location is necessary.

The present work is a first step in the validation of CFD models of fire tests in a rack storage configuration. A two dimensional rack storage was considered in this study. The aim of such tests is to find the effects of the flue dimensions on entrainment mechanisms for in-rack fires, and the resulting flame spread and operating times of sprinklers in the rack. These problems are not completely solved today. The simulations validated in this paper are performed on one third scale of real rack storage. The results are, however, expected to be transferable to full scale.

EXPERIMENTS

The simulations in the present study were validations of CFD code to model two dimensional rack storage tests performed at Fire Technology department at the Swedish National Testing and Research Institute. A brief description of the experimental set-up is given here while a more thorough description is given by Ingason.

The rack storage consisted of rectangular Navilite N boxes, 0.59 m long, 0.235 m high and 0.22 wide, held up by two narrow steel columns. Walls were put at each end of the boxes to create two dimensional conditions. Specially made wings were mounted at each wall in order to minimise the boundary effects of the air flowing in through the horizontal flues, see the plan view in Fig. 1. The Navilite N plates used to build the boxes were 9.5 mm thick the thermal data for Navilite N being: heat conductivity 0.12 W/m °C, specific heat 800 J/kg °C and density 700-780 kg/m³.
SIMULATIONS

The simulations have been calculated with the CFD code, JASMINE, from the Fire Research Station (FRS)\(^5\). The JASMINE code has been successfully used in studying fire problems for a wide range of different geometries, such as road tunnels\(^6-10\), high rise buildings\(^11\), warehouses\(^12\), hospital bed rooms, corridors, ceiling jets\(^13,17\) etc. Version 1.1 (1988) was used, which is based on CHAM's (UK) PHOENICS code (1983). Originally, JASMINE was designed to analyse smoke migration in fire situations. It is based on full solutions of the fundamental laws of conservation of mass, energy, momentum and chemical species. The equations are of the form:

\[
[\text{accumulation}] = [\text{diffusion}] + [\text{convection}] + [\text{sources}]
\]

The standard k-e model, modified by a buoyancy production term which arises naturally in the derivation of the equation for turbulent kinetic energy, is used for turbulence calculations. The input to the JASMINE code consists of any number of parameter specifications including the specification of boundary conditions, fire source, heat source, ventilation and openings depending on the system under study. All boundary conditions are inserted as source or sink terms of one or more variables, which are represented in linear expressions\(^5\) of the type:

\[
S_\phi = C_\phi (V_\phi - \phi)
\]  

where \(\phi\) is the local grid node value of the variable, and \(C_\phi\) and \(V_\phi\) are the 'coefficient' and the 'value', respectively. The different boundary types used in the program are: walls, mass inflow (or mass outflow), fixed pressure boundaries and fixed flux boundaries. Adiabatic or non-adiabatic boundaries can be chosen.

JASMINE uses a special type of wall function for evaluation of the influence of wall properties, such as surface roughness and thermal properties. Wall roughness values can readily be selected from among several standard values or they can be defined by the user. The fire source is defined by the fire area, the heat release rate or fuel mass release rate. There are many combustion models in the code which can be selected, for example,
the Laminar Flamelet model and Magnussen's Eddy Dissipation Concept (EDC). The latter model was used in the present study. Heat losses from gases to the wall boundaries were calculated assuming a bulk heat transfer coefficient which takes account of both radiative and convective effects.

The output from JASMINE consists of calculated values of number of parameters including temperature, species concentrations, pressure, velocity optical density, turbulent kinetic energy and its dissipation, as a function of time, in each control volume. At wall boundaries, the local values of wall temperature, and convective and radiative heat fluxes are calculated. The rates of mass and heat flows and radiative heat losses through openings are also calculated.

The boxes were assumed to be blockages. The sizes of the boxes are assumed unchanged during the fire, i.e. no material consumption of boxes occurs.

**COMPUTATIONAL MESH**

The calculation domain (in two dimensions) was 19 cells in the lateral (y) direction and 76 cells in the vertical (z) direction, respectively. Due to symmetry only half the rack storage was explicitly simulated. The storage configuration is shown in figure 1. To ensure the best numerical accuracy possible at locations where the field gradients (i.e. the temperature, velocity, turbulent kinetic energy, etc.) are expected to be high, the calculation space was divided into finer grid. In this case these regions are the space near the fire source and flues. In the vertical flue (i.e. half of the flue due to symmetry) the space is divided into 5 nodes and in each horizontal flue the space is divided into 10 nodes. This division gives the node-to-node distance of 0.5 to 1 cm, depending on the width of the flues in the given scenario, which was varied between 50 and 100 mm.

**CALCULATIONAL DETAILS**

CFD calculations were made using a transient mode of JASMINE. The time integration was performed in most cases in 3 min. Time step of 1 s was used throughout the calculations. At each time step 90 to 120 iterations were used. In some cases a time step of 0.5 s and a larger number of iterations were used to test the convergence.

The fire source was assumed to be a linear propane burner equal to the one used in the tests. Magnussen's Eddy Dissipation Concept (EDC) was used, in which the reaction rate is related to the slowest of the turbulence dissipation rates of either fuel or oxygen. This is an extension of Spalding's eddy-break-up model, in which the reaction rate was related to the fuel mass fraction alone. Simple, one-step reaction chemistry was considered; i.e.:

\[
C_3H_8 + 5O_2 + 18.8N_2 \rightarrow 3CO_2 + 4H_2O + 18.8N_2
\]

In the above reaction the propane is directly converted to combustion products from the stoichiometric requirement of oxygen reacting with fuel (nitrogen acting as a dilutent).

**RESULTS AND COMPARISONS WITH MEASUREMENTS**
Figures 2, 3 and 4 show comparison of the measured mass flow rates, temperatures and gas velocities at each tier, respectively. The point at each tier is at location corresponding to 2/3 of the box height of each box. Each mark in Fig. 2-4 corresponds to one of four different heights.

Figure 2. Comparison of calculated and measured mass flow rate at each tier. Explanation to legend: digits after Q state RHR in kW, digits after W and H state vertical and horizontal flue widths in mm, respectively. Each mark type includes four points representing calculated and measured points at four different heights. The solid line represents exact correlation between the calculations and experiments.

Figure 3. Comparison of calculated and measured gas temperatures at each tier (for legend explanation, see Fig. 2).
The calculated mass flows agree very well with measured result, as shown in Fig. 2. Generally, the discrepancy is within 25%.

As it can be seen in Fig. 3, the simulated gas temperatures are generally overestimated. For the scenarios with larger RHR the disagreement is very large; the simulated temperatures are more than twice those measured. For the scenarios with lower RHR, i.e. 18.8 kW, the agreement between calculated and measured result is reasonable. The differences are within 30% (overestimated). At lower parts of the plume regions, i.e. near the simulated fire source, the overestimation is largest. Generally, the increased flue widths also yield a larger overestimation of the calculated results.

The simulated vertical gas velocities (Fig 4) are also overestimated, as can be expected due to the overestimated high temperatures which causes faster upward buoyant flow. The general difference between calculated and measured gas velocities is that calculated gas velocities are almost constant with height, rather than increasing with height as they do according to experimental measurements. This is noticed in both the 18.8 and 34.7 kW scenarios.

CONCLUSIONS

The simulated values of gas temperatures, velocities and mass flow rates agree best with measured values in scenarios with the lowest RHR, i.e. 18.84 kW. In scenarios with RHR of 34.7, the calculated temperatures are much higher than measured ones. The reason for this discrepancy is the crude radiation model and that the simulated heat transfer into two upper boxes was neglected (due to the limitation of physical boundary conditions in this version of JASMINE). The very high simulated temperatures at lower levels in the plume can also be attributed to the crude radiation model used. This, together with neglected heat transfer to boxes, makes simulated flame lengths longer, which can explain very high temperatures at lower levels. One of the reason for the overestimated temperature is probably modelling of the fire source. It may be necessary to make further refinements (in these simulations the smallest cell lengths were 5 mm) of the local grid in the vicinity of fire source. The obstruction in the air flow to and from the
fire source and the small space to the fire may cause too great gradients in the
temperature and velocity, and hence in the pressure, which probably affects the solution.

The present results will serve as a basis for formulating a simplified theoretical model of
fires in general rack storage configurations.

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Paper VI

In-Rack Fire Plumes

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In-Rack Fire Plumes

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ABSTRACT

A theoretical and experimental study of in-rack fire plumes in a combustible rack storage is presented. The commodity used in the experiments consisted of multiple-wall corrugated paper cartons equally separated in a steel rack, two cartons wide, two cartons deep and four cartons high. Free-burn tests were carried out mostly in reduced scale with verification in full scale. The centreline in-rack gas temperatures and velocities were measured at four elevations inside the rack storage and the heat release rate was measured above the rack storage with a hood system. In-rack temperatures and velocities are plotted using theoretically obtained quasi-steady correlations assuming a point source of buoyancy at floor level and entrainment only in the vertical flues. The theoretical correlations include convective heat release rate, vertical flue width and height above the floor. Temperatures for both reduced scale and full scale correlate reasonably well with the theoretically obtained relationships which have the same functional form as convective plume flow above a linear fire source (line plume). In order to obtain similar correspondance for the velocity in the two scales it was necessary to use Froude number scaling with storage height as length scale. This does not comply with the results of the line plume model which indicates constant velocity independent of elevation height.

The storage arrangement is found to be important for the initial flame spread and fire growth rate. The reduced scale study shows that the initial fire growth rate decreases with increasing vertical gaps (flues) and that the vertical and lateral flame spread rate increase when the lateral flue height increases. For the initial in-rack fire growth rate, the convective heat release rate was found to be better described by an exponential function rather than a power-law dependence on time to the third power.

KEYWORDS: rack storage fire, convective heat release rate, in-rack plume flow, in-rack temperature, in-rack velocity

NOMENCLATURE

A fire plume area (m²)
Cₚ specific heat (kJ/kg K)
f momentum flux (kg m/s²)
g gravitational acceleration (m/s²)
ΔHₑ heat of combustion (kJ/g)
INTRODUCTION

Fire growth rate and flame propagation are dependent on how the burning material is stored. In rack storage fires this becomes extremely important as the upward flame propagation is usually very rapid. The vertical shafts, or flues, created between adjacent pallets of stored goods tend to work as chimneys and enhance acceleration of flames up to the ceiling. Rack storage fire protection usually consists of in-rack sprinklers which are placed at different elevations in the vertical flues or on the faces of the rack storage. The efficiency of such protection depends on the geometry of the stored goods, their height, the flue dimensions and intervals of occurrence as well as the flammability of the stored goods. The delay prior to in-rack sprinkler actuation is extremely important as this may be critical for control of the fire. To calculate the response time, a knowledge of the flow conditions at the sprinkler is necessary. A theoretical model which can predict these flow conditions is therefore of interest.

Previous work [1,2] on in-rack plume flow in a two dimensional (2D) non-combustible rack storage indicates that the width of the vertical flue is the governing geometric parameter for the in-rack plume flow. Further, variation of the horizontal flue heights was found to have negligible effects on the in-rack plume flow. No consideration was given to the problem of heat transfer from the flame to the surrounding walls which is of importance for flame spread studies. Foley and Drysdale [3], however, recently published a thorough study of this problem where they used a similar configuration. They showed that as the separation between the walls is reduced the heat flux to the walls increases.
Thomas [4] studied some large scale fire test performed in the United Kingdom and found some evidence that the width of the vertical flue affects speed of flame spread. He suggested there is a gap size with a highest spread rate but clearly three dimensional features of the flow are involved too. A number of large scale tests have been carried out by various fire laboratories for the purpose of studying the behaviour of sprinkler systems in controlling rack storage fires [5-11]. Factory Mutual Research Corporation (FMRC) [5-7] carried out a great number of full scale sprinklered and free burning tests demonstrating the rapid spread of flames when different commodities are stored in racks. Previous theoretical work [5-7] on rack storage fires have been directed to predict what occurs above the rack storage rather than what occurs inside it. Yu and Stavrianidis [5] presented an empirical fire growth rate correlation as a power-law dependence on time to the third power for 2,3,4 and 5 tier rack storage fires. The initial fire growth rate was found to be invariant with number of tiers. Variations or effects of different flue separations was not studied.

To date no systematic investigation of the influence of gap size on fire growth rate has been undertaken. A systematic variation in vertical and lateral gap sizes was carried out in the present study. Similarly no theoretical model has previously been available to predict the flow conditions in a three dimensional (3D) rack storage configuration. Such a model is presented and compared to the experimental results.

**THEORETICAL ASPECTS**

An exploratory model for a non-reacting turbulent in-rack plume is outlined here. In rack storage fires, air is entrained into the in-rack fire plume mainly through vertical flues but also through horizontal flues. However, to be able to find a simple theoretical solution we assume in the first approximation that no air is entrained through the horizontal flues. Thus, the rack storage configuration can be simplified from figure 1 a) to the configuration shown in figures 1b) and 1c). The following assumptions have been made:

- plume buoyancy stems from a point;
- the air entrainment velocity at the edge of the in-rack plume is proportional to the local vertical velocity, $u$;
- no air is assumed to be entrained through the horizontal flues;
- the shape of the in-rack plume cross-section is assumed to be the same at all heights, see figures 1b) and 1c);
- $u$ and $\rho$ have top hat profiles;
- no horizontal pressure variation exists between the in-rack plume and the surrounding air;
- the ambient air is of uniform temperature;
- the flue gases have the same molecular weight as air;
- no heat losses occur to the surrounding walls; and
- the drag exerted by the cartons is neglected.

\[ \frac{d}{dz} (\rho uA) = \rho_0 \alpha u \frac{A}{w} \]  \hspace{1cm}  (1)

\[ \frac{d}{dz} (\rho u^2 A) = (\rho_0 - \rho) g A \]  \hspace{1cm}  (2)

\[ Q_c = \rho u A C_p (T - T_0) \]  \hspace{1cm}  (3)

In these equations, \( z \) is the height above the floor, \( A \) is the mean in-rack plume area, \( \alpha \) is the entrainment coefficient at the edge of the in-rack plume, \( w \) is the vertical flue width, \( u \) is the mean in-rack velocity, \( \rho \) is the mean in-rack density (essentially air), \( C_p \) is the specific heat of the in-rack plume and \( Q_c \) is the instantaneous convective heat release rate.

**FIGURE 1.** a) Illustration of a 3D rack storage  b) side view of the simplified model c) birds-eye view of the simplified model.
An interesting observation is that these equations are similar to those for fires with a linear buoyancy source, see Lee and Emmons [12]. To solve eqns (1)-(3) we introduce the mass flux: \( m = \rho uA \) and the momentum flux: \( f = \rho u^2 A \) as two new variables. Substituting these new variables into eqns (1)-(3) yields:

\[
\frac{dm}{dz} = 4\alpha \rho_0 \frac{f}{m} \quad (4)
\]

\[
\frac{df}{dz} = g \frac{(\rho_0 - \rho)}{\rho} m^2 f \quad (5)
\]

\[
Q_c = mC_pT_0 \frac{(\rho_0 - \rho)}{\rho} \quad (6)
\]

Rearrangement and integration of these equations yields the desired relationship:

\[
m^3 = \frac{4\alpha \rho_0 C_p T_0 w}{g} \frac{Q_c}{Q_c} f^3 + C \quad (7)
\]

In order to obtain a simplified functional relationship for further analysis the integration constant \( C \) has to be equal to zero. We substitute \( m = \rho uA \) and \( f = \rho u^2 A \) into eqn (7) which gives the in-rack velocity:

\[
u = \left( \frac{g}{4\alpha \rho_0 C_p T_0} \right)^{1/3} \left( \frac{Q_c}{w} \right)^{1/3} \quad (8)
\]

Integration of eqn (4) up to the height from the floor level, \( z \), gives the mass flux as a function of height:

\[
m = (4\alpha \rho_0)^{2/3} \left( \frac{g}{C_p T_0} \right)^{1/3} (w^2 Q_c)^{1/3} z \quad (9)
\]

By substituting eqn (9) into eqn (6) and making use of the ideal gas law relationship \( \frac{T - T_0}{T_0} = \frac{(\rho_0 - \rho)}{\rho} \) we obtain:

\[
T - T_0 = \Delta T = \left( \frac{T_0}{4\alpha \rho_0 C_p} \right)^{1/3} \left( \frac{Q_c}{w} \right)^{2/3} \frac{1}{z} \quad (10)
\]

The functional relationship for the in-rack velocity and in-rack temperature can now be used to plot the experimental data.

**EXPERIMENTS**
A series of model scale tests was carried out followed by one full scale test. The fuel array consisted of four equally separated cartons at each tier. The commodity was piled up to four levels, or tiers, with equal heights of the horizontal flues between every consecutive tiers. Geometrically, the commodity used in the model scale was approximately 1/3 of the full scale commodity. Both commodities were made of combined corrugated paper cartons and paper sheets and differed only in type and size.

Model scale test

The experiments were carried out under a calorimeter consisting of an exhaust duct and a hood which is located above the fire [13]. The calorimeter had the capacity to measure up to about 2 MW. The hood size was 3 m x 3 m with the lowest point 2.5 m from floor. Both the convective and chemical heat release rates were measured. The convective heat release rate was obtained by measuring the temperature and mass flow rate in the exhaust duct and the chemical heat release rate by measuring the oxygen depletion at the same location. The commodity consisted of 3.7 mm thick single wall corrugated paper cartons folded inside with four layers of 6.5 mm thick single wall corrugated paper sheets on each side of the carton. The total thickness of each side was thus 29.7 mm. The sheets and the paper carton were mounted on a supporting rectangular box made of 9 mm thick rigid Navilite insulation boards. The outer dimensions of each carton were 0.39 m x 0.39 m with a height of 0.305 m and the moisture content was 9 % by weight. A total of 16 cartons were used in each test. Four separated vertical steel rods were used to support the cartons. On each steel rod four cartons were centrally mounted and secured in position. Cone Calorimeter [13] results of the material used are presented in [14]. The average value of the effective heat of combustion $\Delta H_c$ was 13.35 MJ/kg. For comparison, Tewarson [15] has presented a value of $\Delta H_c = 14.2$ MJ/kg for corrugated paper cartons arranged in three dimensional, one or two tier rack storage. The time to ignition was measured in the cone calorimeter at irradiance of 25 kW/m$^2$ and 35 kW/m$^2$. The measured ignition times were 20 seconds and 9 seconds, respectively.

The vertical flue width, $w$, was varied as follows: $w=50$ mm, 75 mm and 100 mm. For $w=50$ mm the height, $h$, of the horizontal flue was varied as follows: $h=50$ mm, 75 mm and 100 mm. For $w=75$ mm and 100 mm $h$ was kept constant at 50 mm. The total stack height $H$ varied depending on $h$ : $H=1.42$ m for $h=50$ mm, 1.52 m for $h=75$ m and 1.62 m for $h=100$ mm. One test was carried out by blocking the lowest horizontal flue ($w=50$ mm and $h=50$ mm).

At the lowest tier, four ignition sources were symmetrically mounted at the bottom of each carton as close as possible to centre flue space of the fuel array. The ignition source consisted of a 12 mm thick insulating fibre board measuring 17 mm x 17 mm, soaked with 2.8 ml heptane and wrapped in a polyethylene bag. The mean flame height of the ignition source was about 0.25 m measured from the bottom of the carton. This height corresponds to 83 % of the carton height. One test was carried out with a 24 mm thick ignition source measuring 25 mm x 30 mm, soaked with 12 ml heptane ($w=50$ mm and $h=50$ mm).
Centreline in-rack temperatures and velocities were measured at four elevations: for $h=50$ mm measurements were made at $z=0.25$ m, 0.61 m, 0.96 m and 1.32 m; for $h=75$ mm: $z=0.28$ m, 0.66 m, 1.04 m and 1.42 m; and for $h=100$ mm: $z=0.3$ m, 0.71 m, 1.11 m and 1.52 m. Each of these positions corresponds to 2/3 of the height of the carton. The velocity probes were mounted in a staggered form (at the centreline and $\pm 45$ mm from the centreline) to avoid interference in these measurements. The ambient temperature varied between 13 - 15 °C during the test series.

The full scale test

One full scale experiment was carried out under the Industry Calorimeter [17,18] in SP’s Fire Hall. The calorimeter, which is of same type as the FMRC Fire Products Collector [19], can measure up to 10 MW. Both the convective and the chemical heat release rates were measured. A Standard Class II commodity was used which consists of double triwall corrugated paper cartons (each 12 mm thick). The double cartons were folded onto a sheet-metal liner and then placed onto a wood pallet. The outer dimensions of each carton were 1.08 m x 1.08 m x 1.08 m and the moisture content was 11% by weight. A Cone calorimeter test was carried out with irradiance of 25 kW/m². The average value of $\Delta H_c$ was 11.5 MJ/kg and the time to ignition 34 seconds. Yu and Kung [6] determined the effective convective heat of combustion using Class II cartons in 4 tiers to be 6.1 MJ/kg. Using an average value of the measured ratio of convective heat release rate to chemical heat release rate during the growing fire period of the full scale test performed here: $Q_c/Q=0.517$ for $38<Q_c<2000$ kW, we find that $\Delta H_c$ for Class II cartons is 11.8 MJ/kg which is in remarkably good agreement with the measured cone calorimeter value of $\Delta H_c=11.5$ MJ/kg.

A double-row steel rack was used to hold the commodity. The width of the vertical flues was 150 mm and the height of the horizontal flues was 300 mm. The total height of the rack storage was 5.52 m. Four ignition sources were symmetrically placed as close as possible to the centre flue space of the fuel array, at the bottom of each carton at the lowest tier. The ignition source consisted of insulating fibre board (similar to cellucotton rolls), 75 mm in diameter and 75 mm long, each soaked with 120 ml heptane and wrapped in a polyethylene bag. The mean flame height of the ignition source was about 0.9 m measured from the bottom of the carton. This height corresponded to 83 % of the carton height. Centreline in-rack temperatures and velocities were measured at four elevations: $z=1.02$ m, 2.40 m, 3.78 m and 5.16 m. Each of these positions corresponds to 2/3 of the height of the carton. The ambient temperature was 11 °C during the test.

Instruments

The in-rack temperatures were measured with welded thermocouples of type K (Chromel-Alumel) with a wire diameter of 0.25 mm. The in-rack velocity was measured with bi-directional probes ($D=16$ mm and $L=32$ mm) [16] located at same elevation as the thermocouples. The thermocouples were attached to each bi-directional probes close to the sensor head. No correction due to radiation effects on the temperature measurements was carried out in this study. The velocity was corrected for variation in the Reynolds number according to
calibration curves reported in [16]. A more thorough description of the test set-up is given in [14]. The data was recorded by a data acquisition system every second in the full scale test and every 1.7 seconds in the small scale tests.

**DATA ANALYSIS AND RESULTS**

In this study, only the initial fire growth period is considered. This period is defined here as the time from ignition until the flames starts to spread upwards on the face of the rack storage. Following the procedure of Yu and Kung [6] for growing rack storage fires, a rolling time-averaging process was applied to all temperature, velocity and heat release rate measurements in order to smooth out fluctuations due to turbulence. The averaging period was 10 seconds. Thus each measurement was averaged using different numbers of data points before and after the point of interest. The time averaged results followed the trends of the unprocessed data very well.

All data points measured prior to the incipient time of fire growth, $t_0$, were disregarded in the data analysis and in plot of the data. The time $t_0$ was defined here as the time when the convective heat release rate started to increase notably in size. This time was obtained by investigating the measured convective heat release rates and by observing the initial flame heights from video recordings. The time $t_0$ agreed well with the time when the mean flame height of the ignition source just started to increase in size.

*Fire growth rate*

Yu and Stavrianidis [5] found that the convective heat release rate correlates with a power-law dependence on time to the third power for rack storage fires. This function worked well for the initial fire growth rate period, i.e., convective heat release rates up to 800 kW per tier which corresponds to 3200 kW for the 4 tier full scale test presented here. After studying a number of full scale tests, Thomas [4] described the fire growth rate of rack storage fires with line ignition by an exponential function, $Q \propto e^{3/2\tau}$, where $\tau$ equal to 10 - 15 seconds was found plausible.

In figure 2, comparison of the measured net convective heat release rate, $\Delta Q_c = Q_c - Q_{c,0}$, is made with heat release rate correlations given in ref. [4] and [5]. $Q_{c,0}$ is the convective heat release rate at $t_0$. The measured $\Delta Q_c$ is shown as a function of the time, $\Delta t = t - t_0$. The fire growth rate coefficient in Yu and Stavrianidis [5] correlation for 4 tier high rack storage is 0.179 and thus the net convective heat release rate $\Delta Q_c = 0.179\Delta t^3$. An exponential curve fit was made to the measured net convective heat release rate in the full scale test; $\Delta Q_c = 2.27e^{0.102\Delta t}$. This correlation follows the measured $\Delta Q_c$ remarkably well up to about 3000 kW. Yu's and Stavrianidis [5] correlation follows the slope of the measured $\Delta Q_c$ very well but the initial time appears to be quite different. This is probably due to differences in the determination of $t_0$. However, since we are primary interested in the in-rack fire plume flow in order to predict the activation time of in-rack sprinklers, we need a correlation which describes the fire growth rate from the time the flames start to accelerate upward from the ignition source. An interesting
observation is that \( \tau \) here is found to be \( 3/(2*0.102) = 14.7 \) seconds which is in good agreement with Thomas [4] prediction of \( \tau \) equal to 10-15 seconds.

The measured convective heat release rates, \( Q_c \), for both small scale and full scale tests are plotted in figure 3. Apparently there is a turning point in the fire growth rate at \( Q_c = 200 \text{ kW} \) in the model scale tests. For \( Q_c < 200 \text{ kW} \), increase of the vertical flue width \( w \) tends to delay the fire growth rate whereas variation of the horizontal flue height \( h \) appear to affect the fire growth rate slightly. However, for \( Q_c > 200 \text{ kW} \), smaller \( w \) tends to retard the fire growth rate whereas larger \( h \) tends to proceed with a similar fire growth rate. In general, a large \( w \) and a small \( h \) appears to give slower initial fire growth rates and flame spread rates.

\[
\dot{Q}_c = Q_c - Q_c,0 \quad \text{[kW]}
\]
\[
\dot{t} = t - t_0 \quad \text{[sec]}
\]

FIGURE 2. Comparison between measured net convective heat release rate and data fits using power law correlation [6] and an exponential correlation [4].

A test with \( w=50 \text{ mm} \) and \( h=50 \text{ mm} \) using large ignition source and a test with same dimensions using a blockage at the lowest horizontal flue, were also carried out. The large ignition source influenced the initial fire growth rate considerably (faster initial fire growth) but it did not affect the slope of the curve. A blockage of the lowest horizontal flue did not have any affects on the fire growth rate. To investigate the reproducibility, two replicates with \( w=75 \text{ mm} \) and \( h=50 \text{ mm} \) were performed. The results were found to be nearly identical.

Estimations of times from video recordings of flame spread are presented in Table 1. Apparently the incipient time, \( t_0 \), is directly related to variation in \( w \). Clearly \( h \) influence the average vertical luminous flame height speed, \( u_f \), considerably. Further, in figure 3, it was observed that \( Q_c \) was slightly affected by variation in \( h \). The average upward flame speed, \( u_f \), in the full scale test agree with the upward flame speed derived by Thomas [4], i.e., \( u_f = 0.1 \text{ m/s} \). Thomas [4] value is based
on a number of full scale tests. It is evident from the model scale tests, however, that \( u_f \) varies considerably with \( w \) and \( h \). There is some inconsistency in the data in Table 1. The behavior of the fire growth rate with \( h \) and the observed flame height appear inconsistent. Furthermore, Figure 3 shows that the fire with \( h=100 \) mm develops slightly slower than the fires with \( h=50 \) mm and 75 mm. Further analysis of the data is necessary.

![Figure 3. Convective heat release rate in the model scale tests (left y-axis) and the full scale test (right y-axis).](image)

**TABLE 1.** In-rack flame spread where \( t=\) time from ignition, \( t_0=\) incipient time of fire growth, \( t_f=\) time from ignition for average flame tip to reach the top of the storage, \( u_f=\) the mean in-rack flame speed, \( t_h=\) time from \( t_0 \) for horizontal flue flames to reach the face of the rack, \( t_E=\) time from \( t_0 \) where the outer face of the rack is engulfed with flames at tier two, three and four.

<table>
<thead>
<tr>
<th>Experiment</th>
<th>( t_0 ) (s)</th>
<th>( t_f ) (s)</th>
<th>( \Delta t_f= t_f- t_0 ) (s)</th>
<th>( u_f ) (m/s)</th>
<th>( t_h ) (s)</th>
<th>( t_E ) (s)</th>
<th>( \Delta t_E= t_E- t_h ) (s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>model ( w=50 ) mm, ( h=50 ) mm</td>
<td>13.0</td>
<td>30</td>
<td>17</td>
<td>0.084</td>
<td>29.0</td>
<td>77.0</td>
<td>48</td>
</tr>
<tr>
<td>model ( w=75 ) mm, ( h=50 ) mm</td>
<td>22.6</td>
<td>45</td>
<td>22.4</td>
<td>0.063</td>
<td>31.4</td>
<td>81.4</td>
<td>50</td>
</tr>
<tr>
<td>model ( w=100 ) mm, ( h=50 ) mm</td>
<td>26.3</td>
<td>55</td>
<td>28.7</td>
<td>0.050</td>
<td>33.7</td>
<td>93.7</td>
<td>60</td>
</tr>
<tr>
<td>model ( w=50 ) mm, ( h=75 ) mm</td>
<td>19.6</td>
<td>32</td>
<td>12.4</td>
<td>0.123</td>
<td>29.4</td>
<td>76.4</td>
<td>47</td>
</tr>
<tr>
<td>model ( w=50 ) mm, ( h=100 ) mm</td>
<td>17.4</td>
<td>30</td>
<td>12.6</td>
<td>0.129</td>
<td>22.6</td>
<td>53.6</td>
<td>31</td>
</tr>
<tr>
<td>full scale test ( w=150, \ h=300 )</td>
<td>40</td>
<td>95</td>
<td>55</td>
<td>0.100</td>
<td>45.0</td>
<td>95.0</td>
<td>50</td>
</tr>
</tbody>
</table>

As the flames spread upwards, they start to spread horizontally through the lateral flues. The time for the flames to reach to the face of the rack storage, \( t_h \), is given in Table 1. It is apparent that the lateral flame spread is dependent on variation in \( w \). It takes longer timer to reach to the face of the rack storage with increasing \( w \). However, an increase in \( h \) appears to shorten this time. The flames usually reached first to the rack storage face in the flue between tiers 3 and 4 in the model scale tests whereas for the full scale test this occurred between tiers 2 and 3.
Finally, times until the rack storage is entirely engulfed with flames on the outside are given. Apparently it too follows the trend of $t_h$.

**Centreline in-rack temperature**

To investigate the validity of the theoretical obtained correlations, the experimental data at tiers 2, 3 and 4 have been plotted on a log-log graph. The experimental data obtained at the lowest tier was disregarded here as it did not fit very well to the obtained correlation. The reason for this is probably that the combustion zone is initially at the lowest tier and the plume flow and $Q_c$ are not fully developed at this height. This behaviour was observed in the 2D rack storage presented in ref. [2]. Based on eqn (10) the in-rack temperature should correlate as:

$$\Delta T \propto \left( \frac{Q_c}{w} \right)^{2/3} \frac{1}{z}$$

(11)

In the log-log graph, the data fall into a single curve for both scales but there is some scatter in the data. By introducing a virtual source height, $z_0$, preliminary calculations show that a plot of $\Delta T$ as a function of $(z-z_0)^{3/2}/(Q_c/w)$ in a log-log graph will fall onto a line with a slope of $-2/3$. The preliminary calculations indicate that $z_0$ is usually above the floor level and increase in height with increasing $Q_c$.

**Centreline in-rack velocity**

Based on eqn (8) the measured in-rack velocity at tiers 2, 3 and 4 has been plotted according to the following relationship:

$$u \propto \left( \frac{Q_c}{w} \right)^{1/3}$$

(12)

In order to obtain similar correspondence for the velocity in the two scales it was necessary to use Froude number scaling with storage height, $H$, as a length scale. This does not comply with the results of a line plume model which indicates constant velocity independent of height $z$. Thus, there appears to be a height dependence of the velocity which in turn indicate that there may be other sets of correlations which could be used to plot the data (e.g. axisymmetric correlations).
FIGURE 4. Excess in-rack temperature plotted as a function of \((Q_c/w)^{2/3}/z\) at tiers 2, 3 and 4.

Here we define the Froude number as:

\[
Fr = \left( \frac{u^2 T_0}{gH\Delta T} \right)
\]  

(13)

Using eqns (11) - (13) we obtain:

\[
\frac{u}{\sqrt{H}} \propto \left( \frac{Q_c}{w} \right)^{1/3}
\]  

(14)

In the log-log graph, the data fall into a single curve for both scales but the scatter in the data is considerable. The experimental data obtained at the first tier did not fit to the obtained correlation.
CONCLUSIONS

A theoretical and experimental study of in-rack fire plumes in a combustible rack storage is presented. Free-burn tests were carried out mostly in reduced scale with verification in full scale. The centreline in-rack gas temperatures and velocities were measured at four elevations inside the rack storage. In-rack temperatures and velocities are plotted using theoretically obtained quasi-steady correlations assuming a point source of buoyancy at floor level and entrainment only in the vertical flues. The theoretical correlations include convective heat release rate, vertical flue width and height above floor. Temperatures and velocities for both reduced scale and full scale correlate reasonably well with the theoretically obtained relationships which have the same functional form as convective plume flow above a linear fire source. In order to obtain similar correspondence for the velocity in the two scales it was necessary to use Froude number scaling with storage height as length scale. This does not comply with the results of a linear plume model which indicates constant velocity independent of elevation height.

For the initial in-rack fire growth rate, the convective heat release rate was found to be better described by an exponential rather than power-law dependence on time to third power.

The reduced scale study shows that the initial fire growth rate is decreased when vertical gaps (flues) increase in size. Furthermore, for a given vertical gap size both the vertical and lateral flame spread rate increase when the lateral flue height increases.
Despite the complexity of rack storage fires, a way to provide necessary flow data to calculate response times of in-rack sprinklers is demonstrated. The model can be easily extended to include flame heights correlations as well.

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Paper VII

Flame Heat Transfer in Storage Geometries
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Flame Heat Transfer in Storage Geometries

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Abstract

The paper presents measurements of the heat flux distribution to the surface of four square towers exposed to buoyant turbulent flames. The steel towers represent an idealisation of a rack storage configuration at reduced scale. Each tower was 1.8 m high and 0.3 m x 0.3 m wide. The fuel was supplied from a circular gas burner at the floor. Three different gaseous fuels were used: carbon monoxide (CO), propane (C₃H₈), and propylene (C₃H₆). These fuels cover a wide range of flame sootiness resulting in distinctly different flame heat fluxes. At the same overall heat release rates the peak heat fluxes from C₃H₈ flames were twice those from CO flames, whereas the peak heat fluxes from C₃H₆ flames were 2.8 times those from CO flames. Heat fluxes were measured by thermocouples spot-welded to the back of the steel sheets. They were measured at 51 different locations. This measurement method turns out to be simple, accurate and robust in addition to being inexpensive.

Formulas are provided for the flame heat flux distribution in terms of the overall fire heat release rate, fuel sootiness and separation distance between the towers. The formulas are suitable for direct use by engineering models of fire growth in storage geometries. The paper also provides additional data needed for the development of more general CFD models capable of predicting fire growth of other geometries.

Notation

\[ b \] characteristic flame width (m)
\[ b_{ST} \] thermal plume width (m)
\[ C_p \] specific heat (kJ/kg K)
\[ D \] burner diameter (m)
\[ E \] heat released per O₂ consumed (kJ/kg)
\[ g \] acceleration of gravity (m/s²)
\[ \Delta H_c \] heat of combustion per unit mass of the fuel (MJ/kg)
\[ k \] thermal conduction (w/m K)
\[ l_s \] smoke-point flame-height (mm)
\[ L_f \] flame height (m)
\[ m_f \] fuel flow rate (kg/s)
\( m_{ex} \) mass flow rate in the exhaust duct (kg/s)
\( M \) molecular weight (kg/kmol)
\( q'' \) heat flux to tower wall (kW/m²)
\( Q \) chemical heat release rate (kW)
\( Q_c \) convective heat release rate (kW)
\( Re \) Reynolds number
\( S \) sootiness (defined by equation (5))
\( t \) time (s)
\( T \) temperature (K)
\( T_a \) ambient temperature (K)
\( T_0 \) average gas temperature inside enclosure before ignition (K)
\( \Delta T \) temperature rise above ambient (K)
\( u \) in-rack velocity (m/s)
\( u_a \) velocity at floor (m/s)
\( w \) width of a vertical flue (m)
\( x \) horizontal distance (m)
\( z \) height from floor level (m)

**Greek symbols**
\( \rho_o \) density at ambient temperature (kg/m³)
\( \rho \) density within the rack storage (kg/m³)
\( \alpha \) expansion factor
\( \phi \) oxygen depletion factor defined by equation (3)
\( \delta \) thickness of steel sheet (m)

**Introduction**

In recent years there have been numerous studies of fire spread up vertical surfaces of combustible materials [1-6]. The physics of the burning of single walls is reasonably well understood, particularly for situations in which the flow is constrained by side-walls to remain two-dimensional. Considerably less is known about fire growth in more general three-dimensional situations. Warehouse geometries are a particularly important example. Warehouses typically contain huge quantities of goods which are stored to great heights in racks designed for easy access by personnel. Unfortunately such storage arrangements also maximise the fuel surface area accessible to flames. Fires can rapidly spread up between opposing fuel surfaces. Heat from flames burning on one surface augments the heat transfer from the flames burning on the opposing surface. It all adds up to the rack-storage geometry being perhaps the most hazardous of all fire geometries.
Presently there are no data available for flame heat fluxes in three-dimensional rack-storage geometries. The most recent study on heat fluxes was carried out by Foley and Drysdale [7] using a propane flame between two parallel walls. They found decreasing heat flux with increasing separation between the parallel walls. The heat fluxes were quite sensitive to how the air entered near the base of the burner. Other studies [8,9] of parallel wall flames have shown that the separation distance between vertical surfaces to be important, whilst the size and positioning of the flues between horizontal surfaces are unimportant.

The present work provides measurements of the distribution of heat flux to the surface of four equally separated rectangular towers exposed to buoyant flames issuing from a circular gas burner at the floor. Three gaseous fuels were used: carbon monoxide (CO), propane (C$_3$H$_8$), and propylene (C$_3$H$_6$). These fuels were chosen to cover a wide range of sooting tendencies resulting in distinctly different flame heat fluxes. The differences are surprisingly large. For the same overall fire heat release rates the peak heat fluxes from C$_3$H$_8$ flames were twice those from CO flames, whereas the peak heat fluxes from C$_3$H$_6$ flames were 2.8 times those from CO flames. Researchers have often not appreciated the importance of fuel type on flame heat transfer.

Flame heat fluxes are usually measured by Gardon or Schmidt-Boelter heat flux gages. Numerous gages would have been required for obtaining reasonably accurate heat flux distributions. The problem was solved by thermally insulating the back of the steel sheets and then spot-welding thermocouples to the rear surface. This was done at 51 locations. The heat transfer is determined from the rate of rise of the steel temperature. The method is simple, reliable and accurate as well as being much less expensive than traditional gages.

The important questions are how: (a) the flame dimensions and (b) wall heat fluxes depend on the: (1) fire heat release rate, (2) fuel type and (3) wall separation distance. Experimental data are needed to guide the development of models capable of predicting fire growth rates in terms of tabulated properties of combustible materials. The present study addresses these questions by providing formulas for the flame heat flux distribution in terms of the overall fire heat release rate, fuel sootiness and separation distance. The results can be directly incorporated into engineering models of fire growth in storage geometries. The paper also provides additional data needed for the development of more general CFD models capable of predicting fire growth of other geometries. In this regard we provide correlations of: the velocities and temperatures of the flame gases in the central flue, the gas temperature distributions in the horizontal direction, and measurements of flame radiation.

**Experimental Set-up**

The experimental arrangement is shown in Figures 1 - 5. It is named after the well-known "World Trade Center" (WTC) towers in New York City. The steel towers were placed on a floor consisting of a perforated steel sheet covered by a layer of pebbles with a steel grating on top (high load capacity). Heated air was
supplied through the floor. It was heated above 40 °C to minimise water vapour from the combustion gases condensing on the tower walls. The floor was mounted above a non-operating heat resistance furnace. The air was heated up by a 22 kW heater attached to the duct system as shown in Figure 1. Each tower was 1.8 m high and 0.3 m x 0.3 m wide. The four towers were enclosed by four walls of 15 mm gypsum boards (1.9 m high) to maintain a uniform flow of air around the steel towers. The gaps between the towers, here called the flues, was varied. The air flow coming up through the floor was blocked inside the flues (see Figure 3).

Figure 1. Side view of the experimental set-up. Dimensions in mm.

The tower sides facing the flames consisted of 5 mm thick steel sheets (carbon), while the exterior sides consisted of 13 mm Gypsum boards. The back sides of the steel sheets were insulated with two layers of insulation: a 13 mm soft ceramic fibre insulation Kaowool (k=0.053 W/m °C at 200 °C, ρ = 96 kg/m³ and Cp= 1.07 kJ/kg °C) and a 20 mm rigid insulation board of type Rockwool: Rocky Takboard ( k= 0.039 W/m °C, ρ=200 kg/m³ , Cp not available). The insulation was compressed onto the steel sheets with pin spots.

Data was collected every second by a Orion data logger and then transferred to a Macintosh Power Book 520 for further analysis. The total time of each test was 2.5 minutes. The tests included flue widths of 0.06 m, 0.082 m, 0.1 m and 0.15 m where the majority of the tests were with a flue width of 0.1 m. The overall flame heat release rates ranged from 22.5 kW to 147.5 kW for three gaseous fuels: carbon monoxide (CO), propane ( C₃H₈) and propylene (C₃H₆). The fuels were
supplied from a gas cylinder with flow rate controlled by a BROOKS 5850E electronic mass flow controller. The temperature of the supplied fuel was close to ambient, i.e. between 15 to 20 °C. The total heat release rate was determined by continuously weighing the gas cylinder and multiplying inferred mass flow rate by the chemical heat of combustion given by Tewarson \[10\] as $\Delta H_c = 10.1 \text{ MJ/kg}$ for CO, $\Delta H_c = 43.3 \text{ MJ/kg}$ for C$_3$H$_8$ and $\Delta H_c = 40.76 \text{ MJ/kg}$ for C$_3$H$_6$.

![Diagram](image)

*Figure 2. Detail of 60 mm gas burner and porous floor. Dimensions in mm.*

The gas burners having exit orifice diameters of 40 and 60 mm were used. The inner surfaces of the nozzles were streamlined with a contraction ratio of 1.5 to 1.0. The burner orifices were placed 50 mm above floor level to prevent the flame from “belly flopping” onto the floor. Such “belly flopping” occurred for all three burners when the nozzles orifices were flush with the floor and the Froude numbers were less than 0.65 to 0.7. After streamlining and raising the nozzle orifices above the floor, as shown in Figure 2, the “belly flopping” disappeared.

The total heat flux, i.e. radiation plus convection, was measured by 0.5 mm type K thermocouples spot-welded to the back of the steel sheets. The surface of the steel was cleaned and polished before welding the thermocouples. A clamp was spot-welded 10 mm below each thermocouple to hold them in position. The thermocouples are located as shown in Figure 4 and attached as shown in Figure 5.

Three ray radiometers were used to measure flame radiance (e.g. flux per unit solid angle) across the flue at heights 0.5 m, 1 m and 1.7 m. Each radiometer was mounted such that the radiometer detector was located 0.85 m from the centre-line. The radiometers were calibrated by viewing the interior of a calibration oven at 1000 °C. The ray radiometers were designed and constructed at FMRC \[11\]. They are assembled from: (1) 25.4 mm in diameter stackable lens tubes from Thor Labs Inc. (2) Dexter M5 thermopile detector, (3) a zero reference shutter driven by an air piston, and (4) a focusing CaF$_2$ lens at the forward end. The spectral response of these radiometers is quite flat from 1 to 10 microns.
Figure 3. Plan view A-A of Figure 1. Dimensions in mm.

Figure 4. Location of heat flux thermocouples as well as gas temperature, velocity and radiation probes.
A Schmidt-Boelter gage [see e.g. ref. 12] was mounted flush to the steel surface 1 m above the floor level and 60 mm from the tower corner. Its sensing element is a water-cooled thermopile, with a varying number of junctions, depending on the measuring range. The sensing element measures the total heat flux to the thermopile.

The centre-line gas temperatures and velocities were measured at four different heights: $z = 0.57, 1.07, 1.42$ and $1.77$ m by sheathed 0.25 mm type K thermocouples and bi-directional probes [13], respectively. Thermocouples were also distributed horizontally to measure the width of the thermal plume at two elevations, $z = 1.07$ m and $z = 1.77$ m. At height $z = 1.07$ m the thermocouples were placed at distances: $x = 0.1$ m, 0.15 m, 0.2 m, 0.25 m, 0.3 m and 0.35 m from the centre-line; and at height $z = 1.77$ m $x = 0.1$ m, 0.2 m, 0.25 m, 0.3 m, 0.35 m, 0.4 m, 0.45 m, 0.5 m and 0.55 m from the centre-line. (These horizontal thermocouples are omitted in Figure 4 to avoid confusion with the wall heat transfer thermocouples).

The fire products were collected by an exhaust system (hood). Tests were carried out both with and without the exhaust system operating. When the exhaust system was operating, the exhaust flow was maintained slightly greater than inflow of air at floor level. When the exhaust system was turned off, the exhaust gases spilled over the top of the gypsum walls. Normally this exhaust system is not equipped with instruments to measure chemical or convective heat release rates. However, to get an estimate of the convective heat release rate, $Q_c$, and an indirect estimate of the overall heat loss to the walls, the oxygen content and the centre-line gas temperature in the exhaust system were measured. The convective heat release rate was determined by the equation:

$$Q_c = m_c c_p (T - T_a)$$  \hspace{1cm} (1)

where the measured gas temperature, $T$, was obtained with a 0.25 mm type K thermocouple in the centre of the duct and the ambient temperature was measured in the test hall. The heat capacity of air was assumed to be 1.01 kJ/kg K and the
mass flow rate, $m_{ex}$, in the duct was obtained from the following equation (Janssens and Parker [23]):

$$m_{ex} = \frac{Q(1 + \phi(\alpha - 1))M_a}{E\phi M_{O_2}(1 - X_{H_2O}^0 - X_{CO_2}^0)}$$

(2)

where $Q$ is the chemical heat release rate; $\alpha$ is an expansion factor put equal to 1.1; $M_a$ the molecular weight of air (28.85 g/mol); $M_{O_2}$ is molecular weight of oxygen (32 g/mol); $E$ is the amount of energy released by complete combustion per unit mass of oxygen consumed (kJ/kg oxygen); $X$ is the mole fraction; and $\phi$ is the oxygen depletion factor given by the following equation:

$$\phi = \frac{X_{O_2}^A - X_{O_2}^0}{(1 - X_{O_2}^A)X_{O_2}^0}$$

(3)

The constant $E$ for propane and propylene is according to Tewarson [10] 12,900 kJ/kg and 13,400 kJ/kg, respectively. The values of $X_{H_2O}^0$ and $X_{CO_2}^0$ were neglected in the calculation of $m_{ex}$. The chemical heat release rate, $Q$, was based on the measured fuel mass flow rate and the chemical heat of combustion given earlier. The average fraction of the convective heat release rate ($Q_c/Q$) for all the tests measured was 0.42 which is considerably lower than that for free burning flames, or 0.6 to 0.7 [10]. The fraction increases slightly with increasing $Q$.

Much effort was devoted to obtaining visually symmetric flames within the flues. This was difficult because the flames tended to lean away from the axis of symmetry. Turning the exhaust off seemed at first to stabilise the flames, but analysis afterward showed that this was not the case (see Table 1). With the exhaust system operating, cold air ($T_a$ in Table 1) was drawn in over the gypsum walls (enclosure) and due to density difference the colder air fell down to the floor level and increased the flame flutter. Increasing the supply velocity slightly through the floor did not improve the results. It is reasonable to believe that incorporation of horizontal flues would have improved the stability and the symmetry of the flames; but that would have complicated the test set-up. An objective judgement of flame symmetry was obtained by comparing the heat fluxes at 8 symmetric points at two different elevations. Of 41 tests only 7 tests could be regarded as being very symmetric. All of them had a flue width of 100 mm with the exhaust system turned on. Other tests did not have as good symmetry but yielded acceptable gas temperatures and velocities. All the tests provided useful flame heights, except those tests with flame heights higher than the towers. Table 1, summarises information for selected tests.
Table 1. Technical information about the tests used for the analysis.

<table>
<thead>
<tr>
<th>Test</th>
<th>Fuel</th>
<th>D (mm)</th>
<th>W (mm)</th>
<th>T₀ (°C)</th>
<th>Tₐ (°C)</th>
<th>Q (kW)</th>
<th>Qc (kW)</th>
<th>mex (kg/s)</th>
<th>ua (m/s)</th>
<th>Re</th>
<th>fan*</th>
<th>Sym**</th>
</tr>
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<tr>
<td>7</td>
<td>C₃H₆</td>
<td>60</td>
<td>100</td>
<td>51.4</td>
<td>-</td>
<td>85.7</td>
<td>-</td>
<td>0.22</td>
<td>4960</td>
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<td>vg</td>
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<td>100</td>
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<td>-</td>
<td>87.5</td>
<td>-</td>
<td>0.22</td>
<td>5064</td>
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<td>vg</td>
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<td>CO</td>
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<td>100</td>
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<td>-</td>
<td>22.5</td>
<td>-</td>
<td>0.22</td>
<td>2628</td>
<td>on</td>
<td>vg</td>
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</tr>
<tr>
<td>12</td>
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<td>100</td>
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<td>-</td>
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<td>17.7</td>
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<td>vg</td>
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<tr>
<td>13</td>
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<tr>
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<td>100</td>
<td>49.4</td>
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<td>142.4</td>
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<td>0.22</td>
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<td>60</td>
<td>150</td>
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<td>19</td>
<td>140.4</td>
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<td>0.22</td>
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<tr>
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<td>40</td>
<td>63</td>
<td>44.2</td>
<td>20</td>
<td>123.4</td>
<td>-</td>
<td>0.22</td>
<td>9857</td>
<td>off</td>
<td>a</td>
<td></td>
</tr>
<tr>
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<td>C₃H₆</td>
<td>40</td>
<td>63</td>
<td>48.2</td>
<td>21</td>
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<td>0.22</td>
<td>10782</td>
<td>off</td>
<td>a</td>
<td></td>
</tr>
</tbody>
</table>

*on = exhaust fan on, off = exhaust duct closed i.e. smoke leaking out under hood
**Symmetry: p = poor, a = acceptable, g = good, vg = very good
T₀ = avg. gas temperature just before ignition, Tₐ = ambient temp. outside the gypsum walls, mex = mass flow rate in the exhaust duct, ua = avg. vel. at floor (calc. from air mass flow rate), Re = Reynolds number

The total heat flux, \( q'' \), at different locations is obtained from the energy equation:

\[
q'' = \rho C_p \delta \frac{dT}{dt}
\]  

(4)

where \( \rho = 7800 \text{ kg/m}^3 \) is the density; \( C_p = 0.460 \text{ kJ/kg °C} \) is the specific heat; and \( \delta = 0.005 \text{ m} \) is the thickness of the steel sheet. The derivative, \( dT/dt \), was determined from the measured steel temperatures. One can readily show that errors due to: (1) heat loss from the back of the steel sheets, (2) finite temperature gradients across the thickness of the sheets, and (3) interference with neighbouring points in the axial and/or lateral directions are all negligible (i.e. always less than 5%). The largest errors are due to reradiation, \( \varepsilon \sigma T^4 \), from the steel sheet. However, this error can be readily corrected. The heat flux measured by a thermocouple is compared in Figure 6 with that measured by the Schmidt-Boelter gage. The thermocouple is at the same location as the Schmidt-Boelter gage except on the opposite tower. The values presented in Figure 6 are averaged over a certain time period. The Schmidt-Boelter gage reads somewhat higher possibly due to: (1) calibration, (2) reradiation but most likely (3) the fact that the Schmidt-Boelter gage is water cooled over a small area. Both Gardon gages and
Schmidt-Boelter gages generally read somewhat high due to local cooling of a small area containing the sensor.

Figure 7 shows steel temperatures measured at four lateral locations: \( x = 0.02 \) m, 0.06 m, 0.14 m and 0.29 m from the corner of a tower at height of \( z = 0.5 \) m. The temperatures increase quite linearly during the test. Thus despite significant differences in lateral heat transfer rates errors due to large temperature differences are not noticeable. In general we can say that the thermocouple method applied here gives reliable results.

To avoid water vapour from the combustion gases condensing on the steel and disrupting the heat transfer measurements, the heat fluxes were averaged over a 50 sec. time period including only steel temperatures above 60 °C.

Figure 6. Comparison of heat fluxes measured by Schmidt-Boelter (SB) gage and a thermocouple (Tc).
Figure 7. Steel temperatures from propylene flames with $Q=87.5$ kW and flue width of 100 mm. $x$ is measured from corner of tower.

Discussion of Results

Flame Heat Transfer

The maximum heat flux to the tower walls presumably occurs at the corners of the towers, i.e. closest to the centre of the exposed flames. Measurements were made at different elevations close (20 mm) to the corner of one of the towers. Because of problems in obtaining symmetrical results, only a few tests were found suitable for analysis, that is, those tests which were obviously symmetric. Figure 8 shows the heat fluxes the eight symmetric tests. Additional information about these tests are given in Table 1. The heat fluxes are plotted against the dimensionless flame height, $z/L_f$, at $x = 20$ mm, where $x$ is the horizontal distance from the corner of the tower.

Notice the dramatic effect of fuel type on the heat fluxes. As expected sootier flames produce greater heat fluxes. The heat transfer also increases with fire size. The shape of the heat flux distributions along the flame height is typical of buoyant turbulent flames. Carbon monoxide (CO) flames have their maximum heat flux at about 55% of the flame height; whereas propane ($C_3H_8$) and propylene ($C_3H_6$) flames have their maxima closer to 40% of their respective flame heights. All the heat fluxes become comparable at, or just above, their flame tips. The CO flames, being non-luminous (e.g. blue), have no soot and consequently minimal heat transfer by radiation. The reduced radiant cooling of
CO flames results in higher gas temperatures near the flame tip as shown in Figure 12. On the other hand propane (C3H8) flames are quite luminous, release considerably more radiation and produce greater heat fluxes as seen in Figure 8.

![Graph showing heat flux vs. z/Lf for different fuels]

**Figure 8.** The total heat flux at x = 20 mm plotted as a function of z/Lf for CO, C3H8 and C3H6. Flue width was 100 mm in all cases.

Soot forms rather early in the propane flames and then is almost completely oxidised in the upper regions of the flame resulting in a relatively high completeness of combustion (95 %) and little emission of smoke. In contrast propylene (C3H6) flames produce copious amounts of soot. Some of the soot escapes from the propylene flames before it can be oxidised resulting in a lower completeness of combustion (87.3 %) and modest release of smoke [10]. The radiation from all this soot produces the greatest heat fluxes among the three fuels. It also tends to cool the flames. We point out that there are many solid and liquid fuels, particularly aromatic fuels, which are much more sooty than propylene; so one should anticipate having even greater heat fluxes for aromatic fuels despite their lower completeness of combustion.

There is a large body of literature [14-18] showing that in buoyant turbulent diffusion flames the smoke-point of a fuel correlates the: (1) soot concentration in the flames, (2) release of smoke, (3) the radiant heat release and (4) incompleteness of combustion. Quantitatively, the soot concentration (e.g. sootiness) occurring in buoyant turbulent diffusion flames is inversely proportional to the smoke-point of the fuel. Here we define the sootiness, $S$, of a fuel to be inversely proportional to its smoke-point flame-height, $l_s$:
\[
S = \frac{l_{S,C_3H_8}}{l_{S,Fuel}}
\]  

and equal to unity for propane as shown in Table 2.

**Table 2. The smoke-point \( l_s \) and sootiness \( S \) for three gaseous fuels.**

<table>
<thead>
<tr>
<th>Fuel</th>
<th>Smoke-point ( l_s ) (mm)</th>
<th>Sootiness ( S )</th>
</tr>
</thead>
<tbody>
<tr>
<td>CO</td>
<td>( \infty )</td>
<td>0.0</td>
</tr>
<tr>
<td>( C_3H_8 )</td>
<td>162</td>
<td>1.0</td>
</tr>
<tr>
<td>( C_3H_6 )</td>
<td>29</td>
<td>5.6</td>
</tr>
</tbody>
</table>

Using this definition of \( S \) all the heat flux data of Figure 9 are correlated in Figure 10 by the function:

\[
\frac{q'' L_f^{4/3}}{Q} = f \left( \frac{z}{L_f}, S \right) = \frac{\left(0.569 + 0.684S^{0.4}\right)(z / L_f)^{0.73}}{\left[1 + (z / L_f)^{2.8}\right]^{1.6 + 0.8S^{0.4}}} 
\]  

(6)

where the flame height \( L_f \), is given by the formula:

\[
L_f = 0.315Q^{2/5} - 3.54w
\]  

(7)

This equation is based on a curve fit (least square method) of the experimental flame height data excluding those flame heights higher than the towers. The linear correlation coefficient, R, is 0.986.

The function, \( f \left( \frac{z}{L_f}, S \right) \), is plotted in Figure 9 for the three fuels. Its functional form allows for integration from \( z/L_f = 0 \) to \( \infty \). It is entirely empirical. The ample data speaks for itself. The formula should be applicable to all fuels having sootiness \( S = 5.6 \). We used MS Excel software to simultaneously minimise the sum of the square errors between the above function and all the data. The functional form was arrived at by first seeking a common functional form which matched the data for each individual fuel and then generalising the function for the various values of \( S \).
One is naturally interested in the dependence of the heat flux on the overall fire heat release rate, $Q$. The $4/3$ power on the left was not imposed by the authors. It was suggested by the least squares optimisation of the full set of data. The exponent appears a bit low. A value nearer 1.6 or 1.7 would have resulted in a fixed fraction of the overall heat release being transferred back to the walls. (Here it is assumed that the area of heat transfer is proportional to the flame height, $L_f$, multiplied by the flame width which presumably increases with the 0.7 power of the flame height as discussed below). In any case assuming a fixed fractional heat feedback the correlation suggests that the heat flux increases with heat release rate to a power between 0.33 and 0.47 depending on the exponent of in the above formula. Assuming an exponent equal to 0.4 or mid-way between these two values, one infers a heat flux increasing proportional to the overall flame height, $L_f$. This simple result needs verification by experiments covering a much greater range of flame heights or fire heat release rates.

The extent of heat transfer in the horizontal direction is also important. It controls the overall rate of heat release for growing fires. Our heat flux data in the horizontal direction is considerably more sparse. We only have data at two elevations, 0.50 and 1.0 meters. To determine the characteristic flame width, $b$, at each of these elevations, we interpolated the measurement at both elevations to fit the lateral curve where the heat flux was exactly equal to the value of the heat flux at the flame tip, $z/L_f = 1$, according to the above formula. A total of fifteen lateral positions were obtained in this manner. The numbers in the formula below were then optimised to yield the best overall fit to the fifteen measured values. The formula is bell shaped in a horizontal direction and matches the axial function $f(z/L_f, S)$ along the vertical axis, $x = 0.0$ m.
\[
\frac{q''L_f^{4/3}}{Q} = \exp \left[ -6 \left( \frac{x(1 + 0.3S^{0.4})}{z^{0.2}L_f^{0.5}W^{0.3}} \right)^2 + \int \left( \frac{z}{L_f}S \right) \right]
\]  

(8)

Figure 10. Curves of constant heat flux equal to value at flame tip.

The several coefficients and exponents shown above were determined by a simultaneous least squares fit to the data. Figure 10 shows the degree of fit for the three fuels. The data are insufficient to provide much confidence in the various coefficients and exponents contained in this formula. However, it should be reasonably valid for conditions similar to the experiment.

Flame Radiance

The radiance of the flame along the flue centre-line was measured by three ray radiometers at elevations \( z = 0.5, 1.0 \) and 1.7 m. Figure 11 shows the flame radiance versus dimensionless height, \( z/L_f \), for three tests. For these tests the ‘average’ flame heights were determined visually with a ruler. The method is subjective, but since the fluctuations of the flames were substantially less than those observed in free axi-symmetric plumes, the method appears to reasonable. From Figure 11 one notices that the highest total \( \pi \times \) radiance is obtained for propylene, was about 70 kW/m\(^2\), whilst for propane it was about 40 kW/m\(^2\). The lowest value was for carbon monoxide, or about 18 kW/m\(^2\). It is evident from these measurements that the relationship between soot production and radiation is
important. Very sooty flames yield relatively high radiation, whilst flames with no soot production yields relatively low radiation.

**Figure 11.** Flame $\pi x$ radiance ( $\pi x$ radiant flux per unit solid angle) vs. $z/L_f$.

**The In-Rack Fire Plume**

Figure 12 shows the non-dimensional flame height $L/w$ plotted against $Q^{2/3}/w$ for the World Trade Center (WTC) experiments for different flue widths $w$. For comparison the Figure 12 also shows flame height data from a large scale experiment with combustible material [19]. The data match surprisingly well. The large scale experiment was carried out under a fire products calorimeter having a capacity of up to 10 MW [20,21]. It involved a 2x2x4 high rack storage arrangement of a Standard Class II commodity. The commodity consisted of double tri-wall corrugated paper cartons (each 12 mm thick) containing sheet-metal liners and placed on a wood pallet. Each carton measured 1.08 m x 1.08 m x 1.08 m. The total height of the rack storage arrangement was 5.52 m. The width of the vertical flues was 150 mm and the height of the horizontal flues was 300 mm. The rack was four tiers high with four pallet loads on each tier. The ignition source consisted of insulating fibre board (similar to cellu-cotton rolls), 75 mm in diameter and 75 mm long, each soaked with 120 ml heptane and wrapped in a polyethylene bag. The mean flame height of the ignition source was about 0.9 m measured from the bottom of the lowest carton.

The correspondence between the WTC experiments and the large scale test is remarkably good. The authors had expected the flame heights to correlate with $Q^{2/3}/w$ suggestive of planar (i.e. two-dimensional) because the flue widths do
not change with height. Such model is outlined in reference [22]. Instead both sets of experiments correlate with the same linear function of $Q^{2/5} / w$ suggestive of axi-symmetric plumes. The agreement of the correlation with the full-scale tests is very encouraging. This result should be important for future fire spread studies of rack storage fires.

![Graph showing correlation of flame height](image)

**Figure 12.** Correlation of the non-dimensional flame height $L_f / w$ versus $Q^{2/5} / w$ for both the present WTC tower tests and full scale rack-storage test of a Class II Commodity.

Figure 13 shows the excess centre-line temperature plotted against the non-dimensional flame height, $z/L_f$, for the three fuels. The non-luminous (blue) CO flames are hotter than the luminous such as propane (bright yellow S=1) and propylene (nearly orange S=5.7) flames. The CO flames having no soot lose less heat by thermal radiation. The plotted temperature data for symmetric tests in Figure 13 clearly show this trend. It is apparent that the gas temperature does not only depend on the chemical heat release rate but also on the sootiness of the fuel. For $z / L_f \geq 0.7$ the temperature decreases with the -5/3 power of the dimensionless flame height. Here the flame heights were calculated from Equation (7) to reduce scatter. Figure 14 shows the normalised centre-line velocity, $u/L_f^{1/2}$, plotted against the non-dimensional flame height, $z/L_f$. Here again the flame heights were calculated from Equation (7) to reduce scatter. One sees that the normalised centreline velocity, $u/L_f^{1/2}$, are relatively constant values $z / L_f \geq 0.6$. 


Figure 13. The centreline excess gas temperature versus the non-dimensional flame height for the three fuels (w=100 mm).

Figure 14. The normalised centreline gas velocity $u / u_{L_f}^{1/2}$ plotted against the non-dimensional flame height $z / L_f$. 
Figure 15 shows the horizontal profile of the fire plume gas temperature at two elevations, \( z=1.07 \) and \( z=1.77 \) m. The profiles are plotted for the two heat release rates, \( Q=87.5 \) kW and \( Q=126.5 \) kW, with a flue width of \( w=100 \) mm. It is apparent that the temperature profile varies significantly in form depending on the height from floor. Any assumption that temperature profiles in horizontal sections are similar at all heights must be treated with caution. Based on temperature profiles such as those presented in Figure 15, the width of the thermal plume was determined. The results are plotted in Figure 16 where the thermal plume width, \( b_{\Delta T} \), is plotted as a function of height, \( z \). The thermal plume width, \( b_{\Delta T} \), is defined here as the width to the point where the temperature rise has declined to \( 1/2 \Delta T_0 \). The width is measured from the flue centre-line. A curve fit of the data, which was obtained at two elevations \( z=1.07 \) m and \( z=1.77 \) m for symmetric tests (test #s 8,11,12,13,15,17,24 and 25, see Table 1), yield the following equation:

\[
\frac{b_{\Delta T}}{L_f} = 0.177 \frac{z}{L_f} \tag{9}
\]

where \( L_f \) is calculated from equation (7).

**Figure 15.** Horizontal profile of fire plume gas temperature for two different heat release rates.
Figure 16. Normalised thermal plume width, $b_{\Delta T}/L_f^{1.1}$, plotted versus nondimensional flame height, $z/L_f$.

Figure 17. The heat flux averaged at $x=140$ mm and $z=1$ m for propane and propylene flames at different flue widths.
There is the important question whether the flue width has any important effect on the flame heat fluxes. This becomes important for fire spread. The data presented here do not show any noticeable effects for flue widths within the tested range (63 mm to 150 mm). This can be observed by plotting the average heat fluxes for different flue widths. By averaging the heat fluxes at $x=60$ mm and 140 mm, we were able to include tests which were not symmetric. The plot in Figure 17 shows the results for propylene and propane flames $x=140$ mm at $z=1$m.

**Conclusions**

The present study addresses questions of how: (a) the flame dimensions and (b) wall heat fluxes depend on the: (1) fire heat release rate, (2) fuel type and (3) wall separation distance. It provides the experimental data needed to guide the development of models capable of predicting fire growth rates in terms of tabulated properties of combustible materials. By providing formulas for the flame heat flux distribution in terms of the overall fire heat release rate, fuel sootiness and separation distance it is possible to directly incorporate the formulas into engineering models designed to predict fire growth in storage geometries. The paper also provides additional data needed for the development of more general CFD models capable of predicting fire growth of other geometries.

The heat fluxes were measured by thermocouples spot-welded onto the backside of the four steel towers. The measuring technique was found to be simple, accurate and rugged in addition to being inexpensive. The maximum error was estimated to be less than 5 %. The effect of fuel type on the measured heat fluxes was found to be surprisingly large. For the same overall fire heat release rates the peak heat fluxes from $C_3H_8$ flames were twice those from CO flames, whereas the peak heat fluxes from $C_3H_6$ flames were 2.8 times greater those from CO flames. As expected sootier flames produced greater heat fluxes. The heat transfer also increases with fire size. All the heat fluxes become comparable at or just above their flame tips. There is a distinct correlation between the sootiness of the flame and the measured gas temperatures. The low radiant cooling as in the case of carbon monoxide (CO) results in higher gas temperatures. Propylene ($C_3H_6$) flames on the other hand are highly luminous and release considerably more radiation and produce greater heat fluxes than CO resulting in more cooling of the flame envelope. Hence, the measured gas temperatures are considerably less than of CO. There are many solid and liquid fuels, particularly aromatic fuels, which are much more sooty than propylene; so one can anticipate even greater heat fluxes for aromatic fuels despite their lower completeness of combustion. It has not been generally appreciated that flame heat transfer can depend so much on the type of fuel.

In order to calculate the heat flux to the tower walls a general heat flux function is provided. It consists of the function $f$ describing the heat flux distribution along the vertical axis and the function $g$ describing the extent of heat transfer in the horizontal
direction or, 
\[
\frac{q''L_f^{4/3}}{Q} = f \left( \frac{z}{L_f}, S \right) g(x, L_f, z, w, S)
\]
where \(q''\) is the heat flux in kW/m\(^2\), \(L_f\) is the flame length given by equation (7), \(Q\) is the chemical heat release rate in kW, \(z\) is the height in meter from floor, \(S\) is the sootiness defined here as the inverse value of its smoke-point and equal to unity for propane, \(x\) is the distance from the tower corner, \(z\) is the height from floor in meter and \(w\) is the flue width in meters. The formula should be applicable to all fuels having sootiness in the range \(S=5.6\).

In the case of fixed fractional heat feedback to the walls the correlation suggests that the heat flux increases with heat release rate with a power between 0.33 and 0.47 depending on the exponent of \(L_f\) in the above formula. Assuming an exponent equal to 0.4 or mid-way between these two values, one infers a heat flux increasing proportional to the overall flame height. This simple result needs to be verified by experiments covering a much broader range of flame heights or fire heat release rates.

The plotted flame height data and excess gas temperature data appear to follow axi-symmetric fire plume laws. For axi-symmetric power law correlations, the flame height should follow the chemical heat release rate, \(Q\), to a power of 2/5, the excess centreline gas temperature should follow the instantaneous convective heat release rate, \(Q\), to a power of 2/3 and the -5/3 power to the height, \(z\). The centreline gas velocity should follow \(Q\) to a power of 1/3 and the -1/3 power to \(z\). Comparison of the flame height data was done with flame height data from a large scale experiment with combustible material. The agreement was remarkably good and this is a very encouraging result because use of model scale tests like the one presented here appears to yield reasonable results.

Great effort was put in obtaining symmetric flames within the flues. This was very difficult since the flames tended to lean from the symmetric axis. It is reasonable to believe that incorporation of horizontal flues would improve the stability and the symmetry of the flames but that would complicate the test set-up considerably.

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