ISSN 1173-5996

A STUDY OF FULL-SCALE ROOM FIRE EXPERIMENTS

BY

P. A. TONY ENRIGHT

Supervised by

Dr Charles M. Fleischmann

Fire Engineering Research Report 95/2
August 1995

This report was presented as a project report
as part of the M.E.(Fire) degree at the University of Canterbury

School of Engineering
University of Canterbury
Private Bag 4800
Christchurch, New Zealand

Phone 643 366-7001
Fax 643 364-2758

Abstract

This research is the first step towards providing the technical basis upon which the University of Canterbury could design and build a full-scale room fire laboratory.

A preliminary review of current full-scale room fire experimental methods is presented. The role of scale on fire experiments is discussed. Fundamental principles of room fire behaviour are examined. The derivation of the principle of oxygen consumption calorimetry in measuring fire heat release rate is exhaustively presented. The two zone fire induced vent flow model is rederived from inviscide flow theory incorporating the latest measurement techniques.

This document recommends that a full-scale room fire facility should be at least equivalent to the apparatus described in ISO 9705. The issues and considerations set down in the ASTM Standard Guide E-603 should also be addressed. The facility should also have the ability to be used for open air full-scale experiments.

Acknowledgments

"Cui dono lepibum novum libellum. Arido modo pumice exploitum ?"
Cattalus c84-c54 BC.

Its with great pleasure that I take this opportunity to sincerely thank all those who have contributed, directly or otherwise, to this project.

First and foremost I owe a debt of gratitude to my two supervisors, Andrew Buchanan and Charles Fleischmann. Andy, thanks for working so hard to make the Fire Masters program a reality. Thanks also for all that helpful advice and editorial work along the way. Charley, thank you for your continual guidance and the generous supply of encouragement and enthusiasm. I leave the program prematurely, but in awe of the world that you have introduced to me. An awe I would not have experienced under the direction of a lesser tutor or friend. Thanks.

Thanks to the New Zealand Fire Service for funding the Fire Engineering program. Specific thanks to those members of the NZFS staff that have provided helpful advice along the way; Wayne Bedford, John Sinclair and Joe Hefford.

Thanks to all my family; Paddy, Josie, Vicky, Rex, Wade, Jodi, Stephanie and David. A very special thanks to my brother Wade, an Electrical and Electronic Engineering PhD candidate, who was never short of rich philosophical observations along the way. Thanks Wade, your contribution was inspiring.

Thanks to the friendly and efficient staff of the Engineering Library; Heather, Pat, Helen, Diane, Jill, Adam, Vanessa and Gwen. This work would have been a very amateur attempt without your valuable assistance. Thanks also to the BRANZ library which held a large proportion of that information not available at Canterbury University.

Thanks to my fellow post-grad Civil Engineering students. A special mention for the full time 'fire-boys'; Darin Millar, Ivan Bolliger, Faran Rahmanian and Hans Gerlich. A great bunch of fellows.

Thanks to the cafe ladies; Leslie, Jean and Phil. The technical support staff especially Brandon, Peter and my aerobics buddy Catherine Price from the Civil Engineering office.

Thanks to John Henry, Building Control Engineer at the Christchurch City Council who was as much a mentor as an employer. Simular thanks go to Hamish MacLennan, Director of the Holmes Consulting Group Fire Engineering Division.

Thanks finally to the supportive friends and flatmates I've had during this period. Especially Helen Pullar, Tania Daly, Pamela Parker, Adrian Paterson, Daryl Strawbridge, Jessie Blackdog and Jed Soane.

Table of Contents

Abstract.......... i
Acknowledgments.......... iii
Table of Contents.......... v
List of Tables and Figures.......... vii
Nomenclature.......... ix

1. Introduction.......... 1

Part A: Part A: Tests and Experimental Methods

2. Full-scale Room Fire Tests and Experiments 3
3. The Role of Scale in Fire Tests and Experiments 9

Part B: Determination of Room Fire Properties

4. Measurement of Heat Release Rate Using the Oxygen Consumption Technique..... 29
5. Measurement of Mass Flow Rate of the Exhaust Gases.......... 63
6. Measurement of Fire Induced Vertical Vent Flows.......... 77

7. Summary and Conclusions.......... 97

References.......... 99
Bibliography.......... 105

List of Tables and Figures

Tables

Chapter 5

Table 1 Exponent ‘n’ of velocity distribution.... 69

Table 2 Velocity shape factor.... 69

Table 3 Position of mean local velocities.... 73

Figures

Chapter 2

Figure 1 The Monasto Calorimeter.... 4

Figure 2 The ASTM Room Fire Test.... 5

Figure 3 The NORDTEST Room Fire Test.... 7

Chapter 3

Figure 1 Overall view of cone calorimeter apparatus.... 12

Figure 2 Gas analyser instrumentation layout.... 14

Figure 3 Smoke obscuration measuring system.... 17

Figure 4 The NBS Furniture Calorimeter.... 23

Figure 5 General schematic of the ICAL apparatus.... 24

Figure 6 Comparison of peak heat release rates.... 26

Chapter 4

Figure 1 Gas analysis configuration 1.... 31

Figure 2 Gas analysis configuration 2.... 42

Figure 3 Gas analysis configuration 3.... 46

Figure 4 Gas analysis configuration 4.... 52

Chapter 5

Figure 1 Schematic of room fire test.... 63

Figure 2 Velocity Profile.... 65

Chapter 6

Figure 1 Rockett’s two zone model.... 79

Figure 2 Pressure - height relationship.... 83

Figure 3 Layout of measuring apparatus.... 95

Nomenclature

Greek Symbols

α = Expansion factor (--)

ρ_0 = Gas density in zone 0 (kg/m^3)

ρ_1 = Gas density in zone 1 (kg/m^3)

ρ_2 = Gas Density in zone 2 (kg/m^3)

ρ_3 = Gas Density in zone 3 (kg/m^3)

ρ_a = Density of ambient air (kg/m^3)

ρ_e = Density of exhaust gas (kg/m^3)

$\rho_1(z')$ = Density of the compartment gases at measuring point z′ (kg/m^3)

ρ_{ref} = Density of ambient air at temperature T_{ref} and atmospheric pressure (kg/m^3)

σ_f = Specific extinction area (m^2/kg)

ϕ = Oxygen depletion factor (--)

μ = Kinematic viscosity of the gas/fluid (kg/m/s)

Symbols

a_i = Area of element i (m^2)

A = Cross sectional area of the duct (m^2)

A_{cc} = Cone Calorimeter sample area (m^2)

b = Scaling factor (m^2/kg)

B = Height of vent (m)

c_{in} = Orifice discharge coefficient ‘in’ (--)

c_{out} = Orifice discharge coefficient ‘out’ (--)

C = Orifice calibration constant (--)

C_1, C_2, C_3 = Numerical constants (m/s)

d = Characteristic dimension of exhaust duct (m)

D = Diameter (m)

E = Hugett’s constant (MJ/kg)

E_{CO} = Net heat released per unit mass of O_2 consumed for the combustion of CO to CO_2 (MJ/kg of Oxygen consumed)

E_g = Heat release associated with the consumption of one kg of O_2 when combustion occurs with gas 'g' (MJ/kg of O_2)

$f(\Re_e)$ = Function of the Reynolds Number (--)

$[FF]$ = Frame factor (--)

H = Compartment height (m)

$\Delta h_{c,eff}$ = Effective heat of combustion of sample (MJ/kg)

$\Delta h_{c,eff}(t)$ = Transient effective heat of combustion of sample (MJ/kg)

$(\Delta h_c)_g$ = Heat of combustion of gas 'g' (MJ/kg)

I = Intensity of light transmitted over distance l (cd)

$I_i(t)$ = Instantaneous value of transient intensity of light (cd)

I_0 = Intensity of light at the source (cd)

k = Extinction coefficient (m^{-1})

k_c = Shape factor (--)

$k_i(t)$ = Instantaneous value of transient extinction coefficient (m^{-1})

l = Path length (m)

L = Compartment length (m)

m_i = Mass of species i (kg)

m_f = Final mass of sample (kg)

$[M]$ = Mass of full-scale specimen excluding non-combustible parts (kg)

M_i = Molecular weight of species i (kg)

$\dot{m}_a$ = Mass flow rate of ambient air into the system (kg/s)

$\dot{m}_b$ = Mass flow rate of ignition source (kg/s)

$\dot{m}_e$ = Mass flow rate of the exhaust gases (kg/s)

$\dot{m}_g$ = Mass flow rate of gas to the burner (kg/s)

$\dot{m}_i$ = Mass flow rate of species i (kg/s)

$\dot{m}_{in}$ = Mass flow rate of air in through vent (kg/s)

$\dot{m}_{out}$ = Mass flow rate of air 'out' through vent (kg/s)

m_r = Mass accumulated inside the compartment (kg)

$\dot{m}_v$ = Mass flow rate of water vapour and pyrolozates emerging from the compartment surfaces (kg/s)

$\dot{m}_{CO_2}$ = Mass flow rate of carbon dioxide in the exhaust gases (kg/s)

$\dot{m}_{H_2O}$ = Mass flow rate of water vapour in the exhaust gases (kg/s)

$\dot{m}_{N_2}$ = Mass flow rate of nitrogen in the exhaust gases (kg/s)

$\dot{m}_{O_2}$ = Mass flow rate of oxygen in the exhaust gases (kg/s)

$\dot{m}_{CO}$ = Mass flow rate of carbon monoxide in the exhaust gases (kg/s)

$\dot{m}^o_{CO_2}$ = Mass flow rate of carbon dioxide into the system (kg/s)

$\dot{m}^o_{H_2O}$ = Mass flow rate of water vapour into the system (kg/s)

$\dot{m}^o_{N_2}$ = Mass flow rate of nitrogen into the system (kg/s)

$\dot{m}^o_{O_2}$ = Mass flow rate of oxygen into the system (kg/s)

$(\Delta\dot{m}_{O_2})_g$ = Change in the mass flow rate of oxygen in the exhaust gases due to the consumption of oxygen by the ignition source (kg/s)

n = Number of moles (--)

n_1 = A Reynolds number dependent, power coefficient (--)

n_i = Number of moles of species i (--)

N = Number of elements (--)

P_0 = Gas pressure in zone 0 (Pa)

P_1 = Gas pressure in zone 1 (Pa)

P_a = Pressure of the ambient air (Pa)

$P_{ambient}$ = Ambient pressure (Pa)

$P_{compartment}$ = Compartment pressure (Pa)

$P_s(T_a)$ = Saturation pressure of water vapour at T_a (Pa)

ΔP = Differential pressure (Pa)

ΔP_{01} = Pressure differential at vent between zones 0→1 (Pa)

ΔP_{32} = Pressure differential at vent between zones 2→3 (Pa)

ΔP_{int} = Pressure loss within the interior of the compartment (Pa)

ΔP_c = Differential pressure across the duct center-line (Pa)

ΔP_i = Differential pressure at measuring point i (Pa)

Q_{in} = The volumetric flow rate 'in' through vent (m^3/s)

$\dot{q}$ = Heat release rate (kW)

$\dot{q}_{fs}$ = Full-scale heat release rate (kW)

$\dot{q}_i(t)$ = Instantaneous value of heat release rate (kW)

$\dot{q}_{ig}$ = Heat release rate of the ignition source (kW)

$\dot{q}_{Total}$ = Hypothetic heat release rate were combustion complete (kW)

$\dot{q}''_{bs}$ = Bench-scale heat release rate per unit area (kW/m^2)

r = Radial position at which the velocity $\bar{v}_i$, is measured (m)

R = Pipe/duct radius (m)

$\Re_e$ = Reynolds number (--)

$[SF]$ = Style factor (--)

$\dot{S}$ = Smoke release rate (m^2/s)

$T_1(z')$ = Temperature of the compartment gases at measuring point z′ (K)

T_a = Ambient absolute temperature (K)

T_e = Exhaust gases absolute temperature (K)

$T_d(z')$ = Temperature distribution at the vent as a function of height (K)

T_{ref} = Reference temperature (K)

v_0 = Gas velocity in zone 0 (m/s)

v_1 = Gas velocity in zone 1 (m/s)

$\bar{v}$ = Mean velocity of the exhaust gas (m/s)

$\bar{v}_i$ = Local mean velocity of the exhaust gas through area element i (m/s)

v_2 = Gas Velocity in zone 1 (m/s)

$v(y)$ = Velocity at point y, along the diameter (m/s)

$\dot{V}$ = Volumetric flow rate of the exhaust gases at the smoke meter (m^3/s)

W = Compartment width (m)

X_i = Mole fraction of species i (--)

X^A_{CO} = Mole fraction of carbon monoxide present in the analyser during test (--)

$X^A_{CO_2}$ = Mole fraction of carbon dioxide present in the analyser during test (--)

$X^A_{H_2O}$ = Mole fraction of water vapour present in the analyser during test (--)

$X^A_{N_2}$ = Mole fraction of nitrogen present in the analyser during test (--)

$X^A_{O_2}$ = Mole fraction of oxygen present in the analyser during test (--)

$X^{A^o}_{CO}$ = Mole fraction of carbon monoxide present in the analyser prior to test (--)

$X^{A^o}_{CO_2}$ = Mole fraction of carbon dioxide present in the analyser prior to test (--)

$X_{H_2O}^{A^o}$ = Mole fraction of water vapour present in the analyser prior to test (--)

$X_{N_2}^{A^o}$ = Mole fraction of nitrogen present in the analyser prior to test (--)

$X_{O_2}^{A^o}$ = Mole fraction of oxygen present in the analyser prior to test (--)

z_D = Height of thermal discontinuity (m)

z_N = Height of neutral plane (m)

z' = Measuring point (m)

z'' = Measuring point in quiescent corner of compartment (m)

Chapter 1

Introduction

The purpose of this document is to produce the basis upon which the University of Canterbury could design a full-scale room fire facility. Users of such a facility should be referred to this document as background to the underlying fundamental principles.

This document consists of a review of full-scale room fire experimental methods. Common types of full-scale room fire apparatus are considered. These consist of the Monasto Room Calorimeter, the ASTM standard guide and the ISO/NORDTEST standard room. The concept of the proposed University of Canterbury Room Calorimeter is introduced.

The role of scale on fire experimental methods are discussed. In particular the use of the cone calorimeter as a bench-scale tool is covered. Its function in predicting full-scale (open air) behaviour is examined.

Fundamental principles of room fire behaviour are examined. In particular, detailed analysis's of the properties of heat release rate and fire induced vent flows are presented.

This document does not cover other significant room fire properties such as temperatures and heat flux. Similarly some significant fuel properties such as ignition, smoke production and toxicity are not included. While these properties are of interest to investigators, due to time constraints they are not embodied in this document. It would be logical for this work to be extended to cover these topics, at a future date.

The technical content of this document is divided into two sections, parts A and B. Part A: "Tests and Experimental Methods" is a general discussion of the topic. Part B "Determination of Room Fire Properties" is a detailed examination of fundamental principles.

The researching of this topic involved a large component of literature review. Of the material referenced, some papers and authors deserve a special mention as their contribution was disproportionately significant.

Four papers of particular importance to this research were, in order of their publication date:

Thornton, W. 1917. “The Relationship of Oxygen to the Heat of Combustion of Organic Compounds.” *Philosophical Magazine and Journal of Science.* v33 p196-203.

Huggett, C. 1980. “Estimation of Rate of Heat Release by Means of Oxygen Consumption” *Fire and Materials.* v4 No. 2 p61-65.

Parker, W. 1982. “Calculations of the Heat Release Rate by Oxygen Consumption for Various Applications” NBSIR 81-2427. National Bureau of Standards Gaithersburg, MD.

Janssens, M. 1991. “Fundamental Thermophysical Characteristics of Wood and their Role in Enclosure Fire Growth” Dissertation for the Degree of Doctor of Philosophy, University of Ghent (Belgium), National Forest Products Association, Washington, DC.

Finally a special mention is due to Vytenis Babrauskas, in recognition of the prolific number of papers he has produced related to this research.

Chapter 2

Full-scale Room Fire Tests and Experiments

Introduction

Extensive empirically based full-scale room fire experiments have been conducted in earnest since the 1950s. These early experiments were usually conducted as fire endurance tests, with the average room temperature considered the bounding condition.

It was also at about this time that some of the first post flashover room fire theories were being developed. It took until the mid 1970s for reasonable pre flashover room fire theories to become available. Temperatures were still considered the property of primary importance but it was also becoming apparent that it was useful to measure other quantities, such as smoke and gas production.

Investigators then began to think not only in terms of a structural member's fire endurance, but also of room items' and interior finishing's reaction to the fire and contribution to the fire growth.

There was a distinct lack of cohesion and poor agreement between these early tests. Emmons (1968) highlighted how different materials can rank in a different order, depending on the reaction to fire test conducted. This indicating that the tests, among other things were instrument dependant. In other words they were non-fundamental.

During this period came the development of a bi-directional velocity probe (McCaffrey and Heskestad 1976). This device is extremely beneficial to investigators in that it is reasonably insensitive to angular velocity and robust enough to measure velocities in a soot-laden and high temperature environment. This enables much better measurements of enclosure fire characteristics.

Also at about this time the heat release rate was beginning to be recognised as the fire property of most significance. In other words if you could manage the heat you could manage the hazard. Sensible enthalpy heat flow calorimetry was being proposed as a tool to indirectly measure this property (see the Monsanto Room Calorimeter in the next section).

The most significant of all the developments, in terms of measurement of the heat release rate, occurred with the application of the principal of oxygen consumption (Huggett 1980). This set the way for the heat release rate to be measured directly. This principal and its application form the basis of most of this document.

Common Types of Existing Full-scale Room Fire Tests and Experiments

The Monsanto Room Calorimeter

At about the same time as the implementation of the principal of oxygen consumption in room fire experiments, at Monsanto Chemical, Warren Fitzgerald developed a full-scale room calorimeter based on a statistical method of correlating various room temperature measurements and relating these to the heat release rate (Fitzgerald 1978).

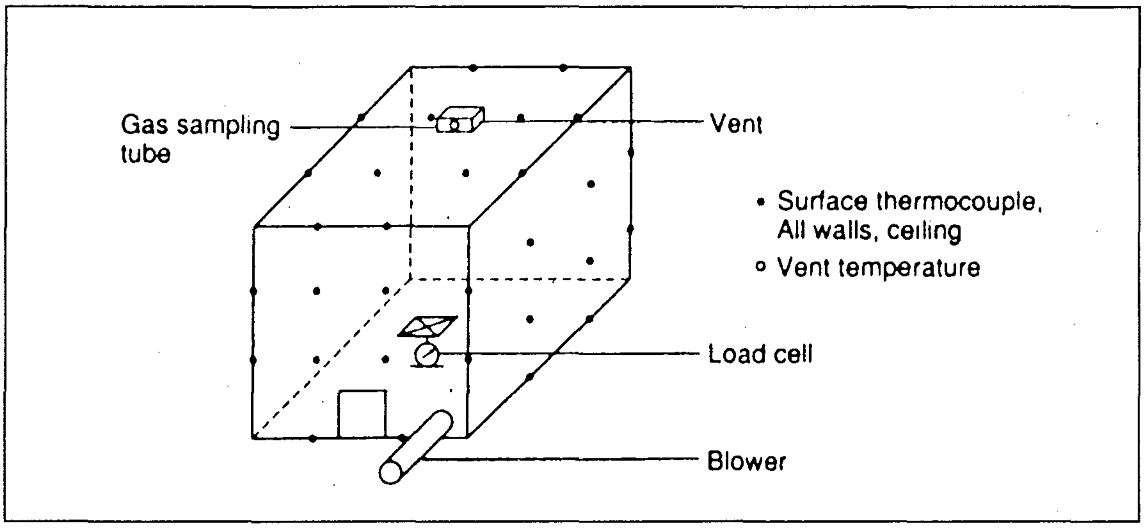

Figure 1: The Monasto Calorimeter (Babrauskas 1992b)

The room was cubic and of dimensions of 2.7 m in length width and height with a single vertical door-type vent. Thermocouples were located; in the compartment, on the bounding surfaces and in the gas exhaust duct. Air was forced into the room by a small supply duct. See figure 1. The specified capacity of this experiment was 140 kW which is considered a low value of heat release rate. The emphasis of the Monasto Calorimeter was the measurement of heat release rate of discrete room items such as furniture.

The Monsanto Calorimeter never really gained favour as an experimental tool, partially because of its limited capacity, but probably more significantly because of the very successful adaptation of the principal of oxygen consumption in room fire experiments elsewhere.

The ASTM Guide to Room Fire Tests and Proposed Standard Full-scale Test for Surface Products

With the development of the principal of oxygen consumption in 1980 came many full-scale room fire experiments, with widely varying geometry's and scenarios. These were being conducted independently by many different research institutes.

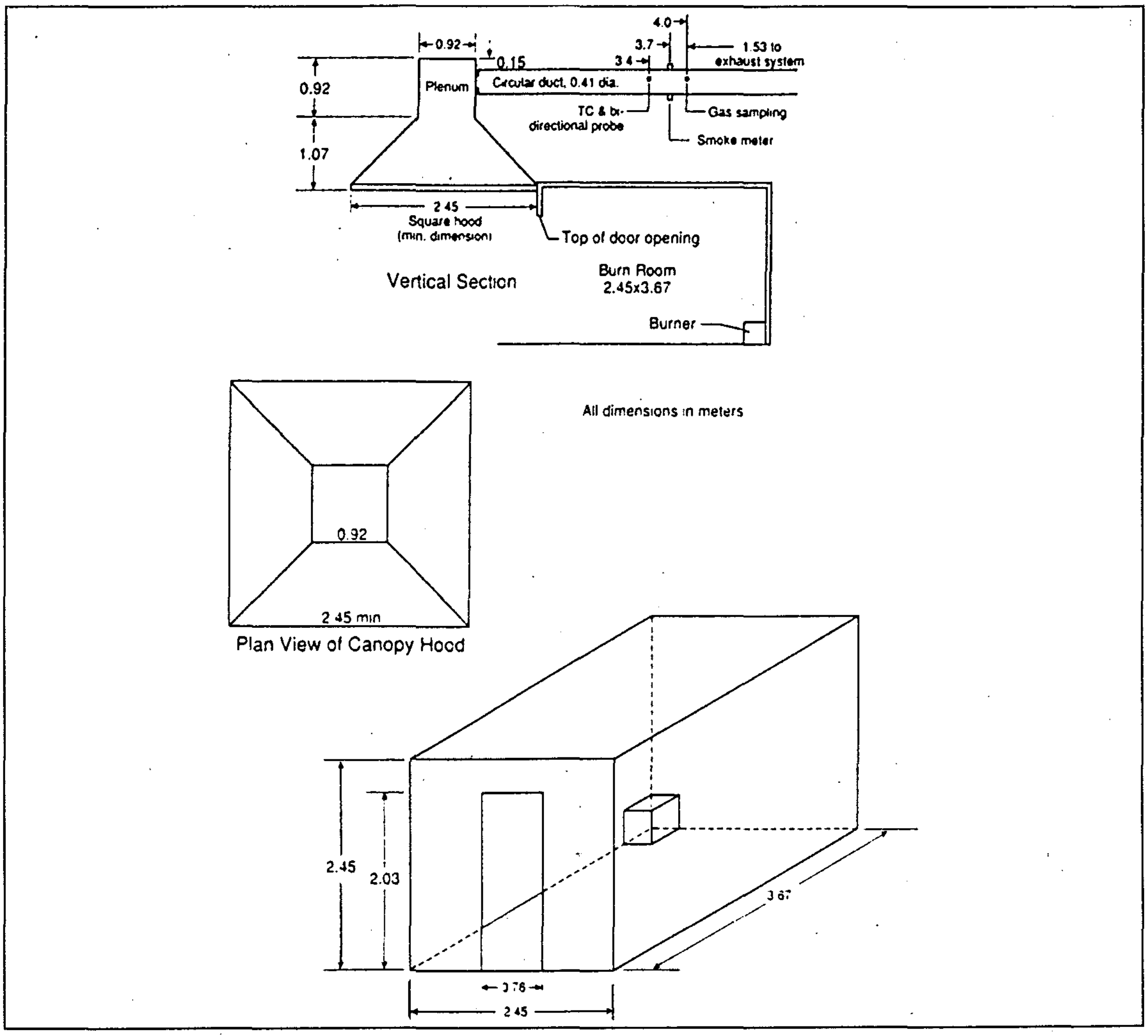

Figure 2: The ASTM Room Fire Test Geometry, Reproduced from Babrauskas (1992b)

One of the principal intentions of these experimental programs was the validation of models relating bench-scale to full-scale. The need arose to standardise a test method for full-scale room fire experiments.

A standard guide ASTM (1977) was developed from work at the University of California. The room geometry is shown in figure 2 is not a standard test method, it is a guide. It aims to set out good standard practice in designing and conducting room fire experiments, rather than proscribing conditions. For example it suggested fire properties which are desirable to measure such as; heat release rate, heat flux, temperatures, air velocities, fire propagation, smoke production, gas concentrations and toxicity.

From this guide, ASTM published a proposed full-scale room fire standard test method, ASTM (1982). It is specifically aimed at testing the various wall and ceiling lining materials. The ASTM proposed standard follows the geometry suggested in the guide, also includes proscriptive information with respect to the ignition source and output. A significant world-wide round robin testing program was conducted in an attempt to expedite the acceptance of this proposal (White 1986, Beitel 1990).

Babrauskas (1992b) discusses some non referenced work from the University of California which highlights a problem with this proposed standard apparatus, in terms of overly complex exhaust systems. This problem appears to have contributed to the standard's withdrawal. However, variations of the proposal continue to be used in a number of laboratories.

The ISO Standard, Full-scale Room Fire Test for Surface Products

ISO 9705:1993(E), is titled "Fire Tests - Full-scale Room Test for Surface Products". The scope of this standard describes the test conditions as simulating a pre flashover, well ventilated, compartment fire, with an open door. This is a test of the reaction of the wall and ceiling linings to a fire. In other words it is a reaction to fire test not a fire endurance test.

The test method has similarities to the ASTM guide in terms of the geometry (in this case 2.4 × 3.6 × 2.4 m) and calorimetry equipment. This standard is a prescriptive test method and as such goes into a higher level of detail. Such details include specifying an ignition source (with some scope for alternatives), the ignition source location and transient energy output levels.

The genesis of the ISO room fire test was the NORDTEST room fire test which is shown in figure 3.

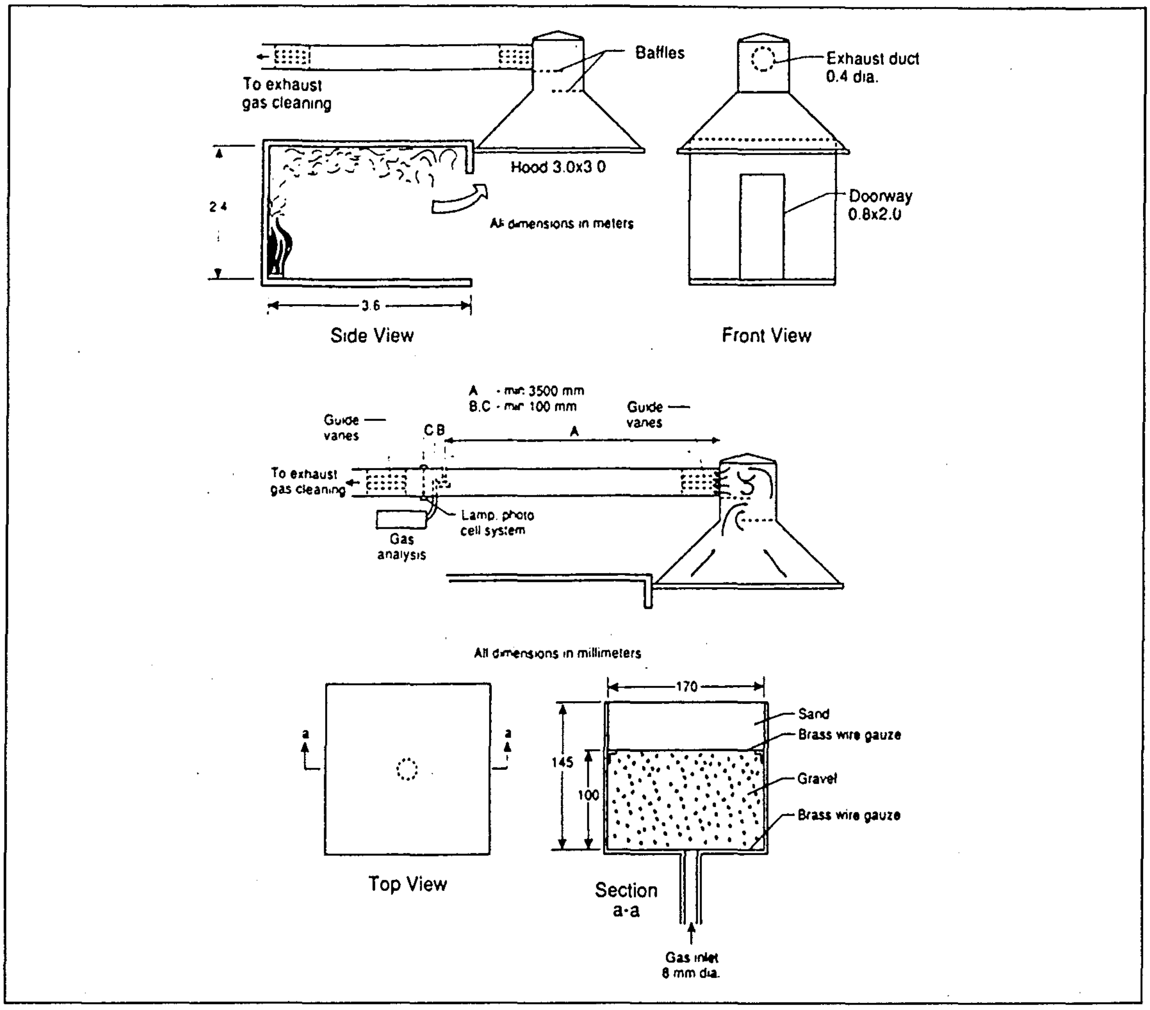

Figure 3: The NORDTEST Room Fire Apparatus. Reproduced from Babrauskas (1992b).

The Proposed University of Canterbury Full-scale Room Calorimeter

In considering such a research tool as a room calorimeter being developed at the University of Canterbury, the first step is to review the simular existing standard methods. The major examples are discussed in the previous section.

It is worth reiterating that the existing international and US full-scale room fire standards only relate to testing room linings as a building material. Standards do not yet exist for other types full-scale room fire experiments.

Currently the University of Canterbury are exploring the possibility of setting up a full-scale room fire experiment program. In fact a series of exploratory full-scale backdraft experiments have recently been conducted (Fleischmann, Bolliger and Millar 1995).

It is probable that the future emphasis at Canterbury will continue to be on experiments other than those standardised. For example, as well as continuing the backdraft work, there is a strong possibility that the University may wish to explore post flashover temperature histories in full-scale room fires.

The facility, in which these experiments will be conducted would not be dedicated to the round robin testing of lining materials, such as the ISO 9705 room. However common sense dictates that it should be at least equivalent. To this, ideally, all the ASTM Standard Guide recommendations should eventually adopted. This will be dictated by funding, heat release rate measurements should be prioritise highest with possibly smoke production and toxicity given lowest priority. To this would be such 'custom built' additions specific to Canterbury's aims.

Such features might include:

- The ability to roll the compartment away, lower the hood and extract systems (or raise the fuel), and use this as a furniture calorimeter.
- Additional 'mobile' compartments which could be rolled into and out of place.
- Varied ventilation conditions, such as a window and horizontal vent.
- Gas analysis of total unburnt hydrocarbons.
- Inclusion of fire safety detection and suppression systems such as; forced ventilation and sprinklers.

This list is reasonably arbitrary, in no particular priority and by no means exhaustive. These additional features will be dependant upon a defined experimental program being prepared by the University.

Chapter 3

The Role of Scale in Fire Tests and Experiments

Introduction

This chapter describes the cone calorimeter and the furniture calorimeter, in terms of their complimentary role with respect to full-scale room fire experiments. It discusses the use of bench-scale tests to predict free burning full-scale behaviour. Converting this to full-scale compartment conditions is briefly explored in terms of reduced scale modeling. If this last step appears 'hazy' then it fairly reflects the state of our knowledge at present.

Originally bench scale fire tests were used to rank products or materials according to intuitively selected performance criteria. Most often this was the sample's ignitability, but also some measures of flame spread and smoke production were attempted.

Separately to this, there were many full-scale fire resistance tests of structural members being conducted in furnaces in testing laboratories. In research laboratories full-scale room fire experiments were being conducted on lining materials for their contribution to the pre flashover fire growth.

There was little to no cohesion in fire experiments and most of these were empirically based.

However, as discussed in the previous chapter "The Full-scale Room Fire Experiment", the development of the *principal of oxygen consumption* (Huggett 1980), resulted in reliable full-scale heat release rate assessment methods becoming available. The modern room fire test and the furniture calorimeter were designed from this fundamental basis.

This dramatically changed the experimental side of fire engineering. It was realised that due to the availability of this validating full-scale data, quantitative models could be written to predict full scale behaviour from bench scale results. Combining previously disjointed tests.

In fact it was further realised (Babrauskas 1985) that room fire experiments may not be the best way of characterising burning behaviour in cases such as furniture involvement. This is because the free burning heat release rate is a more generalised form of data than the compartment-bounded condition.

Efforts were made to reconcile bench scale data to full scale properties. This went a long way with the development of the cone calorimeter. Prior to this existing bench scale instruments such as the Ohio State University Rate of Heat Release Apparatus were commonly used. Such calorimeters incorporated a sensible enthalpy flow measurement technique to describe a 'convective component' of the heat release rate. Babrauskas (1992a) describes sensible enthalpy flow measurements as "misleading" on a bench-scale. Fire protection engineering science had evolved from the early fire experiments to more fundamental fire assessment methods (Williamson and Dempsey 1993).

Recent work has been undertaken to develop an intermediate-scale calorimeter. This is primarily, but not exclusively, for experimental assemblies in the vertical mode. This development is discussed later in this chapter.

This chapter does not go so far as to describe scaling applications associated with reduced scale-models of systems in detail. In brief these applications are based on the dimensionless groups derived from governing differential equations. The use of dimensionless analysis leads to good generalised results. There is an art to this type of modelling, for it is not possible to comprehend and include all of the variables present, hence trade offs are necessary. A fuller description can be found in Quintiere (1989).

The following is a brief review of the various types of small and intermediate-scale experimental methods and a discussion of their relationships.

The Cone Calorimeter - A Bench Scale Heat and Visible Smoke Release Rate Apparatus Using an Oxygen Consumption Calorimeter

The cone calorimeter is a bench scale tool used to measure heat release rates and associated fire properties of a small product sample. This section describes the instrument, discusses its components and introduces the underlying theoretical considerations.

The cone calorimeter, so named because of the shape of its heater, was developed at the National Institute of Standards and Technology (NIST), then the National Bureau of Standards (NBS), in the early 1980s (Babrauskas 1982). Its purpose is to measure the properties on a bench-scale of a product sample subject to the fire process. The properties of interest were identified at the conception as being:

- Ignition properties.
- Flame spread behaviour.
- Heat release rate per unit area.
- Smoke release rate per unit area.

The long term objective is to use the data gained from these tests as the basic parameters in full-scale fire modelling and computer simulation.

The primary function of a cone calorimeter in determining these properties, is the measurement of the heat release rate of a product sample and hence an estimation of that product's contribution to the fire. This allows approximations of the product's properties to be made, independent of a knowledge of the exact combustion chemistry.

Smoke generation is the second most significant parameter in defining the overall fire hazard. The cone calorimeter was subsequently modified to include smoke measurements along with some toxicity measurements (Babrauskas 1987).

The American Society for Testing Materials (ASTM) approved a cone calorimeter standard in 1990. This was given the designation ASTM E 1354 1990, and titled: Standard Test Method for Heat and Visible Smoke Release Rates for Materials and Products Using an Oxygen Consumption Calorimeter. A similar standard was published by the International Standards Organisation (ISO), designated ISO 5660 1992. The National Fire Protection Association (NFPA) issued a parallel standard to both of these in 1990. It targets the specific application of testing upholstered furniture composites, NFPA 264A. As a result of the development of the cone calorimeter NIST was awarded an 'R&D 100' award in 1988. This recognition in the field of research and development, is additionally significant in that this is the first such award for a reaction to fire test.

Description of the Apparatus

Figure 1 is an overall view of the apparatus, showing the position of all the main components. This figure has been reproduced from the original NBS report on the apparatus (Babrauskas 1982).

The cone calorimeter apparatus consists of many components and systems. The general properties of these components are prescribed in ASTM E 1354. For simplicity, in a review such as this it is convenient to group these components and systems into the following items or categories:

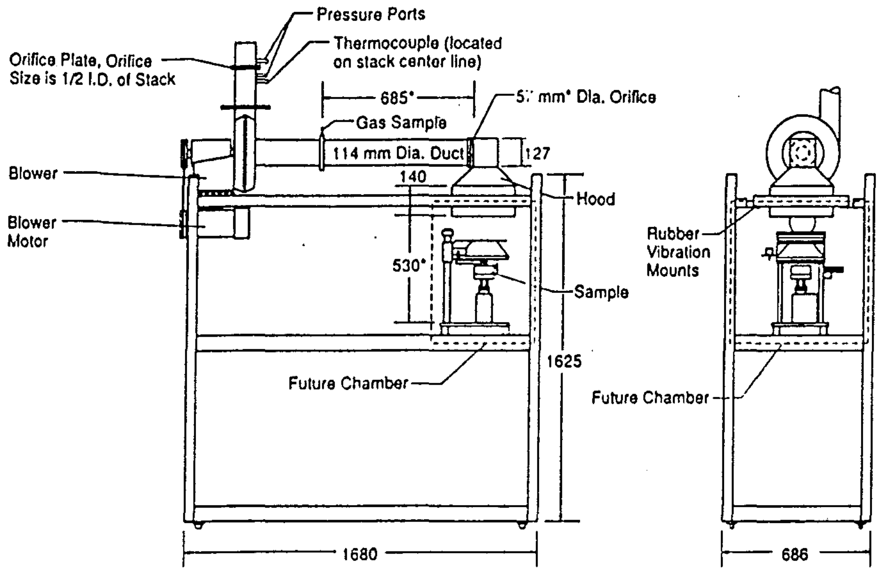

Figure 1: Overall View of Apparatus. Reproduced from ASTM E 1354.

- Conical Heater. The heater is similar to that used in the ignitability test ISO 5657, except for a greater heat flux capacity of up to 100 kW/m^2 from a maximum operating level of 5 kW at 240 V.

  The 8 mm diameter and 3.4 m long element consists of heating wire packed in MgO refractory and swaged in a high temperature alloy sheath. The element is wound into a cone shape and inserted in a double-walled steel truncated cone.

Electronic control is effected by fixed thermocouples in direct contact with the heater element. The element should be able to be maintained at a steady temperature to within +/- 2 ^{0}C.

The cone and sample can be arranged both vertically or horizontally.

- Exhaust System. The hood is of dimensions adequate to ensure test products are removed. The ducting is designed to enhance mixing and flow development via a restrictive orifice. An orifice flow plate, a centre-line bi-directional probe and a centre-line thermocouple provide data for mass flow rate calculations. The exhaust gases are driven by a high temperature centrifugal fan in the range of $0.012m^3/s$ to $0.035m^3/s$.

  It is worth noting that in most respects this part of the system is simply a scaled down version of the system described in the chapter "Mass Flow Rates of the Exhaust Gases".

- Sample Mounting and Load Cell. Samples are cut to 100 mm × 100 mm in size, can be up to 50 mm thick and are contained in a robust stainless steel mounting. The sample is encased on all non-exposed sides in aluminium foil to avoid spilling of molten material. This measure also limits edge burning. The bottom unexposed face is lined with a low density refractory fibre blanket of a specified minimum thickness. This prevents excessive heat transfer through the bottom of the sample. The cone heater is lowered on a sliding assembly to a height of 25 mm above the leading sample surface.

  In the case of a vertically orientated test specimen, the above measures are adopted with the addition of a drip trough to catch molten material. It is inappropriate to test materials which predominantly melt.

  Because of the robust and necessarily heavy nature of the sample mounting, the load cell has to be designed to measure relatively heavy loads, up to several kilograms, while maintaining reasonable accuracy. A load cell of a capacity of 0.5 kg and accuracy 0.1 g is specified. Through the use of a mechanical tare the cell can be adjusted to measure loads in 0.5 kg ‘steps’ from 0.0 kg to 4.0 kg.

The load cell is protected to limit conductive and radiative heat transfer, from the mounting and heater respectively. Conductive resistance is provided by a 13 mm layer of low-medium density refractory material used as a spacer between the sample and pan. There are stainless steel spacer rods between the mounting and the load cell. These cooled by convection in ambient air further limit conductive heat transfer to the load cell.

Radiative heat flux is shielded by an additional layer of the refractory material. The cell should also be covered using an empty sample holder, during warm ups and between tests.

Some materials will intumesce upon heating. Some will come into contact with the heater before pilot ignition. Such materials should not be tested.

- <u>Instrumentation for Gas Analysis</u>. The gas train, as with the exhaust duct and hood, is basically a scaled down version of what is used in full-scale experiments. Figure 2, has been reproduced from Redfern (1990) and illustrates the train.
- The ring sampler, with the inlets on the downstream side, draws the gas sample into the analyser train. A soot filter removes the larger particles from the ‘smoke’ to prevent damage to the sensitive analysers. The sample is passed through a cold trap to cool down. The cooling causes the moisture component of the ambient air to condense. Thus the sample is now relatively cool and almost dry.

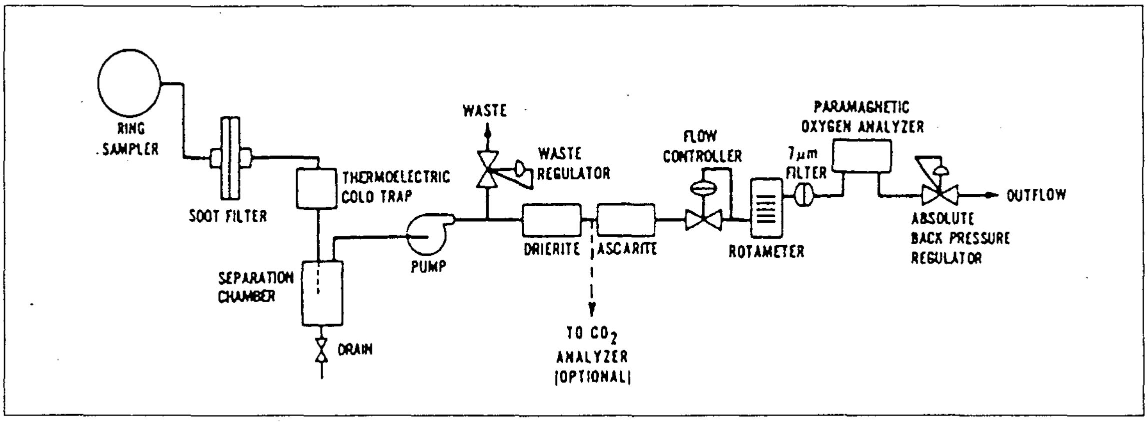

Figure 2: Gas Analyser Instrumentation Layout. Reproduced from Redfern (1990)

The gas sample is then drawn through a pump. After the pump a significant proportion of the sample is 'bled out' of the train. This results in an increased velocity for the remaining gases. Hence a reduction in the transport time lags downstream.

The oxides of carbon are now removed via chemical desiccants. This is mostly only carbon dioxide as carbon monoxide production in this open-to-the-air system is usually negligible. The only significant species present in the gas after this scrubbing are oxygen and nitrogen.

The gas sample is then passed through a flow regulator to avoid fluctuations in the analyser. A second moisture traps further 'dries' the sample. A second but finer filter (7 μm) protects the para-magnetic type oxygen analyser.

Downstream of the oxygen analyser an absolute back pressure regulator shields the system from varying ambient conditions.

Other gas analysers may be added to the train. The second most appropriate is a carbon dioxide analyser. Other possible analysers may include H_20 and TUHC (Total Unburnt Hydro-Carbon). These additions increase the accuracy of the heat release rate calculations. Suitable equations for various analyser configurations can be found in references such as Parker (1982) and Janssens (1991a).

Similar to a full-scale experiment situation, an oxygen analyser with the capability to measure in the range of 0 to 25 % is required. However unlike a full-scale experiment the oxygen analyser usually only measures in the narrow band of 18-21 %. Because measurements are being taken in this small range, experimental distortions such as 'noise' and 'drift' need to be minimised. The maximum allowable drift over 30 minutes is 50 ppm.

Minimum accuracy required for; the oxygen analyser, is to within at least 50 ppm, temperatures to within 0.5 ^{0}C and all other channels to within 0.01 % of full-scale instrument output. The system should be able to record data for a minimum of 60 minutes at a maximum interval of 5 s between readings.

- Other Associated Instrumentation. Complimentary to the above systems are:

(i) *Data Collection and Processing.* Measurements are made of the; elapsed time, oxygen analyser, differential pressure at the orifice plate, thermocouples, load cell and smoke measuring system. These must be collected and reduced into meaningful data. Collection and processing are done in separate passes. Software routines are available for this which produce the results in a report format.

(ii) *Calibration.* There are two phases to the calibration. Calibration of the analysing instruments and calibration of the heater's temperature control.

In the first instance a special methane burner using 99.9 % pure methane is utilised at a flow corresponding to 10 kW. Calibrating data is reported by Babrauskas (1982).

Secondly a water-cooled total heat flux gauge is mounted on the cone calorimeter. This can be manoeuvred to a position corresponding to the center of the exposed face of the sample. Heat flux calibrations are done before testing as required. The gauge is not used during testing.

The heat flux gauge is itself calibrated by a second but separate gauge of simular type and measuring range. This second calibrating gauge, which is used for no other purpose, is required to be annually tested at a standardising laboratory.

(iii) *Ignition Circuit and Timer.* A spark plug, powered by a 10 kV transformer, is positioned with the gap over the middle of the specimen. This provides pilot ignition of the sample. The spark plug is on a sliding mechanism and so can be removed from the fire plume after ignition. The circuitry is designed so as to minimise electromagnetism interference.

Time to sustained flaming is measured as ignition time. This is required to be accurate to within 1 s.

(iv) *Smoke Measurements*. Smoke measurement is in the form of obscuration, reported directly as an extinction coefficient. The system comprises of a small helium-neon laser beam projected across the exhaust duct and the associated electronics required to derive the extinction coefficient. It is located at a position downstream of the gas sampling train and before the exhaust fan. A section of this system is shown in figure 3.

The optical instruments are not protected from smoke particles via windows on the duct. Instead the suction in the duct is relied upon to purge the particles. As an extra precaution two small diameter tubes locate the two detector arrangements. It is anticipated that should any smoke penetrate into these tubes, then the particles would deposit upon the walls before reaching the sensitive instrumentation.

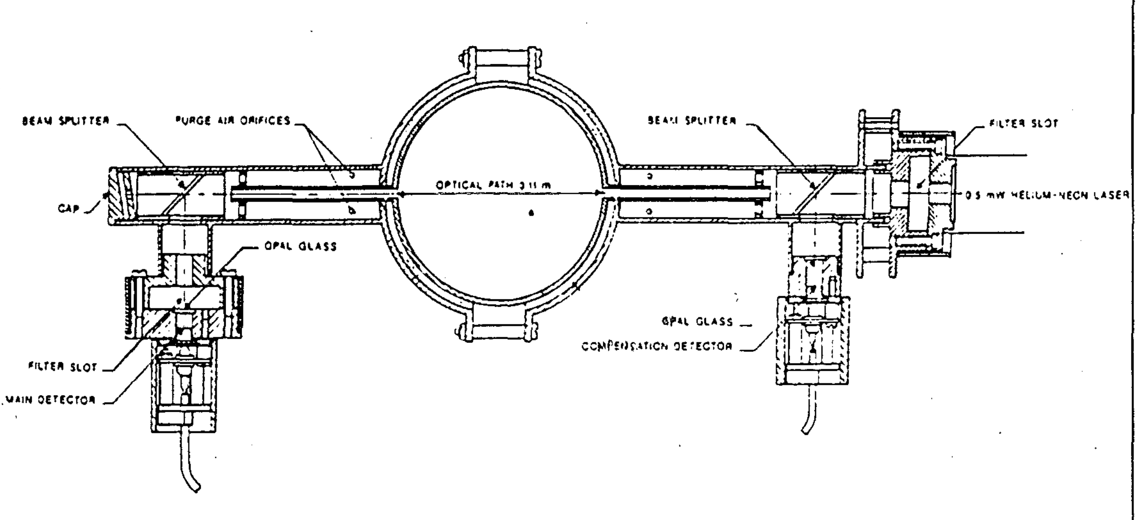

Figure 3: Smoke Obscuration Measuring System. Reproduced from ASTM E 1354.

For further details on the mechanics of the cone calorimeter, especially relating to; calibration, hazards of use, or operating procedures, refer to ASTM E 1534 and the accompanying publication by Twilley and Babrauskas (1988).

Theoretical Considerations for Analysis of Data

Individual material standards or performance standards may prescribe whatever calculations are required of the cone calorimeter data. The analysis will depend on which properties are of interest to that particular standard.

Common to all standards are; heat release rate, mass loss rate (hence effective heat of combustion), and smoke obscuration measurements. Therefore the relevant calculations are presented here.

Whilst it is not covered in this section, it is worth mentioning that properties measured from the cone calorimeter can also be used to predict flame spread characteristics. These considerations are discussed in the chapter titled "Measurement of Flame Spread Properties".

Heat Release Rate

The derivations of heat release rate equations, with respect to oxygen consumption calorimetry are covered in great detail elsewhere in this document. Therefore they are not repeated here. Instead only the final expression is reproduced, for a gas train analysing only oxygen,

$$\dot{q} = 1.10EC\sqrt{\frac{\Delta P}{T_e}}\left(\frac{0.2095 - X^A_{O_2}}{1.105 - 1.5X^A_{O_2}}\right)$$

where

$\dot{q}$ = Heat release rate (kW)

E = Hugett's constant (MJ/kg)

(= 13.1 MJ/kg for arbitrary fuels)

(= 12.54 MJ/kg for the calibration methane gas)

C = Orifice calibration constant (--)

ΔP = Orifice pressure differential (Pa)

T_e = Temperature of exhaust gases at orifice (K)

$X^A_{O_2}$ = Mole fraction of oxygen present in the analyser (--).

Also the net heat release rate can be described as

$$\dot{q}_{Net} = \frac{\dot{q}}{A}$$

where

A = Sample area (m^2)

It will be noted that the first expression is in a different form to the equation derived later for the full-scale heat release rate. This is due to various simplifying assumptions in the case of the cone calorimeter and also the absence of the oxygen reduction factor term, ϕ.

The reason for these inconsistencies is that where possible this document endeavours to use the same nomenclature as the relevant international standards. In the case of bench-scale and full-scale standards, there is not a consistency of nomenclature.

Mass Loss Rate and Effective Heat of Combustion

A five point numerical integration of the scanned data is used to calculate the effective heat of combustion from the mass loss rate *-dm/dt,* after each time interval '*i*'. For '*n*' number of scans there are five equations to be considered.

For the First Scan (i=0)

$$-\left(\frac{dm}{dt}\right)_{i=0} = \frac{25m_0 - 48m_1 + 36m_2 - 16m_3 + 3m_4}{12\Delta t}.$$

For the second scan (i=1)

$$-\left(\frac{dm}{dt}\right)_{i=1} = \frac{10m_0 + 3m_1 - 18m_2 + 6m_3 - m_4}{12\Delta t}.$$

For the ith scan ($1 < i < n$)

$$-\left(\frac{dm}{dt}\right)_{i} = \frac{-m_{i-2} + 8m_{i-1} - 8m_{i+1} + m_{i+2}}{12\Delta t}.$$

For the second to last scan ($i = n - 1$)

$$-\left(\frac{dm}{dt}\right)_{i=n-1} = \frac{-10m_n - 3m_{n-1} + 18m_{n-2} - 6m_{n-3} + m_{n-4}}{12\Delta t}.$$

For the last scan ($i = n$)

$$-\left(\frac{dm}{dt}\right)_{i=n} = \frac{-25m_n + 48m_{n-1} - 36m_{n-2} + 16m_{n-3} - 3m_{n-4}}{12\Delta t}.$$

A transient effective heat of combustion can be derived as,

$$\Delta h_{c,eff}(t) = \frac{\dot{q}_i(t)}{-\left(\frac{dm}{dt}\right)}.$$

Where

$\Delta h_{c,eff}(t)$= Transient effective heat of combustion of sample (MJ/kg)

$\dot{q}_i(t)$ = Instantaneous value of heat release rate (kW)

$-\left(\frac{dm}{dt}\right)$ = Mass loss rate (kg/s).

Or the average effective heat of combustion can be derived as,

$$\Delta h_{c,eff} = \frac{\sum_{i=0}^{n} \dot{q}_i(t)\Delta t_i}{m_i - m_f}.$$

Where

$\Delta h_{c,eff}$ = Average effective heat of combustion of sample (MJ/kg)

m_i = Initial mass of sample (kg)

m_f= Final mass of sample (kg).

Smoke Obscuration

Light passing through the exhaust gases is attenuated according to Lambert-Beer's law.

$$k = \frac{1}{l}\ln\frac{I_0}{I}$$

Where

k = Extinction coefficient (m^{-1})

l = Path length (m)

I_0 = Intensity of light at the source (cd)

I = Intensity of light transmitted over distance l (cd)

In the case of the cone calorimeter tests, the extinction coefficient and intensity of transmitted light will become transient. Consider Lambert-Beer's law as a time-dependant function.

$$k_i(t) = \frac{1}{l}\ln\frac{I_0}{I_i(t)}.$$

Where

$k_i(t)$ = Instantaneous value of transient extinction coefficient (m^{-1})

$I_i(t)$ = Instantaneous value of transient intensity of light (cd).

The extinction coefficient is not very useful as it depends on the exhaust duct flow rate. Instead we can first express smoke opacity in the form of a smoke release rate,

$$\dot{S} \approx k\dot{V}.$$

Where

$\dot{S}$ = Smoke release rate (m^2/s)

$\dot{V}$ = Volumetric flow rate of the exhaust gases at the smoke meter (m^3/s).

Then for correlations between bench-scale tests and full-scale experiments and in model validation, Babrauskas (1981) has suggested the use of a specific extinction area, σ_f. Defined as;

$$\sigma_f = \frac{k\dot{V}}{-\left(\frac{dm}{dt}\right)}.$$

Where

σ_f = Specific extinction area (m^2/kg).

An average specific extinction area is calculated over the test period as,

$$\sigma_f = \frac{\sum_{i=0}^{n} k_i \dot{V}_i \Delta t_i}{m_i - m_f}.$$

This concludes the description of the theoretical considerations in the measurement of; heat release rate, mass loss rate (hence effective heat of combustion), and smoke obscuration.

The Furniture Calorimeter: A Full-Scale Open Air Heat Release Rate Apparatus Using an Oxygen Consumption Calorimeter

Open air calorimetry on a full scale, what is commonly called furniture calorimetry, has been recently revived as an important fire hazard assessment method (Babrauskas 1992b). This is on the basis that while surface linings need to be tested in their end use configuration, discrete items such as furniture can be burned in the open air to obtain free burning data. This universal data can then be applied to other various design inputs such as computer fire growth models.

The furniture calorimeter is so called because of its early use in testing upholstered furniture. The primary example of a furniture calorimeter is the NBS Furniture Calorimeter shown in figure 4.

There is a Swedish standard furniture calorimeter testing method NT FIRE 032 (NORDTEST 1987). This standard is based on the original NBS Furniture Calorimeter. The only major difference is that NORDTEST specify a wooden crib ignition source and

NBS specify a gas burner ignition source. ASTM is considering adopting a furniture calorimeter standard which includes the gas burner ignition source.

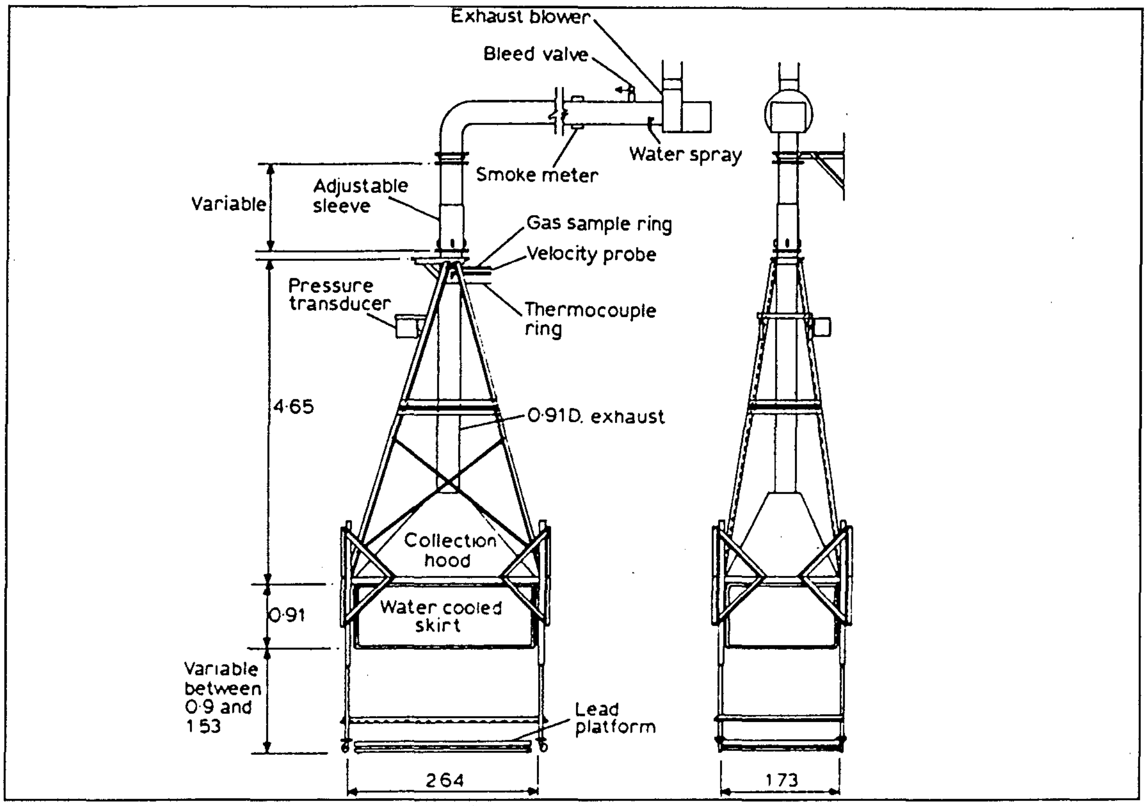

Figure 4: The NBS Furniture Calorimeter, reproduced from Babrauskas (1992b).

The exhaust system and gas analysis are essentially as described in the cone calorimeter, except obviously for the scale. For a more detailed description of this piece of apparatus one could refer to Cooper (1994).

An Intermediate-Scale Open Air Heat Release Rate Apparatus Using an Oxygen Consumption Calorimeter

In much of the experiments to date, emphasis has been on bench- and full-scale heat release rate methods. Shaw and Urbas (1993) describe a new, but relatively little-used, intermediate-scale experimental method known as ICAL, Figure 5.

Advantages of this system are that it offers a lower capital and operating cost and less required laboratory space. Further, the oxygen consumption calorimetry methodology and exhaust collection systems are the same as those currently being used for the ISO and ASTM full-scale room fire tests in many laboratories. Hence the data acquisition and inter-laboratory information transfer are relatively easy.

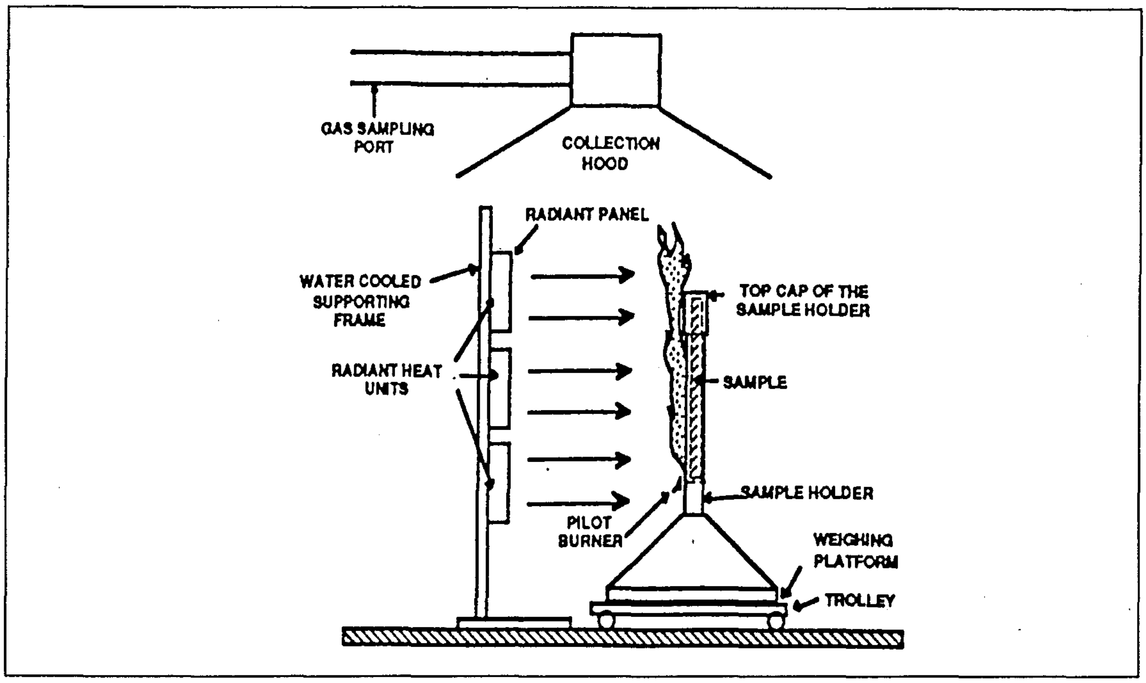

Figure 5: General Schematic of the ICAL apparatus. Shaw and Urbas (1993)

Scale Effects: Bench-Scale Tests to Predict Full-Scale Behaviour

The best possible test of building products such as lining materials, or a room item such as a piece of furniture, would be a full-scale room fire experiment. Due to cost and other considerations such as variations in room geometries, it is obviously not practical to carry out a large number of these experiments. It is therefore necessary to develop models which will predict full scale behaviour from bench scale results.

These models involve various input parameters relating to the materials and environment. These input parameters, coming from bench-scale experiments, need to be adjusted for scale effects where appropriate.

Analytical solutions on all but the simplest geometries are beyond our current skills. Future efforts will be more successful given the increasing knowledge of the fire physics and increasing computing power.

In the meanwhile a more empirical type of approach is taken. Babrauskas and Wickstrom (1989) suggest the following seven step methodology:

- Identify the governing physical and chemical principles of the phenomenon to be measured.
- Design a candidate bench-scale test using these principles.
- Identify the range, best to worst, of relevant full-scale product behaviours and assemble specimens having those traits.
- Assemble a data base by testing this range of specimens at full-scale and gather data using instruments appropriately designed to measure the governing physical and chemical phenomena.
- Conduct bench-scale tests, varying those features of fire behaviour which cannot be assigned to known constant values.
- Attempt to correlate the bench-scale results against the full-scale data base not only by ranking but also for quantitative values.
- Select those bench-scale test protocol features which lead to the best correlation with the full-scale data.

A classic example of this type of procedure was described earlier by Babrauskas (1983). A series of full-scale furniture calorimeter experiments were conducted on sofas and upholstered chairs, varying the; frame, filling and fabric materials. Specimens of filling/fabric composites were then tested on a bench-scale with the cone calorimeter.

A predictive model for the peak heat release rate was developed:

$$\dot{q}_{fs} = b.\dot{q}''_{bs}.[M].[FF].[SF]$$

Where

$\dot{q}_{fs}$ = Full-scale heat release rate (kW).

$\dot{q}''_{bs}$ = Bench-scale heat release rate per unit area (kW/m^2).

b = Scaling factor (m^2/kg).

$[M]$ = Mass of full-scale specimen excluding non-combustible parts (kg).

$[FF]$ = Frame factor (--).

$[SF]$ = Style factor (--).

The scaling factor b, was determined from a data correlation and found to be a best fit at $b = 0.63$ m^2/kg.

The frame factor [*FF*] was found to be:

1.66 for non-combustible frames.

0.58 for melting plastic frames.

0.30 for wood frames.

0.18 for charring plastic frames.

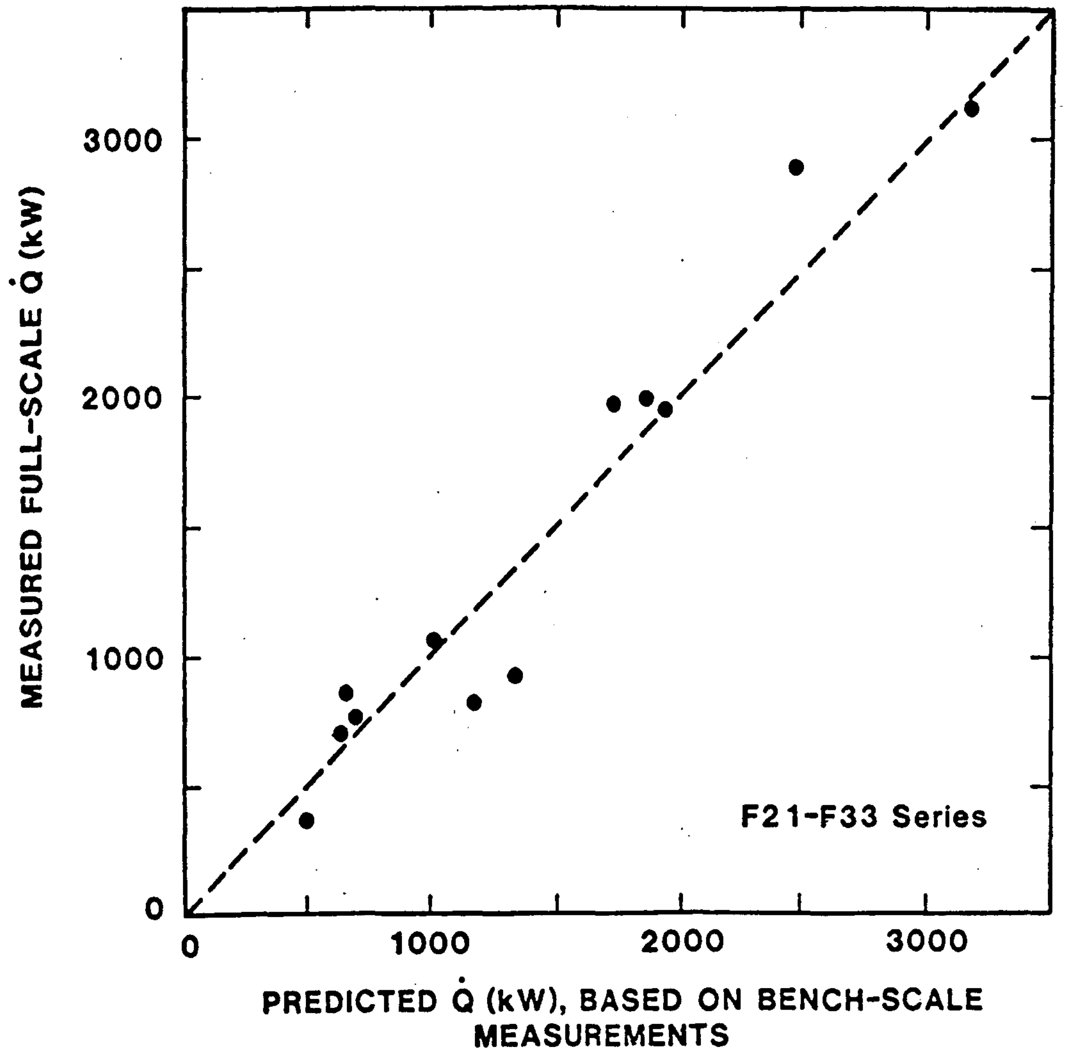

Figure 6: Comparison between peak full-scale heat release rate values for upholstered furniture measured in the furniture calorimeter and estimated values based on bench-scale cone calorimeter measurements.

The style factor is included to account for enhanced fire involvement of 'curvy' surfaces. The value is set between:

1.0 for rectilinear items and

1.5 for highly curvy ones.

A full description of the test specimens can be found in Babrauskas (1983). Figure 6 has been reproduced from this reference to illustrate the good correlation between full-scale peak heat release rates and the values predicted from the bench scale results.

More recently work has been done on a HEMFAST code (Dietenberger 1988) with the goal of improved quantitative furniture fire predictions from improved bench-scale techniques. This code was included in the FAST computer model for fire growth and smoke transport.

A more modern and detailed code, the Furniture Fire Model (FFM), also for use in FAST, is described in detail in Dietenberger (1992). This code uses bench-scale data to predict full-scale result for furniture 'items' fires. In many cases it is acceptable to trade off the level of accuracy obtained from the FFM code, which relies on state of the art equipment. For a simpler method, refer to the predicted-correlative approach proposed by Ames, Babrauskas and Parker (1992).

Chapter 4

Measurement of Heat Release Rate Using the Oxygen Consumption Technique

Introduction

By definition the primary purpose of a calorimeter is to measure heat. The question is; how is this done ?

Initially between the 1920s and 1970s in the first full-scale room fire experiments, the heat release rate was calculated from sensible enthalpy measurements of the combustion stream (Babrauskas 1991). There were numerous experiments recorded over this period, however the results lacked consistency and generality. There are systematic errors associated with this approach.

In the early 1980s the then new *principle of oxygen consumption* was applied to measure the heat release rate. This was done in experiments at the National Institute of Standards and Technology (NIST, formerly NBS) and the University of California, Berkeley. These experiments are considered to be the first to have successfully measured the heat release rate of a full-scale room fires.

The Principle of Oxygen Consumption

The principle of oxygen consumption is based on Thornton's rule. Thornton (1917) discovered the fact that for a large number of organic liquid and gas fires, a more or less constant amount of heat is released per unit mass of oxygen consumed during complete combustion.

Huggett (1980) established that this principle also applied to organic solids. He measured the constant as 13.1 MJ/kg of oxygen consumed, on average. Calculations using this figure yield values generally accurate to within 5%.

Therefore the heat release rate can be calculated as being linearly proportional to the amount of oxygen consumed. The amount of oxygen consumed being the difference between the oxygen concentration in the ambient air entering the apparatus and the oxygen concentration in the exhaust gases extracted from the system. Note that the measurement of oxygen concentration is time dependant.

Therefore

$$\dot{q} = E(\dot{m}^o_{O_2} - \dot{m}_{O_2}) \qquad (0)$$

where

$\dot{q}$ = Heat release rate (kW)

E = Heat released per unit mass of O_2 consumed (13.1 MJ/kg of O_2)

$\dot{m}^o_{O_2}$ = Mass flow rate of species O_2 in the incoming air (kg/s)

$\dot{m}_{O_2}$ = Mass flow rate of species O_2 in the exhaust gases (kg/s)

Please note that in this document as a whole, it was not considered necessary to number equations. However, this chapter is the exception to the rule.

Calculation of Heat Release Rate from Oxygen Consumption Measurements

Following are the derivations of equations to be used in the calculation of the heat release rate. These equations have been re-derived from the work of Parker (1982) and Janssens (1991a).

The major assumptions are:

- For unknown fuel composition, the amount of energy released by complete combustion per unit mass of O_2 is constant at 13.1 MJ/kg.
- Gases are assumed to behave ideally.
- Incoming air consists of O_2, CO_2, H_2O and N_2. All inerts not taking part in the combustion reaction are lumped into the nitrogen.
- O_2, CO_2 and CO are measured on a dry basis.

Several configurations of gas analysers are considered, accuracy is increased with the number of species measured.

Configuration 1: O_2 Gas Analysis

This is the simplest, cheapest and least accurate of the various analysis configurations. A sample of exhaust gases is drawn from the duct of the species present:

- The O_2 is measured by volume, para-magnetically.
- The N_2 by volume is calculated from the O_2 present.

But prior to this:

- The CO_2 is removed by a chemical sorbent.
- The H_2O is condensed out by cooling.
- The CO is neglected as being insignificant.

Note that only O_2 and N_2 enter the analyser and in a dry air state. This system is represented schematically in figure 1.

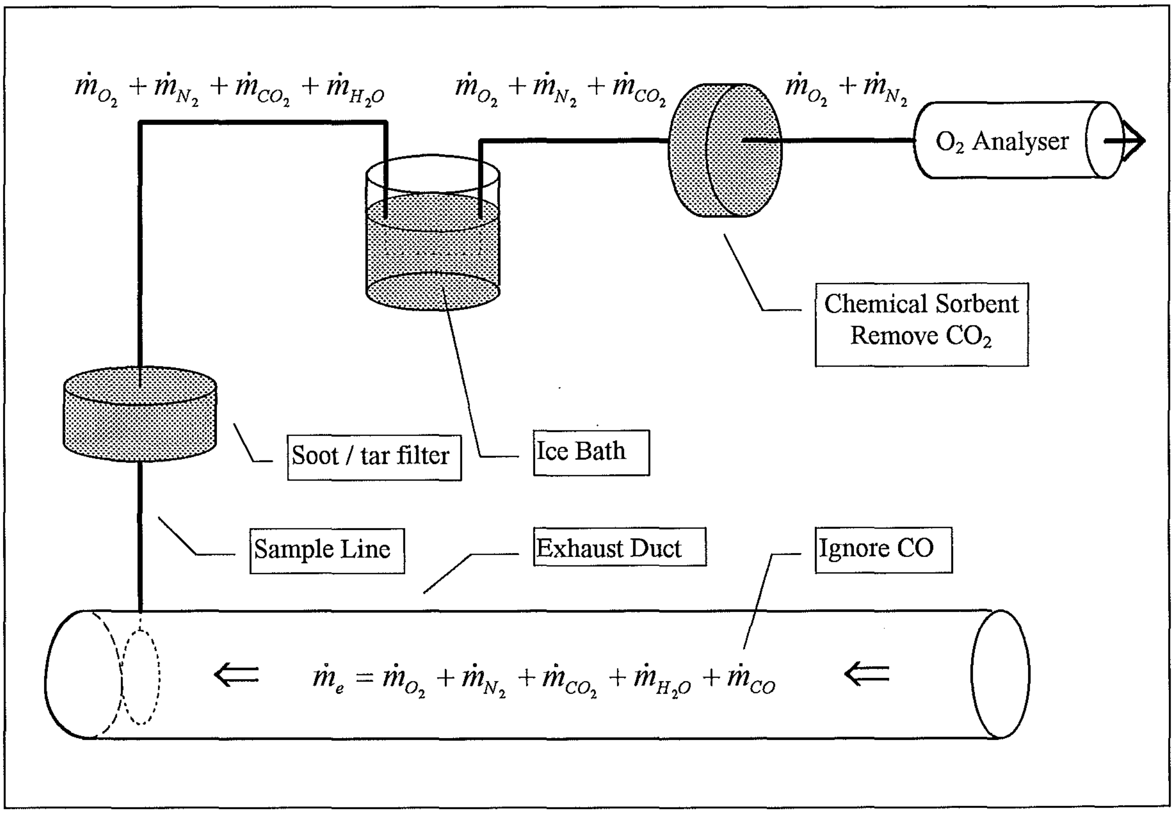

Figure 1: Idealisation of Gas Analysis Configuration 1.

Note that it is assumed that the mass flow rates of the species of products drawn into the analysis configuration, are the same as those in the exhaust duct.

We know that

$$X_i = \frac{n_i}{\sum n} \tag{1}$$

where

X_i = Mole fraction of species i (--)

n_i = Number of moles of species i (--)

$\sum n$ = Sum of moles of all species (--)

Given that

$$m_i = n_i M_i \tag{2}$$

where

m_i = Mass of species i (kg)

M_i = Molecular weight of species I (kg)

We can rearrange (2) and substitute into (1) for

$$X_i = \frac{\frac{m_i}{M_i}}{\sum \frac{m}{M}} \tag{3}$$

By considering the transient case (3) becomes

$$X_i = \frac{\frac{\dot{m}_i}{M_i}}{\sum \frac{\dot{m}}{M}} \tag{4}$$

where

$\dot{m}_i$ = Mass flow rate of species i (kg/s)

This can be rewritten for the measured mole fraction of O_2 prior to an experiment. Note that this differs from the mole fraction in the exhaust as we are measuring dry and without CO_2.

$$X_{O_2}^{A^o} = \frac{\dfrac{\dot{m}_{O_2}^o}{M_{O_2}}}{\dfrac{\dot{m}_{O_2}^o}{M_{O_2}} + \dfrac{\dot{m}_{N_2}^o}{M_{N_2}}} \tag{5}$$

Similarly during an experiment the mole fraction becomes transient and is described by:

$$X_{O_2}^{A} = \frac{\dfrac{\dot{m}_{O_2}}{M_{O_2}}}{\dfrac{\dot{m}_{O_2}}{M_{O_2}} + \dfrac{\dot{m}_{N_2}}{M_{N_2}}} \tag{6}$$

Rearranging (5) in terms of the mass flow rate $\dot{m}_{O_2}^o$:

$$X_{O_2}^{A^o} = \frac{\dfrac{\dot{m}_{O_2}^o}{M_{O_2}}}{\dfrac{\dot{m}_{O_2}^o}{M_{O_2}}\left(1 + \dfrac{\dot{m}_{N_2}^0}{M_{N_2}}\dfrac{M_{O_2}}{\dot{m}_{O_2}^o}\right)}$$

$$X_{O_2}^{A^o} = \frac{1}{\left(1 + \dfrac{\dot{m}_{N_2}^o}{M_{N_2}}\dfrac{M_{O_2}}{\dot{m}_{O_2}^o}\right)}$$

$$\frac{1}{X_{O_2}^{A^o}} = 1 + \frac{\dot{m}_{N_2}^o M_{O_2}}{M_{N_2}\dot{m}_{O_2}^o}$$

$$\frac{\dot{m}_{N_2}^o M_{O_2}}{M_{N_2}\dot{m}_{O_2}^o} = \frac{1}{X_{O_2}^{A^o}} - 1$$

$$\frac{\dot{m}^o_{O_2}}{M_{O_2}}\frac{M_{N_2}}{\dot{m}^o_{N_2}}=\frac{1}{\left(\frac{1}{X^{A^o}_{O_2}}-1\right)}$$

$$\dot{m}^o_{O_2}=\dot{m}^o_{N_2}\frac{M_{O_2}}{M_{N_2}}\frac{1}{\left(\frac{1}{X^{A^o}_{O_2}}-1\right)} \tag{7}$$

Similarly (6) rearranges to:

$$\dot{m}_{O_2}=\dot{m}_{N_2}\frac{M_{O_2}}{M_{N_2}}\frac{1}{\left(\frac{1}{X^{A}_{O_2}}-1\right)} \tag{8}$$

By subtracting (8) from (7) we get the mass loss rate of O₂:

$$\dot{m}^o_{O_2}-\dot{m}_{O_2}=\left\{\dot{m}^o_{N_2}\frac{M_{O_2}}{M_{N_2}}\frac{1}{\left(\frac{1}{X^{A^o}_{O_2}}-1\right)}\right\}-\left\{\dot{m}_{N_2}\frac{M_{O_2}}{M_{N_2}}\frac{1}{\left(\frac{1}{X^{A}_{O_2}}-1\right)}\right\} \tag{9}$$

The mass flow rate of N_2 is conserved

$$\dot{m}^o_{N_2}=\dot{m}_{N_2}$$

Therefore (9) becomes

$$\dot{m}^o_{O_2}-\dot{m}_{O_2}=\dot{m}_{N_2}\frac{M_{O_2}}{M_{N_2}}\left\{\frac{1}{\left(\frac{1}{X^{A^o}_{O_2}}-1\right)}-\frac{1}{\left(\frac{1}{X^{A}_{O_2}}-1\right)}\right\} \tag{10}$$

Simplifying (10)

$$\dot{m}^o_{O_2} - \dot{m}_{O_2} = \dot{m}_{N_2}\frac{M_{O_2}}{M_{N_2}}\left\{\frac{\left(\frac{1}{X^A_{O_2}}-1\right)-\left(\frac{1}{X^{A^o}_{O_2}}-1\right)}{\left(\frac{1}{X^{A^o}_{O_2}}-1\right)\left(\frac{1}{X^A_{O_2}}-1\right)}\right\}$$

$$\dot{m}^o_{O_2} - \dot{m}_{O_2} = \dot{m}_{N_2}\frac{M_{O_2}}{M_{N_2}}\left\{\frac{\frac{1}{X^A_{O_2}}-\frac{1}{X^{A^o}_{O_2}}}{\left(\frac{1}{X^{A^o}_{O_2}}-1\right)\left(\frac{1}{X^A_{O_2}}-1\right)}\right\}$$

$$\dot{m}^o_{O_2} - \dot{m}_{O_2} = \dot{m}_{N_2}\frac{M_{O_2}}{M_{N_2}}\left\{\frac{\frac{\left(X^{A^o}_{O_2}-X^A_{O_2}\right)}{X^{A^o}_{O_2}X^A_{O_2}}}{\left(\frac{1}{X^{A^o}_{O_2}}-\frac{X^{A^o}_{O_2}}{X^{A^o}_{O_2}}\right)\left(\frac{1}{X^A_{O_2}}-\frac{X^A_{O_2}}{X^A_{O_2}}\right)}\right\}$$

$$\dot{m}^o_{O_2} - \dot{m}_{O_2} = \dot{m}_{N_2}\frac{M_{O_2}}{M_{N_2}}\left\{\frac{\left(\frac{1}{X^{A^o}_{O_2}X^A_{O_2}}\right)\left(X^{A^o}_{O_2}-X^A_{O_2}\right)}{\left(\frac{1}{X^{A^o}_{O_2}X^A_{O_2}}\right)\left(1-X^{A^o}_{O_2}\right)\left(1-X^A_{O_2}\right)}\right\}$$

$$\dot{m}^o_{O_2} - \dot{m}_{O_2} = \dot{m}_{N_2}\frac{M_{O_2}}{M_{N_2}}\left\{\frac{\left(X^{A^o}_{O_2}-X^A_{O_2}\right)}{\left(1-X^{A^o}_{O_2}\right)\left(1-X^A_{O_2}\right)}\right\} \qquad (11)$$

Where

$$X^{A^o}_{N_2} = (1 - X^{A^o}_{O_2}) \qquad (12)$$

Substituting (12) into (11) we have

$$\dot{m}_{O_2}^{o} - \dot{m}_{O_2} = \dot{m}_{N_2}\frac{M_{O_2}}{M_{N_2}}\left(\frac{1}{X_{N_2}^{A^o}}\right)\left\{\frac{\left(X_{O_2}^{A^o} - X_{O_2}^{A}\right)}{\left(1 - X_{O_2}^{A}\right)}\right\} \tag{13}$$

and

$$X_{N_2}^{A^o} = \frac{\dfrac{\dot{m}_{N_2}^{o}}{M_{N_2}}}{\dfrac{\dot{m}_{O_2}^{o}}{M_{O_2}} + \dfrac{\dot{m}_{N_2}^{o}}{M_{N_2}}} \tag{14}$$

Invert (14) and substitute into (13) remembering N$_2$ is conserved

$$\dot{m}_{O_2}^{o} - \dot{m}_{O_2} = \dot{m}_{N_2}\frac{M_{O_2}}{M_{N_2}}\left(\frac{\dfrac{\dot{m}_{O_2}^{o}}{M_{O_2}} + \dfrac{\dot{m}_{N_2}^{o}}{M_{N_2}}}{\dfrac{\dot{m}_{N_2}^{o}}{M_{N_2}}}\right)\left\{\frac{\left(X_{O_2}^{A^o} - X_{O_2}^{A}\right)}{\left(1 - X_{O_2}^{A}\right)}\right\}$$

Simplifying

$$\dot{m}_{O_2}^{o} - \dot{m}_{O_2} = M_{O_2}\left(\frac{\dot{m}_{O_2}^{o}}{M_{O_2}} + \frac{\dot{m}_{N_2}^{o}}{M_{N_2}}\right)\left\{\frac{\left(X_{O_2}^{A^o} - X_{O_2}^{A}\right)}{\left(1 - X_{O_2}^{A}\right)}\right\} \tag{15}$$

(15) can be related to the total incoming mass rate where

$$\frac{\left(\dfrac{\dot{m}_{O_2}^{o}}{M_{O_2}} + \dfrac{\dot{m}_{N_2}^{o}}{M_{N_2}}\right)}{\dfrac{\dot{m}_{a}}{M_{a}}} = X_{O_2}^{A^o} + X_{N_2}^{A^o} \; . \tag{16}$$

Balancing the mole fractions of the incoming air

$$1 = X_{O_2}^{A^o} + X_{N_2}^{A^o} + X_{H_2O}^{o} + X_{CO_2}^{o}$$

Therefore

$$X_{O_2}^{A^o} + X_{N_2}^{A^o} = 1 - X_{H_2O}^{o} - X_{CO_2}^{o} \tag{17}$$

Substituting (17) into (16)

$$\frac{\left(\frac{\dot{m}_{O_2}^{o}}{M_{O_2}} + \frac{\dot{m}_{N_2}^{o}}{M_{N_2}}\right)}{\frac{\dot{m}_a}{M_a}} = 1 - X_{H_2O}^{o} - X_{CO_2}^{o}$$

$$\left(\frac{\dot{m}_{O_2}^{o}}{M_{O_2}} + \frac{\dot{m}_{N_2}^{o}}{M_{N_2}}\right) = \frac{\dot{m}_a}{M_a}\left(1 - X_{H_2O}^{o} - X_{CO_2}^{o}\right) \tag{18}$$

Substituting (18) into (15)

$$\dot{m}_{O_2}^{o} - \dot{m}_{O_2} = M_{O_2}\frac{\dot{m}_a}{M_a}\left(1 - X_{H_2O}^{o} - X_{CO_2}^{o}\right)\left\{\frac{\left(X_{O_2}^{A^o} - X_{O_2}^{A}\right)}{\left(1 - X_{O_2}^{A}\right)}\right\} \tag{19}$$

where

$$\dot{q} = E(\dot{m}_{O_2}^{o} - \dot{m}_{O_2})$$

(19) becomes

$$\dot{q} = E\left\{\frac{\left(X_{O_2}^{A^o} - X_{O_2}^{A}\right)}{\left(1 - X_{O_2}^{A}\right)}\right\}M_{O_2}\frac{\dot{m}_a}{M_a}\left(1 - X_{CO_2}^{o} - X_{H_2O}^{o}\right) \tag{20}$$

In this configuration no CO_2 analyser is present. As ambient air is being drawn prior to the experiment, $X^o_{CO_2}$ can be assumed to be constant and negligible at approximately 330 ppm.

There is not a water vapour analyser either. However $X^o_{H_2O}$ is significant and therefore must be measured or calculated.

$X^o_{H_2O}$ can be measured experimentally by bypassing the water vapour trap prior to an experiment and making a comparison of the change in the $X^o_{O_2}$ values.

$$\left(X^{A^o}_{O_2}\right)_{without\_trap} = (1 - X^o_{H_2O})\left(X^{A^o}_{O_2}\right)_{with\_trap}$$

However this is not a desirable option as it involves putting a hot wet sample through the chemical sorbent and O_2 analyser.

Alternatively $X^o_{H_2O}$ can be calculated from the relative humidity of the ambient air:

$$RH = \frac{X_{H_2O}}{\left(X_{H_2O}\right)_{Saturation}} \quad \text{at some constant temperature.}$$

Assuming ideal gas behaviour we can make the simplification

$$RH = \frac{P_a}{\left(P_a\right)_{Saturation}} \quad \text{expressed as a percentage}$$

where

P_a = Pressure of the ambient air (Pa)

Therefore

$$X^o_{H_2O} = \frac{RH}{100}\frac{P_s(T_a)}{P_a}$$

where

$P_s(T_a)$ = Saturation pressure of water vapour at T_a (Pa)

Thus far we can calculate or measure each variable in equation (20) other than the mass flow rate of the incoming air, $\dot{m}_a$.

As is discussed in the following chapter on mass flow rate, we will be measuring the exhaust gases mass flow rate, $\dot{m}_e$ and we therefore need to convert $\dot{m}_a$ in (20) to $\dot{m}_e$.

To simplify matters consider ϕ, the oxygen depletion factor. The oxygen depletion factor is the fraction of incoming air that is fully depleted of it's oxygen during the combustion process.

$$\phi = \frac{\dot{m}^o_{O_2} - \dot{m}_{O_2}}{\dot{m}^o_{O_2}} \tag{21}$$

Substituting (7) and (8) into (21)

$$\phi = \frac{\left\{ \dot{m}^o_{N_2} \dfrac{M_{O_2}}{M_{N_2}} \dfrac{1}{\left(\dfrac{1}{X^{A^o}_{O_2}} - 1 \right)} \right\} - \left\{ \dot{m}_{N_2} \dfrac{M_{O_2}}{M_{N_2}} \dfrac{1}{\left(\dfrac{1}{X^{A}_{O_2}} - 1 \right)} \right\}}{\left\{ \dot{m}^o_{N_2} \dfrac{M_{O_2}}{M_{N_2}} \dfrac{1}{\left(\dfrac{1}{X^{A^o}_{O_2}} - 1 \right)} \right\}}$$

Simplifying

$$\phi = \frac{\dfrac{1}{\left(\dfrac{1}{X_{O_2}^{A^o}} - \dfrac{X_{O_2}^{A^o}}{X_{O_2}^{A^o}}\right)} - \dfrac{1}{\left(\dfrac{1}{X_{O_2}^{A}} - \dfrac{X_{O_2}^{A}}{X_{O_2}^{A}}\right)}}{\dfrac{1}{\left(\dfrac{1}{X_{O_2}^{A^o}} - \dfrac{X_{O_2}^{A^o}}{X_{O_2}^{A^o}}\right)}}$$

$$\phi = \frac{\left(\dfrac{X_{O_2}^{A^o}}{1 - X_{O_2}^{A^o}}\right) + \left(\dfrac{X_{O_2}^{A}}{1 - X_{O_2}^{A}}\right)}{\left(\dfrac{X_{O_2}^{A^o}}{1 - X_{O_2}^{A^o}}\right)}$$

$$\phi = \frac{\left\langle \dfrac{\left\{X_{O_2}^{A^o}\left(1 - X_{O_2}^{A}\right)\right\} - \left\{X_{O_2}^{A}\left(1 - X_{O_2}^{A^o}\right)\right\}}{\left(1 - X_{O_2}^{A^o}\right)\left(1 - X_{O_2}^{A}\right)} \right\rangle}{\left(\dfrac{X_{O_2}^{A^o}}{1 - X_{O_2}^{A^o}}\right)}$$

$$\phi = \frac{\left(X_{O_2}^{A^o} - X_{O_2}^{A^o}X_{O_2}^{A} - X_{O_2}^{A} + X_{O_2}^{A^o}X_{O_2}^{A}\right)}{X_{O_2}^{A^o}\left(1 - X_{O_2}^{A}\right)}$$

$$\phi = \frac{\left(X_{O_2}^{A^o} - X_{O_2}^{A}\right)}{X_{O_2}^{A^o}\left(1 - X_{O_2}^{A}\right)} \qquad (22)$$

Express (22) in terms of moles (mass)

The number of moles of species' in the exhaust at any instant in time is the sum of the number of moles of incoming air not depleted of O_2 , plus the product of the number of moles of incoming air depleted of O_2 multiplied by some expansion factor due to the combustion chemistry.

$$n_e = \left(n_{air\_not\_O_2\_depleted}\right) + \left(n_{air\_O_2\_depleted}\right)\alpha$$

where

n = number of moles (--)

α = expansion factor (--)

The expansion factor, α, is the ratio of the number of moles of combustion products replacing the depleted O_2, to the number of moles of the fraction of incoming air depleted of O_2.

Often as the fuel composition is unknown an average value for α of 1.105 is used. This is also the correct expansion factor for methane.

Therefore

$$n_e = (1-\phi).n_a + \alpha.\phi.n_a$$

Convert to mass flow rates

$$\frac{\dot{m}_e}{M_e} = (1-\phi)\frac{\dot{m}_a}{M_a} + \alpha.\phi.\frac{\dot{m}_a}{M_a}$$

Assume $M_e \approx M_a$ and cancel

$$\dot{m}_e = (1-\phi)\dot{m}_a + \alpha.\phi.\dot{m}_a$$

Rearrange for $\dot{m}_a$

$$\dot{m}_a = \frac{\dot{m}_e}{(1-\phi)+\alpha.\phi}$$

$$\dot{m}_a = \frac{\dot{m}_e}{1+\phi.(\alpha-1)} \qquad (23)$$

Substitute (23) into (20)

$$\dot{q} = E\left\{\frac{\left(X_{O_2}^{A^o} - X_{O_2}^{A}\right)}{\left(1 - X_{O_2}^{A}\right)}\right\}\left(\frac{M_{O_2}}{M_a}\right)\left(\frac{\dot{m}_e}{1+\phi.(\alpha-1)}\right)\left(1 - X_{CO_2}^{o} - X_{H_2O}^{o}\right) \qquad (24)$$

Simplify the O_2 mole fractions in terms of the oxygen depletion factor

$$\dot{q} = E\phi.X_{O_2}^{A^o}\frac{M_{O_2}}{M_a}\frac{\dot{m}_e}{\left(1+\phi.(\alpha-1)\right)}\left(1 - X_{CO_2}^{o} - X_{H_2O}^{o}\right) \qquad (25)$$

Equation (25) will be accurate to within +/-10%. However it assumes complete combustion, meaning it assumes that all CO is converted to CO_2. The results will become less accurate in circumstances where CO production is significant, such as in post-flashover ventilation limited compartment fires.

Configuration 2: O_2 and CO_2 Gas Analysis

In this configuration a CO_2 analyser is added to the system described above. The water vapour is still removed from the sample being analysed and the CO is still neglected.

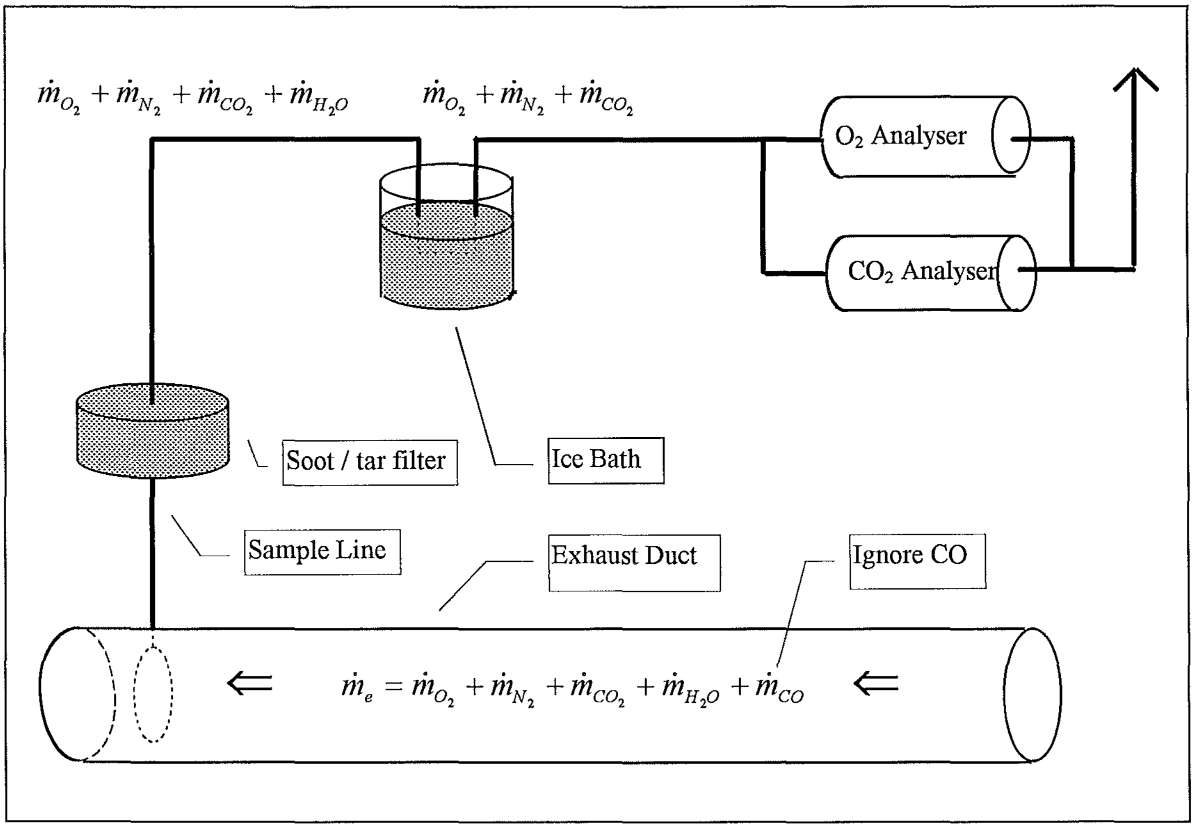

Figure 2: Idealisation of Gas Analysis Configuration 2.

Now (5) and (6) are respectively

$$X_{O_2}^{A^o} = \frac{\dfrac{\dot{m}_{O_2}^o}{M_{O_2}}}{\dfrac{\dot{m}_{O_2}^o}{M_{O_2}} + \dfrac{\dot{m}_{N_2}^o}{M_{N_2}} + \dfrac{\dot{m}_{CO_2}^o}{M_{CO_2}}} \tag{26}$$

and

$$X_{O_2}^{A} = \frac{\dfrac{\dot{m}_{O_2}}{M_{O_2}}}{\dfrac{\dot{m}_{O_2}}{M_{O_2}} + \dfrac{\dot{m}_{N_2}}{M_{N_2}} + \dfrac{\dot{m}_{CO_2}}{M_{CO_2}}} \tag{27}$$

Rearranging (26) for $\dot{m}_{O_2}^o$, where

$$X_{N_2}^{A^o} = 1 - X_{O_2}^{A^o} - X_{CO_2}^{A^o} = \frac{\dfrac{\dot{m}_{N_2}^o}{M_{N_2}}}{\dfrac{\dot{m}_{O_2}^o}{M_{O_2}} + \dfrac{\dot{m}_{N_2}^o}{M_{N_2}} + \dfrac{\dot{m}_{CO_2}^o}{M_{CO_2}}}$$

$$\frac{\dot{m}_{N_2}^o}{M_{N_2}}\left(\frac{1}{\left(1 - X_{O_2}^{A^o} - X_{CO_2}^{A^o}\right)}\right) = \frac{\dot{m}_{O_2}^o}{M_{O_2}} + \frac{\dot{m}_{N_2}^o}{M_{N_2}} + \frac{\dot{m}_{CO_2}^o}{M_{CO_2}}$$

Divide through by $\dfrac{\dot{m}_{O_2}^o}{M_{O_2}}$, where

$$\frac{M_{O_2}}{\dot{m}_{O_2}^o}\frac{\dot{m}_{N_2}^o}{M_{N_2}}\left(\frac{1}{\left(1 - X_{O_2}^{A^o} - X_{CO_2}^{A^o}\right)}\right) = \frac{1}{X_{O_2}^{A^o}}$$

Therefore

$$\dot{m}^{o}_{O_2} = \dot{m}^{o}_{N_2}\frac{M_{O_2}}{M_{N_2}}\left(\frac{X^{A^o}_{O_2}}{\left(1 - X^{A^o}_{O_2} - X^{A^o}_{CO_2}\right)}\right) \tag{28}$$

Similarly rearranging (27)

$$\dot{m}_{O_2} = \dot{m}_{N_2}\frac{M_{O_2}}{M_{N_2}}\left(\frac{X^{A}_{O_2}}{\left(1 - X^{A}_{O_2} - X^{A}_{CO_2}\right)}\right) \tag{29}$$

The expression for the oxygen depletion factor ϕ, is

$$\phi = \frac{\dot{m}^{o}_{O_2} - \dot{m}_{O_2}}{\dot{m}^{o}_{O_2}} \tag{29}$$

Substituting in (28) and (29)

$$\phi = \frac{\dot{m}^{o}_{N_2}\frac{M_{O_2}}{M_{N_2}}\left(\frac{X^{A^o}_{O_2}}{\left(1 - X^{A^o}_{O_2} - X^{A^o}_{CO_2}\right)}\right) - \dot{m}_{N_2}\frac{M_{O_2}}{M_{N_2}}\left(\frac{X^{A}_{O_2}}{\left(1 - X^{A}_{O_2} - X^{A}_{CO_2}\right)}\right)}{\dot{m}^{o}_{N_2}\frac{M_{O_2}}{M_{N_2}}\left(\frac{X^{A^o}_{O_2}}{\left(1 - X^{A^o}_{O_2} - X^{A^o}_{CO_2}\right)}\right)}$$

Where we assume $\dot{m}^{o}_{N_2} = \dot{m}_{N_2}$, therefore simplify

$$\phi = \frac{\left(\frac{X^{A^o}_{O_2}}{\left(1 - X^{A^o}_{O_2} - X^{A^o}_{CO_2}\right)}\right) - \left(\frac{X^{A}_{O_2}}{\left(1 - X^{A}_{O_2} - X^{A}_{CO_2}\right)}\right)}{\left(\frac{X^{A^o}_{O_2}}{\left(1 - X^{A^o}_{O_2} - X^{A^o}_{CO_2}\right)}\right)}$$

$$\phi = \frac{\left(\dfrac{X_{O_2}^{A^o}\left(1 - X_{O_2}^{A} - X_{CO_2}^{A}\right) - X_{O_2}^{A}\left(1 - X_{O_2}^{A^o} - X_{CO_2}^{A^o}\right)}{\left(1 - X_{O_2}^{A^o} - X_{CO_2}^{A^o}\right)\left(1 - X_{O_2}^{A} - X_{CO_2}^{A}\right)}\right)}{\left(\dfrac{X_{O_2}^{A^o}}{\left(1 - X_{O_2}^{A^o} - X_{CO_2}^{A^o}\right)}\right)}$$

$$\phi = \frac{X_{O_2}^{A^o}\left(1 - X_{CO_2}^{A}\right) - X_{O_2}^{A}\left(1 - X_{CO_2}^{A^o}\right)}{X_{O_2}^{A^o}\left(1 - X_{O_2}^{A} - X_{CO_2}^{A}\right)} \qquad (31)$$

Equation (31) gives the modified oxygen depletion factor for this configuration.

Equation (25) will hold using this factor after deleting the subtraction term of $X_{CO_2}^{A}$ in (25) which has been measured and allowed for in (31). Therefore (25) becomes:

$$\dot{q} = E\phi . X_{O_2}^{A^o} \frac{M_{O_2}}{M_a} \frac{\dot{m}_e}{1 + \phi .(\alpha - 1)}\left(1 - X_{H_2O}^{o}\right) \qquad (25A)$$

Equation (25A) will be accurate to within +/-10%. However it assumes complete combustion, meaning it assumes that all CO is converted to CO_2. The results will become less accurate in circumstances where CO production is significant such as in post-flashover ventilation limited compartment fires.

Configuration 3: O_2 , CO_2 and CO Gas Analysis

In this configuration a CO analyser is added to the system. The water vapour is still removed from the sample being analysed.

Now (5) and (6) are respectively

$$X_{O_2}^{A^o} = \frac{\dfrac{\dot{m}_{O_2}^{o}}{M_{O_2}}}{\dfrac{\dot{m}_{O_2}^{o}}{M_{O_2}} + \dfrac{\dot{m}_{N_2}^{o}}{M_{N_2}} + \dfrac{\dot{m}_{CO_2}^{o}}{M_{CO_2}} + \dfrac{\dot{m}_{CO}^{o}}{M_{CO}}} \qquad (32)$$

and

$$X_{O_2}^{A}=\frac{\dfrac{\dot{m}_{O_2}}{M_{O_2}}}{\dfrac{\dot{m}_{O_2}}{M_{O_2}}+\dfrac{\dot{m}_{N_2}}{M_{N_2}}+\dfrac{\dot{m}_{CO_2}}{M_{CO_2}}+\dfrac{\dot{m}_{CO}}{M_{CO}}} \tag{33}$$

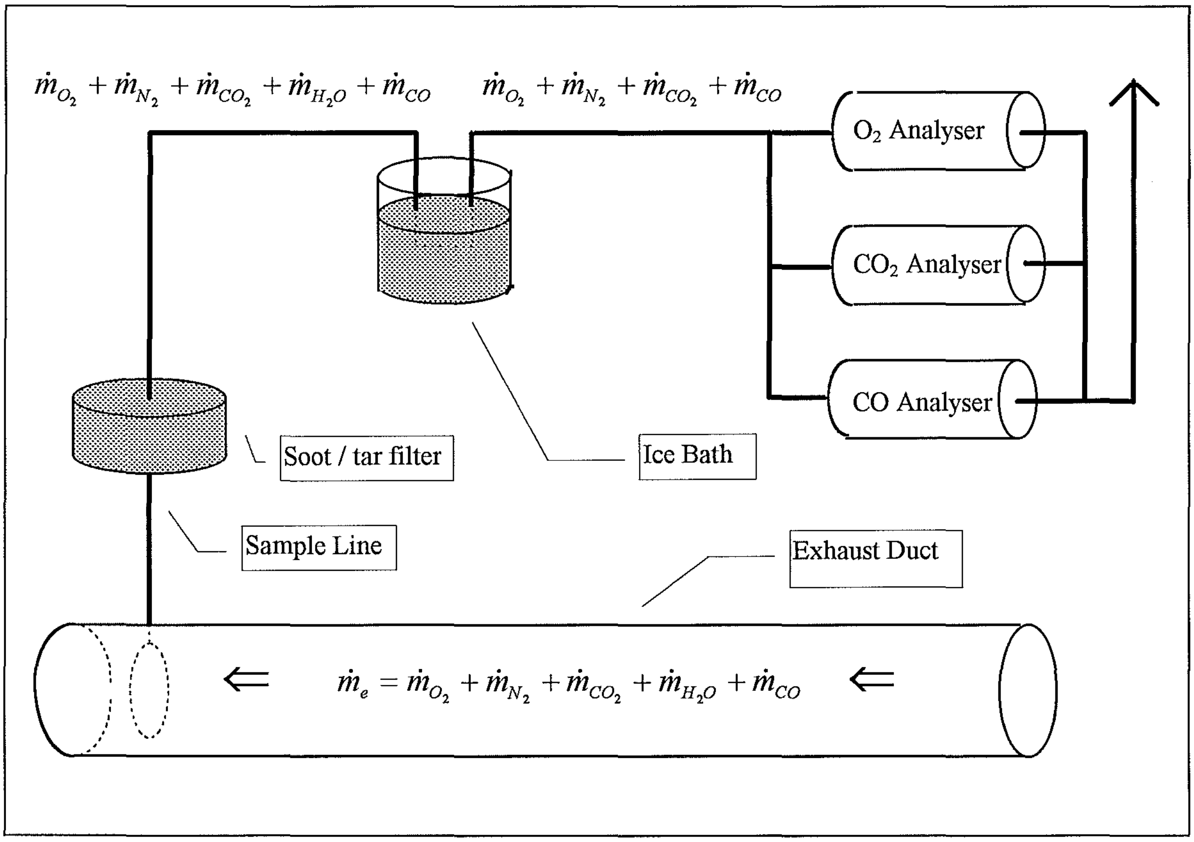

Figure 3: Idealisation of Gas Analysis Configuration 3.

Rearranging (32) for $\dot{m}_{O_2}^{o}$, where

$$X_{N_2}^{A^o}=1-X_{O_2}^{A^o}-X_{CO_2}^{A^o}-X_{N_2}^{A^o}=\frac{\dfrac{\dot{m}_{N_2}^{o}}{M_{N_2}}}{\dfrac{\dot{m}_{O_2}^{o}}{M_{O_2}}+\dfrac{\dot{m}_{N_2}^{o}}{M_{N_2}}+\dfrac{\dot{m}_{CO_2}^{o}}{M_{CO_2}}+\dfrac{\dot{m}_{CO}^{o}}{M_{CO}}}$$

$$\frac{\dot{m}^o_{N_2}}{M_{N_2}}\left(\frac{1}{\left(1-X^{A^o}_{O_2}-X^{A^o}_{CO_2}-X^{A^o}_{CO}\right)}\right)=\frac{\dot{m}^o_{O_2}}{M_{O_2}}+\frac{\dot{m}^o_{N_2}}{M_{N_2}}+\frac{\dot{m}^o_{CO_2}}{M_{CO_2}}+\frac{\dot{m}^o_{CO_2}}{M_{CO_2}}$$

Divide through by $\frac{\dot{m}^o_{O_2}}{M_{O_2}}$, where

$$\frac{M_{O_2}}{\dot{m}^o_{O_2}}\frac{\dot{m}^o_{N_2}}{M_{N_2}}\left(\frac{1}{\left(1-X^{A^o}_{O_2}-X^{A^o}_{CO_2}-X^{A^o}_{CO}\right)}\right)=\frac{1}{X^{A^o}_{O_2}}$$

Therefore

$$\dot{m}^o_{O_2}=\dot{m}^o_{N_2}\frac{M_{O_2}}{M_{N_2}}\left(\frac{X^{A^o}_{O_2}}{\left(1-X^{A^o}_{O_2}-X^{A^o}_{CO_2}-X^{A^o}_{CO}\right)}\right) \tag{34}$$

Similarly rearranging (33)

$$\dot{m}_{O_2}=\dot{m}_{N_2}\frac{M_{O_2}}{M_{N_2}}\left(\frac{X^{A}_{O_2}}{\left(1-X^{A}_{O_2}-X^{A}_{CO_2}-X^{A}_{CO}\right)}\right) \tag{35}$$

The expression for the oxygen depletion factor ϕ, is

$$\phi=\frac{\dot{m}^o_{O_2}-\dot{m}_{O_2}}{\dot{m}^o_{O_2}} \tag{36}$$

Substituting (34) and (35) into (36)

$$\phi=\frac{\dot{m}^o_{N_2}\frac{M_{O_2}}{M_{N_2}}\left(\frac{X^{A^o}_{O_2}}{\left(1-X^{A^o}_{O_2}-X^{A^o}_{CO_2}-X^{A^o}_{CO}\right)}\right)-\dot{m}_{N_2}\frac{M_{O_2}}{M_{N_2}}\left(\frac{X^{A}_{O_2}}{\left(1-X^{A}_{O_2}-X^{A}_{CO_2}-X^{A}_{CO}\right)}\right)}{\dot{m}^o_{N_2}\frac{M_{O_2}}{M_{N_2}}\left(\frac{X^{A^o}_{O_2}}{\left(1-X^{A^o}_{O_2}-X^{A^o}_{CO_2}-X^{A^o}_{CO}\right)}\right)}$$

Where we assume $\dot{m}_{N_2}^{o} = \dot{m}_{N_2}$, therefore simplify

$$\phi = \frac{\left(\frac{X_{O_2}^{A^o}}{\left(1 - X_{O_2}^{A^o} - X_{CO_2}^{A^o} - X_{CO}^{A^o}\right)}\right) - \left(\frac{X_{O_2}^{A}}{\left(1 - X_{O_2}^{A} - X_{CO_2}^{A} - X_{CO}^{A}\right)}\right)}{\left(\frac{X_{O_2}^{A^o}}{\left(1 - X_{O_2}^{A^o} - X_{CO_2}^{A^o} - X_{CO}^{A^o}\right)}\right)}$$

$$\phi = \frac{\left(\frac{X_{O_2}^{A^o}\left(1 - X_{O_2}^{A} - X_{CO_2}^{A} - X_{CO}^{A}\right) - X_{O_2}^{A}\left(1 - X_{O_2}^{A^o} - X_{CO_2}^{A^o} - X_{CO}^{A^o}\right)}{\left(1 - X_{O_2}^{A^o} - X_{CO_2}^{A^o} - X_{CO}^{A^o}\right)\left(1 - X_{O_2}^{A} - X_{CO_2}^{A} - X_{CO}^{A}\right)}\right)}{\left(\frac{X_{O_2}^{A^o}}{\left(1 - X_{O_2}^{A^o} - X_{CO_2}^{A^o} - X_{CO}^{A^o}\right)}\right)}$$

$$\phi = \frac{X_{O_2}^{A^o}\left(1 - X_{CO_2}^{A} - X_{CO}^{A}\right) - X_{O_2}^{A}\left(1 - X_{CO_2}^{A^o} - X_{CO}^{A^o}\right)}{X_{O_2}^{A^o}\left(1 - X_{O_2}^{A} - X_{CO_2}^{A} - X_{CO}^{A}\right)} \qquad (37)$$

Assuming $X_{CO}^{A^o}$ is negligible,

$$\phi = \frac{X_{O_2}^{A^o}\left(1 - X_{CO_2}^{A} - X_{CO}^{A}\right) - X_{O_2}^{A}\left(1 - X_{CO_2}^{A^o}\right)}{X_{O_2}^{A^o}\left(1 - X_{O_2}^{A} - X_{CO_2}^{A} - X_{CO}^{A}\right)} \qquad (38)$$

Equation (38) gives the modified oxygen depletion factor for this configuration.

Equation (25) will hold using this factor after deleting the subtraction term of $X_{CO_2}^{A}$ in (25) which has been measured and allowed for in (38). Therefore (25) becomes:

$$\dot{q} = E\phi . X_{O_2}^{A^o} \frac{M_{O_2}}{M_a} \frac{\dot{m}_e}{1 + \phi.(\alpha - 1)} \left(1 - X_{H_2O}^{o}\right) \qquad (25A)$$

Correction for Incomplete Combustion

If significant amounts of CO are produced then the value for E of 13.1 MJ/kg of O_2 consumed must be modified. This is because during incomplete combustion a significant fraction of the O_2 consumed, f, produces CO, which has not gone on to form CO_2.

There is a difference in the heat of formation of CO and CO_2 necessitating the following correction:

Let us consider the amount of O_2 that would have been required to form CO_2 from the CO present, $\Delta\dot{m}_{O_2}$.

From a mole balance we can say that

$$\Delta n_{O_2} = \frac{1}{2} n_{CO}$$

$$\frac{\Delta\dot{m}_{O_2}}{M_{O_2}} = \frac{1}{2}\frac{\dot{m}_{CO}}{M_{CO}} \qquad (39)$$

We know that

$$X_{O_2}^{A} = \frac{\dfrac{\dot{m}_{O_2}}{M_{O_2}}}{\dfrac{\dot{m}_{O_2}}{M_{O_2}} + \dfrac{\dot{m}_{N_2}}{M_{N_2}} + \dfrac{\dot{m}_{CO_2}}{M_{CO_2}} + \dfrac{\dot{m}_{CO}}{M_{CO}}}$$

and

$$X_{CO}^{A} = \frac{\dfrac{\dot{m}_{CO}}{M_{O_2}}}{\dfrac{\dot{m}_{O_2}}{M_{O_2}} + \dfrac{\dot{m}_{N_2}}{M_{N_2}} + \dfrac{\dot{m}_{CO_2}}{M_{CO_2}} + \dfrac{\dot{m}_{CO}}{M_{CO}}}$$

dividing

$$\frac{X^A_{O_2}}{X^A_{CO}} = \frac{\dfrac{\dot{m}_{CO}}{M_{CO}}}{\dfrac{\dot{m}_{O_2}}{M_{O_2}}} = \frac{\dot{m}_{CO} M_{O_2}}{M_{CO} \dot{m}_{O_2}}$$

$$\dot{m}_{O_2} = \dot{m}_{CO} \frac{M_{O_2}}{M_{CO}} \frac{X^A_{CO}}{X^A_{O_2}} \tag{40}$$

therefore substitute (39) into(40)

$$\Delta \dot{m}_{O_2} = \frac{1}{2} M_{O_2} \frac{X^A_{CO}}{X^A_{O_2}}$$

Where, at any instant in time, the number of moles of oxygen in the exhaust gases n_{O_2} can be related to the number of moles of air n_a via:

$$n_{O_2} = (1-\phi).n_a X^{A^o}_{O_2} \tag{41}$$

Where

$(1-\phi)$ = Unity less the oxygen depletion factor, ie: fraction of O_2 in ambient air not consumed in the combustion process.

Substituting (41) into (40) in terms of flow rates divided by molecular weights

$$\frac{\dot{m}_{O_2}}{M_{O_2}} = (1-\phi)\frac{\dot{m}_a}{M_a} X^{A^o}_{O_2}$$

$$\frac{\Delta \dot{m}_{O_2}}{M_{O_2}} = \frac{1}{2}\left\{(1-\phi)\frac{\dot{m}_a}{M_a} X^{A^o}_{O_2}\right\}\frac{X^A_{CO}}{X^A_{O_2}}$$

$$\Delta \dot{m}_{O_2} = \frac{1}{2}(1-\phi)\dot{m}_a \frac{M_{O_2}}{M_a} X^{A^o}_{O_2} \frac{X^A_{CO}}{X^A_{O_2}} \tag{42}$$

From (23)

$$\dot{m}_a = \frac{\dot{m}_e}{1+\phi.(\alpha-1)}$$

Substituting (23) into (42)

$$\Delta\dot{m}_{O_2} = \frac{1}{2}(1-\phi)\left(\frac{\dot{m}_e}{1+\phi.(\alpha-1)}\right)\frac{M_{O_2}}{M_a}X_{O_2}^{A^o}\frac{X_{CO}^A}{X_{O_2}^A} \qquad (43)$$

By substituting (43) into (0) and adding to (25A) with the modified oxygen depletion factor (38) we get a value for heat, that would have been released if combustion were complete.

$$\dot{q}_{Total} = E(\dot{m}^o{}_{O_2} - \dot{m}_{O_2}) + E(\Delta\dot{m}_{O_2}) \qquad (44)$$

Where

$\dot{q}_{Total}$ = Hypothetic heat release rate were combustion complete (kW)

We can correct (44) for the difference in heat of formation

$$\dot{q} = E(\dot{m}^o{}_{O_2} - \dot{m}_{O_2}) + E(\Delta\dot{m}_{O_2}) - E_{CO}(\Delta\dot{m}_{O_2}) \qquad (45)$$

Where

E_{CO} = Net heat released per unit mass of O_2 consumed for the combustion of CO to CO_2 (≅17.1 MJ/kg of O_2)

Therefore rewriting (25A) as corrected for CO production we have

$$\dot{q} = \left\{E\phi.-(E_{CO}-E)\left(\frac{(1-\phi)}{2}\frac{X_{CO}^A}{X_{O_2}^A}\right)\right\}\left(\frac{\dot{m}_e}{1+\phi.(\alpha-1)}\right)\frac{M_{O_2}}{M_a}X_{O_2}^{A^o}\left(1-X_{H_2O}^o\right)$$

(46)

Configuration 4: O_2, CO_2, CO and H_2O Gas Analysis

In this configuration a water vapour analyser is added to the system.

So that the gas analysers are not subject to the hot exhaust gas the H_2O sample is drawn separately from the duct as shown in figure 4.

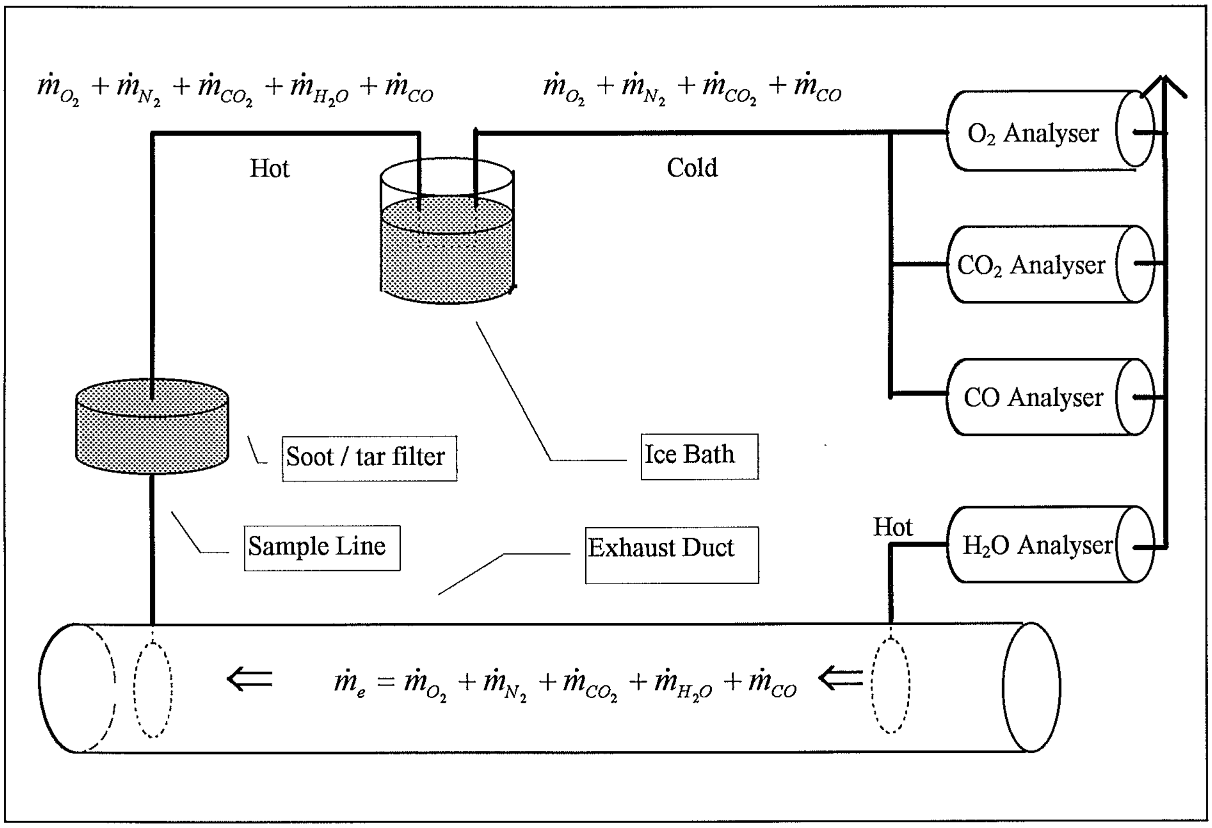

Figure 4: Idealisation of Gas Analysis Configuration 4.

Flow Rate Measurement Correction

In the previous configurations it was assumed that the density of the exhaust gases ρ_e was the same as the density of air. Measurement of the H_2O allows a further correction to be made based on the calculation of the molecular weight of the exhaust gases M_e.

Assuming that O_2, N_2, CO_2, CO and H_2O make up all of the exhaust gases in the full-scale experiment we can write equations for the mass flow rates of each species.

$$\frac{\dot{m}_{H_2O}}{M_{H_2O}} = X_{H_2O}\frac{\dot{m}_e}{M_e} \tag{47}$$

It follows that

$$\frac{\dot{m}_{O_2}}{M_{O_2}} = (1 - X_{H_2O})X^A_{O_2}\frac{\dot{m}_e}{M_e} \tag{48}$$

$$\frac{\dot{m}_{CO_2}}{M_{CO_2}} = (1 - X_{H_2O})X^A_{CO_2}\frac{\dot{m}_e}{M_e} \tag{49}$$

$$\frac{\dot{m}_{CO}}{M_{CO}} = (1 - X_{H_2O})X^A_{CO}\frac{\dot{m}_e}{M_e} \tag{50}$$

From the above equations (47) to (50) we can balance for N_2 where

$$\frac{\dot{m}_{N_2}}{M_{N_2}} = (1 - X_{H_2O})X^A_{N_2}\frac{\dot{m}_e}{M_e} \tag{51}$$

and

$$X^A_{N_2} = (1 - X^A_{O_2} - X^A_{CO_2} - X^A_{CO}) \tag{52}$$

Substituting (52) into (51)

$$\frac{\dot{m}_{N_2}}{M_{N_2}} = (1 - X_{H_2O})(1 - X^A_{O_2} - X^A_{CO_2} - X^A_{CO})\frac{\dot{m}_e}{M_e} \tag{53}$$

we know that

$$\dot{m}_e = \dot{m}_{O_2} + \dot{m}_{N_2} + \dot{m}_{CO_2} + \dot{m}_{CO} + \dot{m}_{H_2O} \tag{54}$$

Substituting (47) to (50) and (53) into (54) and simplifying

$$\dot{m}_e = (1 - X_{H_2O})X^A_{O_2}\frac{\dot{m}_e}{M_e}M_{O2} + (1 - X_{H_2O})(1 - X^A_{O_2} - X^A_{CO_2} - X^A_{CO})\frac{\dot{m}_e}{M_e}M_{N_2}$$
$$+(1 - X_{H_2O})X^A_{CO2}\frac{\dot{m}_e}{M_e}M_{CO_2} + (1 - X_{H_2O})X^A_{CO}\frac{\dot{m}_e}{M_e}M_{CO} + X_{H_2O}\frac{\dot{m}_e}{M_e}M_{H_2O}$$

$$\dot{m}_e = (1 - X_{H_2O})\frac{\dot{m}_e}{M_e}\left(\begin{array}{l} X^A_{O_2}M_{O2} \\ +(1 - X^A_{O_2} - X^A_{CO_2} - X^A_{CO})M_{N_2} \\ +X^A_{CO2}M_{CO_2} \\ +X^A_{CO}M_{CO} \end{array}\right) + X_{H_2O}\frac{\dot{m}_e}{M_e}M_{H_2O}$$

$$M_e = (1 - X_{H_2O})\left(\begin{array}{l} X^A_{O_2}M_{O2} \\ +(1 - X^A_{O_2} - X^A_{CO_2} - X^A_{CO})M_{N_2} \\ +X^A_{CO2}M_{CO_2} \\ +X^A_{CO}M_{CO} \end{array}\right) + X_{H_2O}M_{H_2O} \qquad (55)$$

Substituting in values for the molecular weights of the gas species, where

$M_{O_2} = 32$ kg/kmol

$M_{N_2} = 28$ kg/kmol

$M_{CO_2} = 44$ kg/kmol

$M_{CO} = 28$ kg/kmol

$M_{H_2O} = 18$ kg/kmol

Equation (55) simplifies to

$$M_e = (1 - X_{H_2O})\left(\begin{array}{l} 32X^A_{O_2} \\ +28(1 - X^A_{O_2} - X^A_{CO_2} - X^A_{CO}) \\ +44X^A_{CO2} \\ +28X^A_{CO} \end{array}\right) + 18X_{H_2O}$$

$$M_e = (1 - X_{H_2O})\left(28 + 4X^A_{O_2} + 16X^A_{CO2}\right) + 18X_{H_2O}$$

Expanding

$$M_e = 28 + 4X_{O_2}^A + 16X_{CO2}^A - 28X_{H_2O} - 4X_{O_2}^A X_{H_2O} - 16X_{CO2}^A X_{H_2O} + 18X_{H_2O}$$

Simplifying

$$M_e = 28 + 4X_{O_2}^A + 16X_{CO2}^A - 10X_{H_2O} - 4X_{O_2}^A X_{H_2O} - 16X_{CO2}^A X_{H_2O}$$

$$M_e = 28 - 10 + (4X_{O_2}^A + 16X_{CO2}^A - 10) - X_{H_2O}\left(4X_{O_2}^A + 16X_{CO2}^A + 10\right)$$

$$M_e = 18 + (1 - X_{H_2O})\left(4X_{O_2}^A + 16X_{CO2}^A + 10\right)$$

$$M_e = 18 + 4(1 - X_{H_2O})\left(X_{O_2}^A + 4X_{CO2}^A + 2.5\right) \qquad (56)$$

It will be shown in the following chapter "Mass Flow Rate of the Exhaust Gases" that:

$$\dot{m}_e = \frac{Ak_c}{f(\Re_e)}\sqrt{2p_e \Delta P_c}$$

where

$\dot{m}_e$ = The mass flow rate of the exhaust gases (kg/s)

A = Cross sectional area of the duct (m^2)

k_c = Shape factor (--)

ρ_e = Density of exhaust gas (kg/m^3). This is assumed to be numerically equal to the density of ambient air, unless the actual exhaust density can be calculated a in gas analysis.

ΔP_c = Pressure drop across a bidirectional probe located on the center-line of the duct (Pa).

$f(\Re_e)$ = Some function of the Reynolds number where the characteristic dimension is the inside diameter of the probe.

We can modify for the ambient density of air from the value of M_e via the ideal gas law where

$$\rho_e = \frac{M_{Dry}}{M_e}\rho$$

where

$M_{Dry} = 29$ kg/kmol

Therefore

$$\dot{m}_e = \frac{Ak_c}{f(\Re_e)}\sqrt{2\frac{M_{Dry}}{M_e}\rho.\Delta P} \tag{57}$$

Equation (57) gives a more accurate measurement $\dot{m}_e$. From this, and a measurements for the mole fractions of O_2, CO_2 and CO we can go on to relate $\dot{m}_e$ to $\dot{m}_a$.

<u>Relationship Between Mass Flow Rates</u>

Calculation of $\dot{m}_a$ in terms of $\dot{m}_e$ from a balance of all gases in the exhaust, will lead to a calculation of heat release rate independent of the expansion factor α, increased in accuracy.

From a N_2 balance we have from (53)

$$\frac{\dot{m}_{N_2}}{M_{N_2}} = (1 - X_{H_2O})(1 - X_{O_2}^{A} - X_{CO_2}^{A} - X_{CO}^{A})\frac{\dot{m}_e}{M_e} \tag{53}$$

Similarly

$$\frac{\dot{m}_{N_2}}{M_{N_2}} = (1 - X_{H_2O})(1 - X_{O_2}^{A^o} - X_{CO_2}^{A^o})\frac{\dot{m}_a}{M_a} \tag{58}$$

Therefore

$$(1 - X_{H_2O})(1 - X^A_{O_2} - X^A_{CO_2} - X^A_{CO})\frac{\dot{m}_e}{M_e} = (1 - X_{H_2O})(1 - X^{A^o}_{O_2} - X^{A^o}_{CO_2})\frac{\dot{m}_a}{M_a}$$

$$\frac{\dot{m}_a}{M_a} = \frac{(1 - X_{H_2O})(1 - X^A_{O_2} - X^A_{CO_2} - X^A_{CO})}{(1 - X^o_{H_2O})(1 - X^{A^o}_{O_2} - X^{A^o}_{CO_2})}\frac{\dot{m}_e}{M_e} \tag{59}$$

The molecular weight of the air, M_a, can be defined

$$M_a = M_{dry}\left(1 - X^o_{H_2O}\right) + M_{H_2O}X^o_{H_2O} \tag{60}$$

Calculation of the Oxygen Depletion Factor and Heat Release Rate

The mass flow rate of O_2 into the system can now be defined

$$\frac{\dot{m}^o_{O_2}}{M_{O_2}} = (1 - X^o_{H_2O})X^{A^o}_{O_2}\frac{\dot{m}_a}{M_a} \tag{61}$$

Substituting (59) into (61)

$$\frac{\dot{m}^o_{O_2}}{M_{O_2}} = (1 - X^o_{H_2O})X^{A^o}_{O_2}\frac{(1 - X_{H_2O})(1 - X^A_{O_2} - X^A_{CO_2} - X^A_{CO})}{(1 - X^o_{H_2O})(1 - X^{A^o}_{O_2} - X^{A^o}_{CO_2})}\frac{\dot{m}_e}{M_e}$$

$$\dot{m}^o_{O_2} = \frac{(1 - X_{H_2O})(1 - X^A_{O_2} - X^A_{CO_2} - X^A_{CO})}{(1 - X^{A^o}_{O_2} - X^{A^o}_{CO_2})}\dot{m}_e\frac{M_{O_2}}{M_e}X^{A^o}_{O_2} \tag{62}$$

From rearranging (48)

$$\dot{m}_{O_2} = (1 - X_{H_2O})\dot{m}_e\frac{M_{O_2}}{M_e}X^A_{O_2} \tag{63}$$

From (62) and (63) we can calculate the oxygen depletion factor

$$\phi = \frac{\left(\frac{(1-X_{H_2O})(1-X_{O_2}^{A}-X_{CO_2}^{A}-X_{CO}^{A})}{(1-X_{O_2}^{A^o}-X_{CO_2}^{A^o})}\dot{m}_e \frac{M_{O_2}}{M_e} X_{O_2}^{A^o}\right) - \left((1-X_{H_2O})\dot{m}_e \frac{M_{O_2}}{M_e} X_{O_2}^{A}\right)}{\left(\frac{(1-X_{H_2O})(1-X_{O_2}^{A}-X_{CO_2}^{A}-X_{CO}^{A})}{(1-X_{O_2}^{A^o}-X_{CO_2}^{A^o})}\dot{m}_e \frac{M_{O_2}}{M_e} X_{O_2}^{A^o}\right)}$$

Simplifying

$$\phi = \frac{\left(\frac{(1-X_{H_2O})(1-X_{O_2}^{A}-X_{CO_2}^{A}-X_{CO}^{A})}{(1-X_{O_2}^{A^o}-X_{CO_2}^{A^o})} X_{O_2}^{A^o}\right) - \left(\frac{(1-X_{O_2}^{A^o}-X_{CO_2}^{A^o})(1-X_{H_2O})X_{O_2}^{A}}{(1-X_{O_2}^{A^o}-X_{CO_2}^{A^o})}\right)}{\left(\frac{(1-X_{H_2O})(1-X_{O_2}^{A}-X_{CO_2}^{A}-X_{CO}^{A})}{(1-X_{O_2}^{A^o}-X_{CO_2}^{A^o})} X_{O_2}^{A^o}\right)}$$

$$\phi = \frac{(1-X_{O_2}^{A}-X_{CO_2}^{A}-X_{CO}^{A})X_{O_2}^{A^o} - (1-X_{O_2}^{A^o}-X_{CO_2}^{A^o})X_{O_2}^{A}}{(1-X_{O_2}^{A}-X_{CO_2}^{A}-X_{CO}^{A})X_{O_2}^{A^o}}$$

$$\phi = \frac{(1-X_{CO_2}^{A}-X_{CO}^{A})X_{O_2}^{A^o} - (1--X_{CO_2}^{A^o})X_{O_2}^{A}}{(1-X_{O_2}^{A}-X_{CO_2}^{A}-X_{CO}^{A})X_{O_2}^{A^o}} \qquad (64)$$

Where (64) is identical to (38), therefore (46) will hold but more accurately expressed in terms of $\dot{m}_a$, where

$$\dot{q} = \left\{E\phi .-(E_{CO}-E)\left(\frac{(1-\phi)}{2}\frac{X_{CO}^{A}}{X_{O_2}^{A}}\right)\right\}\dot{m}_a \frac{M_{O_2}}{M_a} X_{O_2}^{A^o}\left(1-X_{H_2O}^{o}\right). \qquad (65)$$

Equation (65) is the final form for this configuration with $\dot{m}_a / M_a$ given by (59)

Net Heat Release Rate

In full-scale room fire tests the ignition source, typically a gas burner or wood crib, will often contribute to oxygen consumption which may mask the heat release rate of a sample being tested. To this end a correction may be applied.

Consider a gas burner as an ignition source, assuming a combustion efficiency χ, of 100 % then:

$$\dot{q}_{ig} = \dot{m}_g (\Delta h_c)_g$$

where

$\dot{q}_{ig}$ = Heat release rate of the ignition source (kW)

$\dot{m}_g$ = Mass flow rate of gas to the burner (kg/s)

$(\Delta h_c)_g$ = Heat of combustion of the gas (MJ/kg)

From Thornton's rule

$$\dot{q}_{ig} = (\Delta \dot{m}_{O_2})_g E_g$$

where

$(\Delta \dot{m}_{O_2})_g$ = Change in the mass flow rate of oxygen in the exhaust gases due to the consumption of oxygen by the ignition source (kg/s)

E_g = Heat release associated with the consumption of one kg of O_2 when combustion occurs with gas "g". ie: numerically simular to *E*, Huggett's average (MJ/kg of O_2)

therefore

$$\dot{q}_{ig} = (\Delta \dot{m}_{O_2})_g E = \dot{m}_g (\Delta h_c)_g \frac{E}{E_g} \qquad (66)$$

We could subtract (66) from (65) to calculate the net heat release rate.

Unfortunately this procedure is not practicable during the tests due to; the non-linear time lags associated with the filling of the compartment and/or hood, the detection lags of the analysers.

Janssens (1991a) suggests that for both the gas burner described above and wood crib ignition sources (which are further complicated by the uncertainty in the Δh_c of the timber), the best procedure is to run a calibration test involving only the ignition source. This test will yield a "base-line" ignition source heat release rate curve, which can be subtracted from the gross heat release rate calculated from (65) to give the net heat release rate.

Summary of Calculation of Heat Release Rate

The following equations relate to those properties required to be calculated in determining the 'gross' heat release rate, that is to say the heat release rate inclusive of the contribution of the ignition source.

Configuration 1: O_2 Gas Analysis

$$\boxed{\dot{q} = E\phi . X_{O_2}^{A^o} \frac{M_{O_2}}{M_a} \frac{\dot{m}_e}{\left(1+\phi.(\alpha-1)\right)}\left(1 - X_{CO_2}^{o} - X_{H_2O}^{o}\right).}$$

Where

$$\phi = \frac{\left(X_{O_2}^{A^o} - X_{O_2}^{A}\right)}{X_{O_2}^{A^o}\left(1 - X_{O_2}^{A}\right)}$$

and

$$\dot{m}_e = \frac{Ak_c}{f(\Re_e)}\sqrt{2\frac{M_{Dry}}{M_e}\rho . \Delta P}\,.$$

Configuration 2: O_2 and CO_2 Gas Analysis

$$\boxed{\dot{q} = E\phi . X_{O_2}^{A^o} \frac{M_{O_2}}{M_a} \frac{\dot{m}_e}{1+\phi.(\alpha - 1)} \left(1 - X_{H_2O}^{o}\right).}$$

Where

$$\phi = \frac{X_{O_2}^{A^o}\left(1 - X_{CO_2}^{A}\right) - X_{O_2}^{A}\left(1 - X_{CO_2}^{A^o}\right)}{X_{O_2}^{A^o}\left(1 - X_{O_2}^{A} - X_{CO_2}^{A}\right)}$$

and

$$\dot{m}_e = \frac{Ak_c}{f(\Re_e)}\sqrt{2\frac{M_{Dry}}{M_e}\rho.\Delta P}\,.$$

Configuration 3: O_2 , CO_2 and CO Gas Analysis

$$\boxed{\dot{q} = \left\{E\phi.-(E_{CO} - E)\left(\frac{(1-\phi)}{2}\frac{X_{CO}^{A}}{X_{O_2}^{A}}\right)\right\}\left(\frac{\dot{m}_e}{1+\phi.(\alpha - 1)}\right)\frac{M_{O_2}}{M_a} X_{O_2}^{A^o}\left(1 - X_{H_2O}^{o}\right).}$$

Where

$$\phi = \frac{X_{O_2}^{A^o}\left(1 - X_{CO_2}^{A} - X_{CO}^{A}\right) - X_{O_2}^{A}\left(1 - X_{CO_2}^{A^o}\right)}{X_{O_2}^{A^o}\left(1 - X_{O_2}^{A} - X_{CO_2}^{A} - X_{CO}^{A}\right)}$$

and

$$\dot{m}_e = \frac{Ak_c}{f(\Re_e)}\sqrt{2\frac{M_{Dry}}{M_e}\rho.\Delta P}\,.$$

Configuration 4: O_2, CO_2, CO and H_2O Gas Analysis

$$\boxed{\dot{q} = \left\{E\phi.-(E_{CO} - E)\left(\frac{(1-\phi)}{2}\frac{X_{CO}^{A}}{X_{O_2}^{A}}\right)\right\}\dot{m}_a \frac{M_{O_2}}{M_a} X_{O_2}^{A^o}\left(1 - X_{H_2O}^{o}\right).}$$

Where

$$\phi = \frac{(1 - X^{A}_{CO_2} - X^{A}_{CO})X^{A^o}_{O_2} - (1 - -X^{A^o}_{CO_2})X^{A}_{O_2}}{(1 - X^{A}_{O_2} - X^{A}_{CO_2} - X^{A}_{CO})X^{A^o}_{O_2}}$$

and

$$\frac{\dot{m}_a}{M_a} = \frac{(1 - X_{H_2O})(1 - X^{A}_{O_2} - X^{A}_{CO_2} - X^{A}_{CO})}{(1 - X^{o}_{H_2O})(1 - X^{A^o}_{O_2} - X^{A^o}_{CO_2})}\frac{\dot{m}_e}{M_e}$$

and

$$\dot{m}_e = \frac{Ak_c}{f(\Re_e)}\sqrt{2\frac{M_{Dry}}{M_e}\rho.\Delta P}\,.$$

Chapter 5

Measurement of Mass Flow Rate of the Exhaust Gases

Introduction

Common to any gas analysis, such as that undertaken in oxygen consumption calorimetry, is a requirement to calculate the total mass flow rate of products, $\dot{m}_e$ in the exhaust duct. Consider the room calorimeter system shown in figure 1.

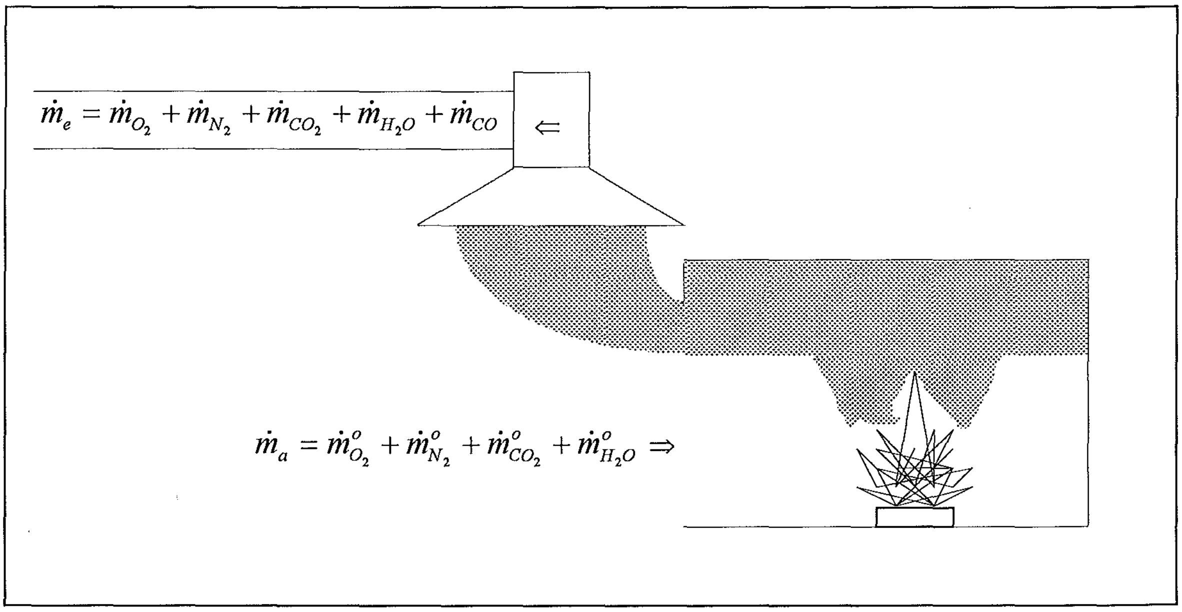

Figure 1: Schematic of Room Fire Experiment

Where

$\dot{m}_a$ = Mass flow rate of ambient air into the system (kg/s)

$\dot{m}^o_{O_2}$ = Mass flow rate of oxygen into the system (kg/s)

$\dot{m}^o_{N_2}$ = Mass flow rate of nitrogen into the system (kg/s)

$\dot{m}^o_{CO_2}$ = Mass flow rate of carbon dioxide into the system (kg/s)

$\dot{m}^o_{H_2O}$ = Mass flow rate of water vapour into the system (kg/s)

and

$\dot{m}_e$ = Mass flow rate of the exhaust gases (kg/s)

$\dot{m}_{O_2}$ = Mass flow rate of oxygen in the exhaust gases (kg/s)

$\dot{m}_{N_2}$ = Mass flow rate of nitrogen in the exhaust gases (kg/s)

$\dot{m}_{CO_2}$ = Mass flow rate of carbon dioxide in the exhaust gases (kg/s)

$\dot{m}_{H_2O}$ = Mass flow rate of water vapour in the exhaust gases (kg/s)

$\dot{m}_{CO}$ = Mass flow rate of carbon monoxide in the exhaust gases (kg/s)

Note that this system assumes that only O_2, N_2, CO_2, and H_20 are present in ambient air. All other inert gases are assumed to be lumped in with the N_2 and are insignificant.

Determination of Mass Flow Rate from Flow Velocity Measurements

The mass flow rate is the product of the volumetric flow rate and fluid density. Measurements of velocity profile are sought to calculate the volumetric flow rates.

There are two primary velocity measurement techniques:

(i) Measuring the pressure drop across an orifice plate, with say static pitot tubes, from which the centre line velocity can be calculated.

(ii) Measuring the center line differential pressure with a bi-directional probe.

Orifice Plate Technique

The flow rate can be calculated via measurements of pressure drop across an orifice plate by applying Bernoulli's equation.

This method is often associated with a visual reading of differential pressure via a manometer tube. In room-fire applications it is more desirable to have readings at fractions of a second intervals. Hence an electronic type pressure transducer is preferred. This leads us the second measurement technique which will be applied.

Center-line Velocity Technique

Currently the most appropriate technique for measuring the mass flow rate of exhaust gases $\dot{m}_e$, is to place a velocity measuring device along the center-line of the duct.

A schematic of the system is shown in figure 2.

Assuming fully developed flow, the shape factor k_c, which is the ratio of average velocity to center-line velocity, will remain constant over the length of the pipe. This principal will hold for any flow regime of a given Reynolds number.

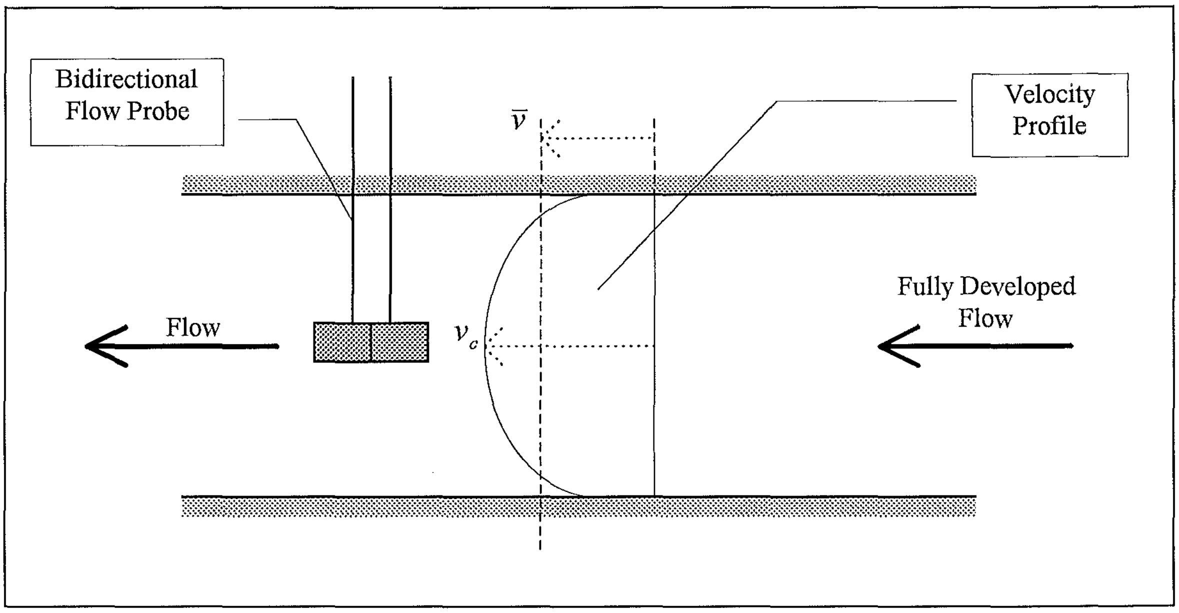

Figure 2: Velocity Profile

The shape factor is given by,

$$k_c = \frac{\bar{v}}{v_c}.$$

where

$k_c =$ Shape factor (--)

$\bar{v} =$ Average velocity (m/s)

$v_c =$ Center-line velocity (m/s)

By measuring the center-line velocity and having a predetermined shape factor (see later section) the average velocity can be calculated.

The average velocity is given by

$$\bar{v} = k_c v_c.$$

The volumetric flow rate can then be expressed as:

$$\dot{V} = Ak_c v_c$$

where

$\dot{V}$ = Volumetric flow rate (m^3/s)

A = Cross sectional area of the duct (m^2)

The volumetric flow rate is related to the exhaust mass flow rate $\dot{m}_e$, via Bernoulli's equation. So if the center-line velocity can be expressed as:

Ideally

$$v_c = \sqrt{\frac{2\Delta P_c}{\rho_e}}$$

where

ΔP_c = Pressure drop on the center-line of the duct (Pa)

ρ_e = Density of exhaust gas (kg/m^3)

then

$$\dot{V} = Ak_c v_c = Ak_c\sqrt{\frac{2\Delta P_c}{\rho_e}}$$

where

$$\dot{m}_e = \rho_e\dot{V}$$

therefore

$$\dot{m}_e = Ak_c\sqrt{2\rho_e\Delta P_c}$$

This equation is an expression for the mass flow rate of the exhaust gases in terms of a single center-line differential pressure measurement.

Determination of the Velocity Shape Factor for Fully Developed Flow

The exhaust gases are assumed to be an incompressible fluid. One flow property of such a fluid is that if fully developed it will maintain a constant velocity profile at all locations further downstream. This profile will remain the same for any other flow with the same Reynolds number.

However as the density and viscosity of the fluid have a temperature dependence, and as the temperature is transient, it holds that the Reynolds number of a gas will also be transient during a fire experiment.

Remembering that

$$\Re_e = \frac{\bar{v}\rho_e d}{\mu}$$

where

$\Re_e$ = The Reynolds number for that flow regime (--)
$\bar{v}$ = Mean velocity of the exhaust gas (m/s)
ρ_e= Density of exhaust gas (kg/m^3)
d = Characteristic dimension, in this case the inside diameter of the pipe (m)
μ = Viscosity of the gas/fluid (kg/m/s)

Example of Sensitivity of Reynolds Number to Varied Flow Temperature and Density

Consider a given pipe diameter and given mean velocity assume ambient conditions:

$$\rho_\infty = 1.17 \text{ kg/m}^3$$

$$\mu_\infty = 1.846 \times 10^{-5} \text{ kg/m/s}$$

(Likely extreme) fire exhaust gas conditions from CFAST modelling:

$\rho_e = 0.23$ kg/m^3

$\mu_e = 4.79 \times 10^{-5}$ kg/m/s

Therefore

$$\frac{(\Re_e)_\infty}{(\Re_e)_e} = \frac{\left(\dfrac{\bar{v}\rho_\infty d}{\mu_\infty}\right)}{\left(\dfrac{\bar{v}\rho_e d}{\mu_e}\right)} = \frac{\left(\dfrac{\rho_\infty}{\mu_\infty}\right)}{\left(\dfrac{\rho_e}{\mu_e}\right)} = \frac{\left(\dfrac{1.17}{1.846 \times 10^{-5}}\right)}{\left(\dfrac{0.23}{4.79 \times 10^{-5}}\right)} \approx 13.2$$

It appears that the Reynolds number for the flow of the fire gases in the exhaust duct may be of the order of 1/10 of the Reynolds number for the flow of ambient air.

In other words, during a room fire experiment the Reynolds number of flow in the duct may decrease by a factor of ten, from ambient, as the experiment progresses. As the profile is calibrated at ambient conditions the effect of this change in Reynolds number must be considered.

Schlicting (1976) summarises the findings of Nikuradse who carried out a thorough experimental investigation into the velocity profiles in smooth pipes for a wide range of Reynolds numbers. Unfortunately Nikuradse's work is not in English and therefore not referenced in this document.

It was noted that the profile became fuller as the Reynolds number increased, or in other words the center-line to average velocity ratio became closer to unity. It is possible to represent these profiles, from the boundary to the center-line with an empirical 1/nth-power-law equation.

$$\frac{v(y)}{v_c} = \left(\frac{y}{R}\right)^{\frac{1}{n_1}}$$

where

n_1 = A function of the Reynolds number (--)

The shape factor can be easily derived from this expression where

$$k_c = \frac{\bar{v}}{v_c} = \frac{2n^2}{(n+1)(2n+1)}$$

Nikuradse (Schlicting 1976) related n to the Reynolds number as follows:

$\Re_e$	4×10^3	2.3×10^4	1.1×10^5	1.1×10^6	2.0×10^6	3.2×10^6
n	6.0	6.6	7.0	8.8	10	10

Table 1: Value of the exponent 'n' of velocity distribution according to Equation 2, in terms of the Reynolds number of the pipe flow. Values from Schlicting (1976).

Solving for the velocity shape factor

$\Re_e$	4×10^3	2.3×10^4	1.1×10^5	1.1×10^6	2.0×10^6	3.2×10^6
k_c	.791	.807	.817	.850	.865	.865

Table 2: Velocity shape factor in terms of the Reynolds number of the pipe flow.

The change in the Reynolds number of the flow of fire exhaust gases, will at worst be in the order of $10^5 \rightarrow 10^4$. By observation of the above data we can neglect this as being insignificant.

So we may conclude that for fully developed turbulent flow in smooth pipes, empirical data such as Nikuradse's may be used to determine the velocity shape factor.

Determination of the Velocity Shape Factor for Non-fully Developed Flow

In the previous section it was concluded that it would be experimentally acceptable to determine the velocity shape factor from existing non-dimensional empirical data of turbulent flow in smooth pipes.

Unfortunately fully developed flow occurs at a pipe length, sufficiently far from the inlet so that it is impractical to construct it in laboratory conditions. This 'inlet length", for turbulent flow in smooth pipes, has again been empirically determined by Nikuradse (Schlicting 1976). Nikuradse has found it to be between 25 to 40 pipe diameters, depending on the Reynolds number of the flow.

Therefore it becomes necessary to develop a method for determining the velocity shape factor, for non-fully developed flow. This involves making a series of velocity or differential pressure measurements across the cross section of the duct to determine the mean velocity. From this the velocity shape factor can be calculated because as with the previous method, the center-line differential pressure and temperature will be continuously monitored.

This calibration to obtain the mean velocity at the preselected measuring point or points is done under various anticipated extract rates and at ambient conditions. It is assumed that, as has been shown with fully developed flow, the effect of a transient Reynolds number in non-fully developed flow is negligible.

Determination of the Overall Mean Velocity at a Pipe Section via Several Local Velocity Measurements

To determine the mean velocity, first consider a cross section through the circular exhaust duct. Consider also a small element of area, a_i within this section.

The volumetric flow rate through this area, $\dot{V}$, is expressed as

$$\dot{V} = A\bar{v} = \sum_{i=1}^{N} a_i \bar{v}_i$$

where

A = Duct cross sectional area (m^2)
$\bar{v}$ = Overall mean velocity of the exhaust gas (m/s)
N = Number of elements (--)
a_i = Area of element i (m^2)
$\bar{v}_i$ = Local mean velocity of the exhaust gas through area element i (m/s)

Therefore the mean velocity of the exhaust gas can be expressed as

$$\bar{v} = \frac{1}{A}\sum_{i=1}^{N} a_i \bar{v}_i$$

Suppose now that each of these N elements were of equal area, circular in the center of the duct and annular radiating outwards.

$$a = a_1 = a_2 = \ldots = a_i = \ldots = a_N$$

then

$$A = Na$$

therefore

$$\bar{v} = \frac{1}{Na}\sum_{i=1}^{N} a_i \bar{v}_i$$

which simplifies to

$$\bar{v} = \frac{\sum_{i=1}^{N} \bar{v}_i}{N}$$

In other words the overall mean velocity of the exhaust gas can be calculated from the averaging of the local mean velocity measurements in each incremental area.

For pipes of circular cross section this relationship can be expressed as:

$$\bar{v} = \frac{2}{R^2}\bar{v}_i \int_0^R r.dr$$

where

R = The pipe/duct radius (m)

r = The radial position at which the velocity $\bar{v}_i$, is measured

This integral is generally solved numerically by either the tangential or log-linear methods.

The solution yields the positions of the gauging points, in terms of y/D, at which the exact local mean velocities will occur. The log-linear method has been found by Winternitz and Fischl (1957) to be the more accurate method of the two as the tangential method tends to overestimate the overall mean velocity.

The Log-Linear Method

This method is based on the assumption that the velocity distribution of non-fully developed flow can often be represented by the equation:

$$v(y) = C_1 + C_2 \log\left(\frac{y}{D}\right) + C_3\left(\frac{y}{D}\right)$$

where

$v(y)$ = The velocity at point y, along the diameter (m)

C_1, C_2 ,C_3 = Numerical constants (m/s)

D = Diameter (m)

Given that this equation holds, and it has been found to do so by Ross and Robertson (1951), then the positions at which the mean local velocity would occur can be determined accordingly. This calculation is somewhat involved and is not reproduced here. However the derivation is appended to Winternitz and Fischl (1957).

Results for some common configurations have been reproduced from Ower and Pankhurst (1977) in table 3.

The overall mean velocity can now be calculated as the average of the measured local mean velocities. Again the velocities are determined in terms of differential pressure measurements. So for say N elements of area:

No. of Areas (N)	**No. of Measuring Points (2N)**	**Location of Measuring Points Along Diameter (y/D)**
2	4	0.043, 0.290, 0.710, 0.957
3	6	0.032, 0.135, 0.321, 0.679, 0.865, 0.968
4	8	0.021, 0.117, 0.184, 0.345, 0.655, 0.816, 0.883, 0.979
5	10	0.019, 0.076, 0.153, 0.217, 0.361, 0.639, 0.783, 0.847, 0.924, 0.981

Table 3: Positions of Mean Local Velocities. (Ower and Pankhurst 1977).

$$\bar{v} = \frac{\sum_{i=1}^{10}\left\{\left(\frac{2(\Delta P_i)}{\rho_e}\right)^{\frac{1}{2}}\right\}}{2N}$$

where

ΔP_i = The differential pressure at measuring point i (Pa)

Substituting this back into the velocity shape factor expression we have:

$$k_c = \frac{\bar{v}}{v_c} = \frac{\sum_{i=1}^{2N}\left\{\left(\frac{2(\Delta P_i)}{\rho_e}\right)^{\frac{1}{2}}\right\}}{2N\left(\frac{2(\Delta P_c)}{\rho_e}\right)^{\frac{1}{2}}}$$

This equation may be simplified

$$k_c = \frac{\sum_{i=1}^{2N}(\Delta P_i)^{\frac{1}{2}}}{2N(\Delta P_c)^{\frac{1}{2}}}$$

A Robust Bidirectional Probe

While not described in great detail in this document, it is worth mentioning the bi-directional probe, developed by McCaffrey and Heskestad (1976).

Apart from its suitable for use in elevated temperatures and 'sooty' conditions, its main advantages are that it is robust, relatively simple and has angular insensitivity to within about +/-50°. It has an accuracy of 5% under these conditions.

An apparatus dependant correction has to be made to the mass flow rate equation. This correction is a function of the Reynolds number with respect to the flow around the probe. In this case the characteristic dimension is the inside diameter of the probe. The mass flow rate relationship becomes

$$\dot{m}_e = \frac{Ak_c}{f(\Re_e)}\sqrt{2\rho_e \Delta P_c}$$

where

$\Re_e$ = The Reynolds number for that flow regime (--)

$f(\Re_e)$ = Some function of the Reynolds number where the characteristic dimension is the inside diameter of the probe.

It has been found that for 40 < $\Re_e$< 3800 then:

$$f\,(\Re_e) = \begin{pmatrix} 1.533 - 1.366\times 10^{-3}\Re_e + 1.688\times 10^{-6}\Re_e^2 - 9.706\times 10^{-11}\Re_e^3 \\ +2.555\times 10^{-13}\Re_e^4 - 2.484\times 10^{-17}\Re_e^5 \end{pmatrix}$$

Otherwise for $\Re_e$> 3800 then

$f\,(\Re_e)$ = 1.08 (ie: becomes asymptotic)

The correction has been derived from comparisons in ambient air with values from a pitot-static tube of 1.00.

It has been found by Janssens (1991b) in room fire experiments that with bidirectional probes of 16 mm diameter and greater, even at very low velocities, $\Re_e$ will exceed 3800, further simplifying the calculations.

Calculation of the Mass Flow Rate of the Exhaust Gases

Assuming the velocity shape factor has been determined from say 10 gauging points, and that the asymptotic value of 1.08 holds for the pressure probes:

The mass flow rate expression becomes:

$$\dot{m}_e = \frac{Ak_c}{1.08}\sqrt{2\rho_e \Delta P_c}$$

Assuming approximate ideal gas behaviour, (approximate only because mass isn't conserved in the process):

$$\rho_e \approx \frac{\rho_a T_a}{T_e}$$

where

ρ_a = Density of ambient air (kg/m^3) (= 1.177 kg/m^3 at 300 K)

T_a = Ambient absolute temperature(K)

T_e = Exhaust absolute temperature (K)

So substituting typical we have:

$$\dot{m}_e = \frac{Ak_c}{1.08}\sqrt{2\left(\frac{\rho_a T_a}{T_e}\right)\Delta P_c}$$

$$\dot{m}_e = \frac{Ak_c}{1.08}\sqrt{2\left(\frac{1.177 \times 300}{T_e}\right)\Delta P_c}$$

Summary of Calculation of Mass Flow Rate

$$\boxed{\dot{m}_e = 24.61 A k_c \sqrt{\frac{\Delta P_c}{T_e}}}$$

Where A is known, k_c calculated from the earlier expression and; ΔP_c and T_e are measured.

Chapter 6

Measurement of Fire Induced Vertical Vent Flows

Introduction

Fire induced convective flow into and from a room, has a significant influence on both pre-flashover and post-flashover fires. In the preflashover fire, the temperatures and heat transfer are affected. This in turn influences the fire spread. In the postflashover fire the rate of air flow into the compartment will affect the burning rate.

There are two main driving forces of gas flow present in a compartment during fire conditions; thermal expansion and buoyancy. Thermal expansion of the air in the compartment causes an out-flow through any available vents. Buoyancy results from a density gradient. This density gradient is analogous to a hydrostatic gradient in that it causes a pressure difference between the compartment air and outside ambient air. In turn this pressure difference causes horizontal gas flows through vertical vents.

In the early stages of fire growth, usually within the first 30 seconds, the buoyant forces become the dominant driving force. For a vent in the vertical plane, the compartment conditions become such that a positive pressure difference near the top of the vent due to a dense upper layer drives an out-flow of gas. Meanwhile the negative pressure difference at the bottom, due to the less than ambient density drives an in-flow of gas. Figure 1 illustrates this concept in terms of a simple 'zone model' for a compartment with a single vertical vent in one wall.

More complex deterministic models have been developed, such as the 'field models'. Field models are finite element or finite difference numerical methods which treat the room as a large number of smaller volumes. The equations of conservation of mass, species, momentum and energy are solved for each volume or 'element'. Field models are expensive in terms of the powerful computation hardware and software required. It is considered (Janssens 1992) that for essentially all fire engineering applications this level of detail is not required and that 'zone models' are more than adequate.

Early Investigations

The first simple theories of buoyancy driven flows in fully developed compartment fires were developed by Kawagoe (1958) and Thomas et. al. (1963). These investigators considered the problem as flow through an orifice using discharge coefficients to correct for the experimental uncertainties.

Rockett (1976) expanded this work and developed a generalised analytical model of fire induced convection. Rockett's model determines net room flow as a function of the ventilation factor, the ratio of gas temperatures above and below the thermal discontinuity and the height of the thermal discontinuity.

Steckler et. al. (1982) validated these orifice flow analogous techniques with a series of fifty-five full scale steady state room fire experiments. Mass flow rates were calculated in two ways. Firstly by integrating the velocity profiles across the door and secondly by applying the orifice flow techniques. Agreement was excellent.

Steckler's work resulted in the development of a practical experimental procedure to obtain vent flow rates. This method is discussed in depth later in this chapter. It is significant in that it precludes the need for velocity measurements which are expensive and cumbersome. Instead only one differential pressure measurement and the temperature profile need be measured.

In a second paper Steckler et al (1984) analysed his earlier data and presented mean flow coefficients for both inflow and outflow. It is significant that these values differ for inflow and outflow. They are a mean value of 0.68 for inflow and 0.73 for outflow.

Jones and Peacock (1989) showed that whilst there could be up to three neutral planes across a vent, in the general case of the two zone model there would only be one.

Janssens (1991) developed a technique to calculate the fire induced flow using only temperature profiles. This replaces the need to make one measurement of differential pressure. Instead the neutral plane is located by a conservation of mass technique. This method is also discussed later in this chapter.

Rockett's Two Zone Model

Rockett has developed expressions for the mass flow rates through a vent, satisfying conservation of mass and momentum. These expressions are constrained at any given instant in time by the height of the thermal discontinuity. Some of Rockett's important theoretical aspects are re-derived here.

Considering figure 1, the expressions for mass flow rates can be found by tracing the pressure changes around the loop 0→1, 1→2, 2→3, 3→0.

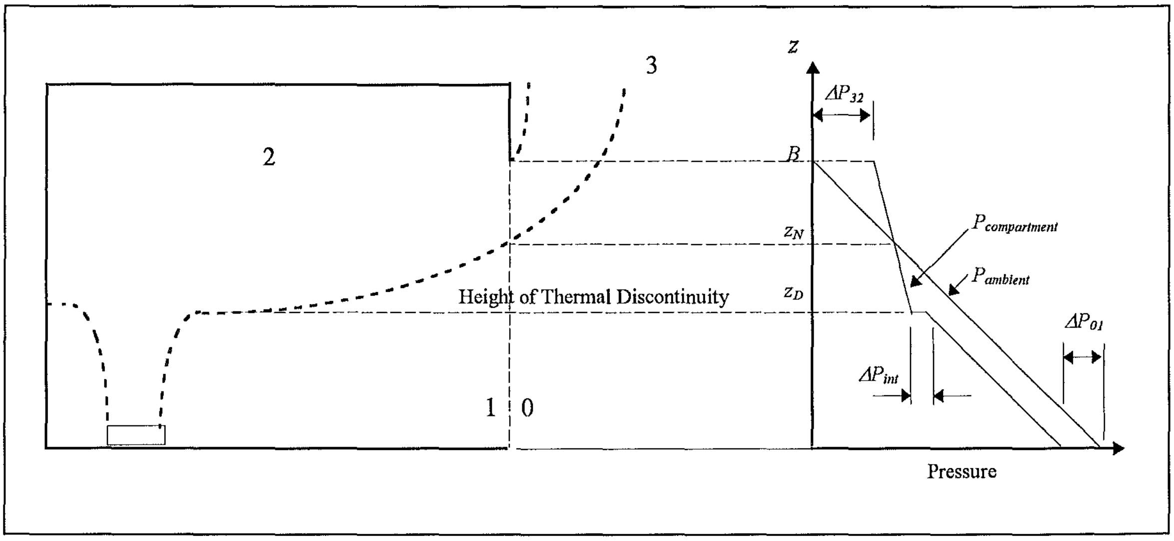

Figure 1: Rockett's pressure-height relationship for fire conditions superimposed against ambient conditions.

Nomenclature associated with figure 1

ΔP_{01} = Pressure differential at vent between zones 0→1 (Pa)

ΔP_{32} = Pressure differential at vent between zones 2→3 (Pa)

ΔP_{int} = Pressure loss within the interior of the compartment (Pa)

$P_{ambient}$ = Ambient pressure (Pa)

$P_{compartment}$ = Compartment pressure (Pa)

B = Height of vent (m)

z_N = Height of neutral plane (m)

z_D = Height of thermal discontinuity (m)

Mass Flow Rate Into a Compartment

The mass flow rate from 0→1 is driven by the pressure differential across the vent. Therefore consider this pressure change. From Bernoulli we have:

$$\frac{v_0^2}{2}+\frac{P_0}{\rho_0}=\frac{v_1^2}{2}+\frac{P_1}{\rho_1}$$

where

v_0 = Gas velocity in zone 0 (m/s)

P_0 = Gas pressure in zone 0 (Pa)

ρ_0 = Gas density in zone 0 (kg/m^3)

v_1 = Gas velocity in zone 1 (m/s)

P_1 = Gas pressure in zone 1 (Pa)

ρ_1 = Gas density in zone 1 (kg/m^3)

assuming

$v_1 \approx 0$

$\rho_0 \approx \rho_1$

Rearranging and substituting for ρ_0

$$\frac{v_0^2}{2}=\frac{P_1}{\rho_0}-\frac{P_0}{\rho_0}$$

$$\Delta P_{01}=\frac{\rho_0 v_0^2}{2}$$

where

ΔP_{01} = The pressure difference between 0 and 1 (Pa)

Where the differential pressure is a function of height and so becomes:

$$\Delta P_{01}(z) = \frac{\rho_0 v_0(z)^2}{2}$$

In other words the kinetic energy of air accelerated from rest along a streamline is equal to the change in static pressure.

Consider now the vent in the fire compartment as an orifice of width 'a'.

We can express the mass flow rate in as:

$$\dot{m}_{in} = c_{in}\rho_0 Q_{in}$$

where

$\dot{m}_{in}$ = Mass flow rate of air in through vent (kg/s)

Q_{in} = The volumetric flow rate 'in' through vent (m^3/s)

c_{in} = Orifice discharge coefficient 'in' (--)

$$\dot{m}_{in} = ac_{in}\int_z \rho_0 v(z)dz$$

Rearrange the differential pressure expression and substitute into the mass flow rate expression.

$$v(z) = \sqrt{\frac{2\Delta P_{01}(z)}{\rho_0}}$$

$$\dot{m}_{in} = ac_{in}\rho_0 \int_z \left(\frac{2\Delta P_{01}(z)}{\rho_0}\right)^{\frac{1}{2}} dz$$

$$\dot{m}_{in} = ac_{in}\int_z \left(2\rho_0 \Delta P_{01}(z)\right)^{\frac{1}{2}} dz$$

Now consider the variable $\Delta P_{01}(z)$ and introduce limits for the region below the neutral plane. Assume $\Delta P_{\text{int}} = 0$. This assumption may not normally be reasonable, but at this stage it will more simply demonstrates the general form of the flow equations. We will re-derive the expressions more accurately when reviewing Steckler's work.

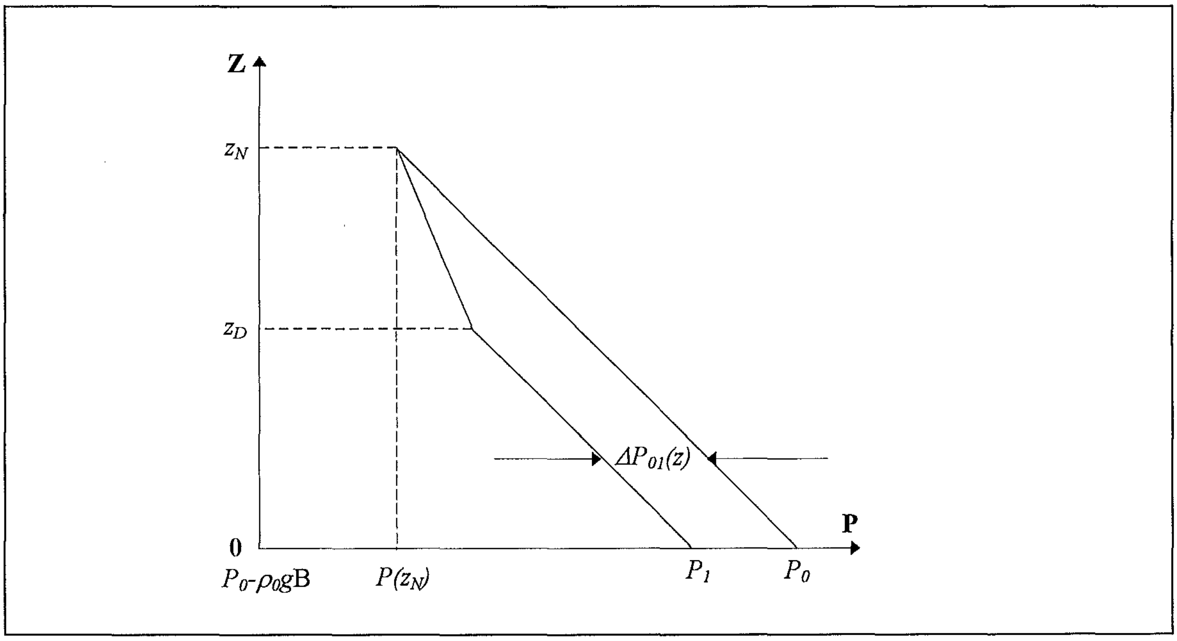

Figure 2: Pressure-height relationship below the neutral plane.

We can express ΔP_0 and ΔP_1 as a function of z, below and above the height of thermal discontinuity, z_D:

Below the Thermal Discontinuity ($0 \leq z \leq z_D$)

Consider the ambient pressure at height z_D.

$$P_0(z_D) = P(z_N) + \rho_0 g(z_N - z_D)$$

Now consider the compartment pressure distribution at some point immediately above z_D. Above, not below as might be expected, as the former pressure determines the magnitude of the differential pressure at all points below the thermal discontinuity.

$$P_1(z_D) = P(z_N) + \rho_2 g(z_N - z_D)$$

where

ρ_2 = Gas Density in zone 2 (kg/m^3)

Note that the difference in pressure is constant for this range of z, therefore:

$$\Delta P_{01}(z) = \Delta P_{01}(z_D) = \{P(z_N) + \rho_0 g(z_N - z_D)\} - \{P(z_N) + \rho_2 g(z_N - z_D)\}$$

$$\Delta P_{01}(z) = (\rho_0 - \rho_2) g(z_N - z_D)$$

Assuming ideal gas behaviour in regions 0 and 1.

$$\rho_0 T_0 = \rho_2 T_2$$

$$\rho_2 = \rho_0 \frac{T_0}{T_2}$$

Substituting

$$\Delta P_{01}(z) = \left(\rho_0 - \rho_0 \frac{T_0}{T_2}\right) g(z_N - z_D)$$

$$\Delta P_{01}(z) = \rho_0 g\left(1 - \frac{T_0}{T_2}\right)(z_N - z_D)$$

<u>Above the Thermal Discontinuity ($z_D \leq z \leq z_N$)</u>

We have that:

$$P_0(z) = P(z_N) + \rho_0 g(z_N - z)$$

and

$$P_1(z) = P(z_N) + \rho_2 g(z_N - z).$$

Therefore

$$\Delta P_{01}(z) = (\rho_0 - \rho_2)g(z_N - z)$$

Again assuming ideal gas behaviour in regions 0 and 1.

$$\Delta P_{01}(z) = \rho_0 g\left(1 - \frac{T_0}{T_2}\right)(z_N - z)$$

Substituting

$$\dot{m}_{in} = ac_{in}\rho_0 \int_0^{z_D} \left\{2g\left(1 - \frac{T_0}{T_2}\right)(z_N - z_D)\right\}^{\frac{1}{2}} dz + ac_{in}\rho_0 \int_{z_D}^{z_N} \left\{2g\left(1 - \frac{T_0}{T_2}\right)(z_N - z)\right\}^{\frac{1}{2}} dz$$

$$\dot{m}_{in} = ac_{in}\rho_0 \left[2g\left(1 - \frac{T_0}{T_2}\right)\right]^{\frac{1}{2}} \left[\int_0^{z_D} (z_N - z_D)^{\frac{1}{2}} dz + \int_{z_D}^{z_N} (z_N - z)^{\frac{1}{2}} dz\right]$$

The importance of the solution to the integral is that it expresses the mass flow rate dependance with respect to the ventilation factor (aB)√(B). The solution is not included here, as it does not directly relate to an experimental method for measurement of mass flow rates, but can be found in Rockett (1976).

Mass Flow Rate Out of a Compartment

This process is simular to the above.

Assuming

$v_2 \approx v_0 \approx 0$

$\rho_3 \approx \rho_0$

where

ρ_3 = Gas Density in zone 3 (kg/m^3)

v_2 = Gas Velocity in zone 1 (m/s)

The earlier expressions for differential pressure and mass flow rate become

$$\Delta P_{32}(z) = \frac{\rho_2 v_2(z)^2}{2}$$

$$\dot{m}_{out} = ac_{out}\rho_0 \int_z v(z)dz$$

where

$\dot{m}_{out}$ = Mass flow rate of air 'out' through vent (kg/s)

c_{out} = Orifice discharge coefficient 'out' (--)

$$\dot{m}_{out} = ac_{out} \int_z (2\rho_2 \Delta P_{32}(z))^{\frac{1}{2}} dz$$

Now the differential pressure $\Delta P_{32}(z)$ can be described by the function

$$\Delta P_{32}(z) = (\rho_3 - \rho_2)g(z - z_N)$$

Simplifying and substituting for the ideal gas law

$$\Delta P_{32}(z) = (\rho_0 - \rho_2)g(z - z_N)$$

$$\Delta P_{32}(z) = \rho_0 g\left(1 - \frac{T_0}{T_2}\right)(z - z_N)$$

Substituting, introducing limits and simplifying

$$\dot{m}_{out} = ac_{out} \int_z \left(2\rho_2\rho_0 g\left(1 - \frac{T_0}{T_2}\right)(z - z_N)\right)^{\frac{1}{2}} dz$$

$$\dot{m}_{out} = ac_{out}\int_{z_N}^{B}\left(2\rho_0\frac{T_0}{T_2}\rho_0 g\left(1-\frac{T_0}{T_2}\right)(z-z_N)\right)^{\frac{1}{2}}dz$$

$$\dot{m}_{out} = ac_{out}\rho_0\left\{2\frac{T_0}{T_2}g\left(1-\frac{T_0}{T_2}\right)\right\}^{\frac{1}{2}}\int_{z_N}^{B}(z-z_N)^{\frac{1}{2}}dz$$

As before the solution to this integral is not included here but can be found in Rockett (1976).

Steckler's General Equations for Calculating Mass Flow Rates from Temperature Profiles and One Differential Pressure Measurement

Rockett's two-zone model assumes only two temperatures in the compartment. One above and one below the interface of thermal discontinuity. Steckler et al (1982) improved this model by measuring the actual temperature profiles using thermocouple 'trees'. So we will leave Rocket's model, in the generalised form above and pursue Steckler's work.

The ambient and quiescent compartment pressure profiles can be calculated from their respective temperature profiles, assuming ideal gas behaviour. The neutral plane is then located via a referenced differential pressure measurement, usually taken and the vent sill height. Hence the ambient and compartment differential pressure profiles can be calculated. The velocity profiles and therefore mass flow rates can be determined assuming flow through an orifice.

This methodology is discussed in more detail, at such time as it occurs in the following derivations.

Mass Flow Rate Into a Compartment

Consider first the expressions for ambient pressure with respect to height, below the neutral plane

$$P_0(z) = P(z_N) + \rho_0 g(z_N - z)$$

Using the ideal gas assumption to express ρ_0 in terms of T_0

$$\rho_0 T_0(z) = \rho_{ref} T_{ref}$$

where

ρ_{ref} = Density of ambient air at temperature T_{ref} and atmospheric pressure (kg/m^3)

T_{ref} = Some reference temperature (K)

therefore

$$\rho_0 = \frac{\rho_{ref} T_{ref}}{T_0}$$

assuming values for

$g = 9.81$ m/s^2

$T_{ref} = 300$ K

$\rho_{ref} \approx 1.17$ kg/m^3

the pressure expression can now be simplified

$$P_0(z) = P(z_N) + \frac{\rho_{ref} T_{ref}}{T_0} g(z_N - z)$$

$$P_0(z) = P(z_N) + \frac{3461}{T_0}(z_N - z)$$

consider a series of measurements, each taken at some varying height z′

$$P_0(z) = P(z_N) + 3461 \int_{z}^{z_N} \frac{1}{T_0} dz'$$

where

z′ = Measuring point (m).

Secondly consider the pressure profile in the compartment

$$P_1(z) = P(z_N) + \rho_1(z')g(z_N - z)$$

where

$\rho_1(z')$ = Density of the compartment gases at measuring point z′ (kg/m^3).

Using the ideal gas assumptions

$$P_1(z) = P(z_N) + \frac{\rho_{ref}T_{ref}}{T_1(z')}g(z_N - z)$$

where

$T_1(z')$ = Temperature of the compartment gases at measuring point z′ (kg/m^3)

$$P_1(z) = P(z_N) + 3461\int_z^{z_N}\frac{1}{T_1(z')}dz'$$

therefore

$$\Delta P_{01}(z) = P_1(z) - P_0(z)$$

$$\Delta P_{01}(z) = \left(P(z_N) + 3461\int_z^{z_N}\frac{1}{T_1(z')}dz'\right) - \left(P(z_N) + 3461\int_z^{z_N}\frac{1}{T_0}dz'\right)$$

$$\Delta P_{01}(z) = 3461\int_z^{z_N}\left(\frac{1}{T_1(z')} - \frac{1}{T_0}\right)dz'$$

Substituting this back into a mass flow rate expression which accounts for the temperature distribution in the plane of the vent

$$\dot{m}_{in} = c_{in}\rho_d(z')Q_{in}$$

where

$\rho_d(z')$ = density distribution at the orifice (ie vent) as a function of vent height. (kg/m^3)

$$\dot{m}_{in} = ac_{in}\int_0^{z_N}\rho_d(z')v(z')dz'$$

$$v(z) = \sqrt{\frac{2\Delta P_{01}(z')}{\rho_d(z')}}$$

$$\dot{m}_{in} = ac_{in}\int_0^{z_N}\rho_d(z')\left(\frac{2\Delta P_{01}(z')}{\rho_d(z')}\right)^{\frac{1}{2}}dz'$$

$$\dot{m}_{in} = ac_{in}\int_0^{z_N}\left(2\rho_d(z')\Delta P_{01}(z')\right)^{\frac{1}{2}}dz'.$$

Therefore the mass flow rate expression becomes

$$\dot{m}_{in} = ac_{in}\int_0^{z_N}\left(2\rho_d(z')3461\int_{z'}^{z_N}\left(\frac{1}{T_1(z'')}-\frac{1}{T_0}\right)dz''\right)^{\frac{1}{2}}dz$$

where

z' = Measuring point in plane of vent (m)

z'' = Measuring point in quiescent corner of compartment (m)

Using the ideal gas assumption and reference temperature and pressure approach

$$\rho_d(z') = \frac{\rho_{ref} T_{ref}}{T_d(z')}$$

where

$T_d(z')$ = Temperature profile at the vent (K)

T_{ref} = 300 K

$\rho_{ref} \approx 1.17$ kg/m^3

$$\rho_d(z') = \frac{352}{T_d(z')}$$

therefore

$$\dot{m}_{in} = ac_{in} \int_0^{z_N} \left(2 \frac{352}{T_d(z')} 3461 \int_{z'}^{z_N} \left(\frac{1}{T_1(z'')} - \frac{1}{T_0} \right) dz'' \right)^{\frac{1}{2}} dz'$$

relative to the compartment this flow rate is negative, in other words drawing air in, therefore a negative sign is introduced

$$\dot{m}_{in} = -1561 ac_{in} \int_0^{z_N} \left(\frac{1}{T_d(z')} \int_{z'}^{z_N} \left(\frac{1}{T_1(z'')} - \frac{1}{T_0} \right) dz'' \right)^{\frac{1}{2}} dz'$$

the final expression is

$$\dot{m}_{in} = 1561 ac_{in} \int_0^{z_N} \left(\frac{1}{T_d(z')} \int_{z'}^{z_N} \left(\frac{1}{T_0} - \frac{1}{T_1(z'')} \right) dz'' \right)^{\frac{1}{2}} dz'$$

Mass Flow Rate Out of a Compartment

The equation for mass flow rate out of the vent is derived in a simular manner. However the limits of both integrals and the discharge coefficient vary.

$$\dot{m}_{out} = 1561 a c_{out} \int_{z_N}^{B} \left(\frac{1}{T_d(z')} \int_{z_N}^{z'} \left(\frac{1}{T_0} - \frac{1}{T_1(z'')} \right) dz'' \right)^{\frac{1}{2}} dz'$$

Note in this case the outflowing gases mainly consist of nitrogen, therefore the densities are simular to air making it acceptable to assume ideal gas behaviour.

Location of the Neutral Plane

As previously noted, the temperature profiles of ambient air and quiescent compartment conditions need to be related. This is done via location of the neutral plane. The neutral plane can be located by solving the differential pressure expression for a reference differential pressure measurement $\Delta P(z_{ref})$ at a reference height z_{ref}. The reference height is usually taken as the soffit of the vent as this is where the differential pressure will be a maximum.

$$\Delta P(z_{ref}) = 3461 \int_{z_{ref}}^{z_N} \left(\frac{1}{T_1(z'')} - \frac{1}{T_0} \right) dz''$$

Note that when the value in the parenthesis becomes negative this indicates the location is above the neutral plane and hence the negative value causes the limits to be reversed.

Once the neutral plane is located the integrals can be solved for mass flow rates. Algorithms have been developed by NBS (now known as NIST), to reduce the data accumulated during room fire experiments. Theses algorithms are referred to as RAPID (Breese and Peacock 1986).

Included in the RAPID algorithms described above are those appropriate to mass flow calculations using Steckler's general equations.

Janssens' Method of Calculating Mass Flow Rates from Temperature Profiles Only

Janssens (1991b) proposes that only temperature profiles need be measured in calculating the vent flows. This is based on conservation of mass within the compartment. The main advantages of this method are that the inaccuracy and expense associated with the use of a differential pressure measuring device are avoided.

Consider the mass balance

$$\frac{dm_r}{dt} = \dot{m}_{in} + \dot{m}_b + \dot{m}_v - \dot{m}_{out} \cdot$$

where

m_r = Mass accumulated inside the compartment (kg)

$\dot{m}_b$ = Mass flow rate of ignition source (kg/s)

$\dot{m}_v$ = mass flow rate of water vapour and pyrolozates emerging from the compartment surfaces (kg/s)

$\dot{m}_b$ and $\dot{m}_v$ can be measured or ignored as insignificant. If the ignition source is anything other than a simple gas burner then $\dot{m}_b$ becomes impractical to measure.

Consider the mass of gases in the room

$$m_r = density \times volume$$

Assuming a rectangular compartment, the volume and density can be expressed as functions of the room height

$$m_r = WL\int_0^H \rho.(z')dz'$$

where

W = compartment width (m)

L = compartment length (m)

H = compartment height (m)

The density profile can be determined from a temperature profile using the earlier ideal gas assumptions and reference temperature and pressures

$$\rho.(z') = \frac{\rho_{ref} T_{ref}}{T_1(z')}$$

$$\rho.(z') = \frac{352}{T_1(z')}.$$

Substituting this back into the integral

$$m_r = 352WL\int_0^H \frac{1}{T_1(z')}\,dz'.$$

Expressed as a rate

$$\frac{dm_r}{dt} = 352WL\frac{d}{dt}\int_0^H \frac{1}{T_1(z')}\,dz'.$$

This expression can be related to the original mass balance

$$352WL\frac{d}{dt}\int_0^H \frac{1}{T_1(z')}\,dz' = \dot{m}_{in} + \dot{m}_b + \dot{m}_v - \dot{m}_{out}.$$

Ignoring $\dot{m}_b$ and $\dot{m}_v$ this expression becomes

$$\dot{m}_{in} - \dot{m}_{out} = 352WL\frac{d}{dt}\int_0^H \frac{1}{T_1(z')}\,dz'.$$

This supersedes the requirement to locate the neutral plane via a referenced differential pressure measurement. Temperature profiles and a mass balance will allow the following expression to be solved for an unknown z_N.

$$352WL\frac{d}{dt}\int_0^H \frac{1}{T_1(z')}dz' = 1561a\left\{\begin{array}{l} c_{in}\int_0^{z_N}\left(\frac{1}{T_d(z')}\int_{z'}^{z_N}\left(\frac{1}{T_0}-\frac{1}{T_1(z'')}\right)dz''\right)^{\frac{1}{2}}dz' \\ -c_{out}\int_{z_N}^{B}\left(\frac{1}{T_d(z')}\int_{z_N}^{z'}\left(\frac{1}{T_0}-\frac{1}{T_1(z'')}\right)dz''\right)^{\frac{1}{2}}dz'\end{array}\right\}$$

Experimental Layout

Figure 3 shows a schematic of the apparatus layout to be used in measuring the fire induced convective vent flows. The differential pressure measurement can be included for RAPID calculations as an alternative to the mass balance approach. Results can then be gathered by both means for comparison.

Thermocouple tree #1 is placed perpendicular to the vent and far enough removed to measure the temperature profile of ambient air.

Thermocouple tree #1 measurements are entered in the expression ⇒ $\int_z \frac{1}{T_0}dz$

Thermocouple tree #2 is placed in the doorway to determine the temperature profile at the vent.

Thermocouple tree #2 measurements are entered in the expression ⇒ $\int_z \frac{1}{T_d(z')}dz$

Thermocouple tree #3 is placed in a quiescent corner of the compartment to measure the temperature profile in the compartment.

Thermocouple tree #3 measurements are entered in the expression ⇒ $\int_z \frac{1}{T_1(z')}dz$

The differential pressure measuring device is placed at soffit height to be used as a reference in the RAPID calculations.

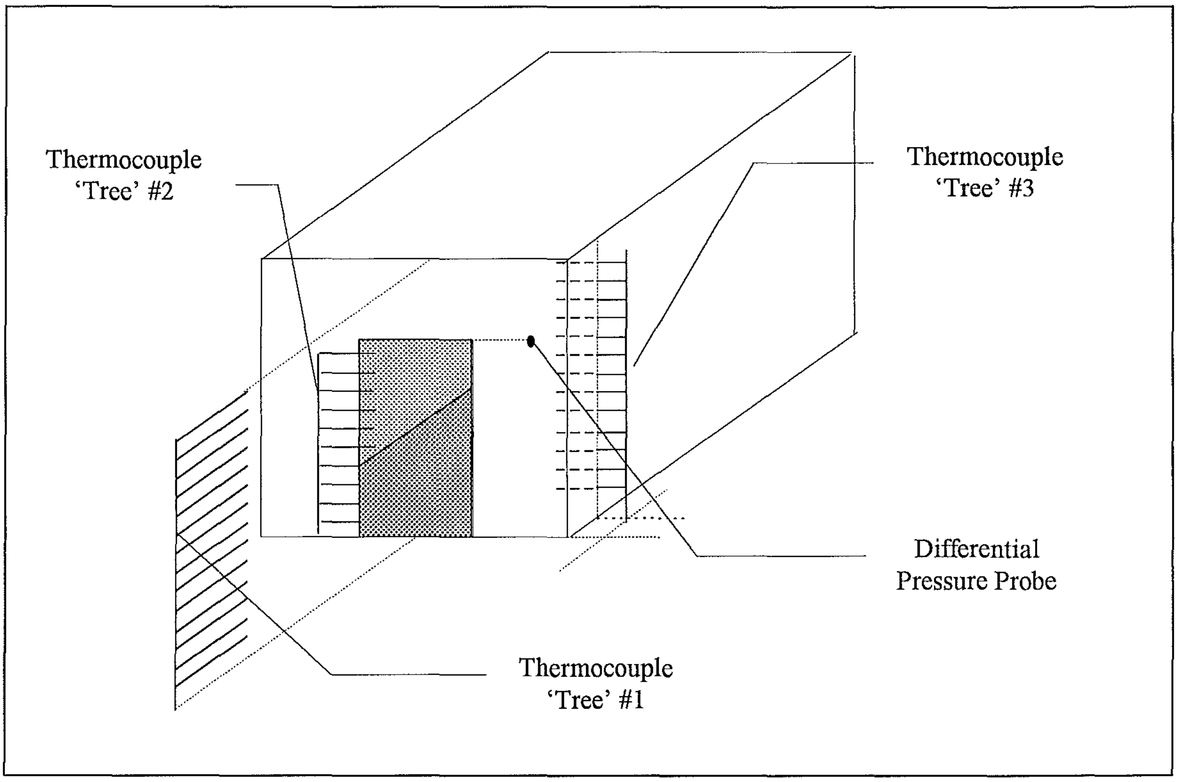

Figure 3: Schematic of layout of vent flow measuring apparatus.

Note that a description of the thermocouples and their performance is not covered in this document. A more detail reference is recommended such as ASTM guide (1994).

Summary of Calculation of Vent Mass Flow Rates

The calculation of the mass flow rates can be accomplished with the measurements of the three thermocouple trees in a two step process.

Step 1: Calculate the location of the neutral plane via inputting the measurements from the three thermocouple trees into Janssens' mass conservation relationship, and solving for z_N.

$$352WL\frac{d}{dt}\int_0^H \frac{1}{T_1(z')}\,dz' = 1561a\left\{\begin{array}{l} c_{in}\displaystyle\int_0^{z_N}\left(\frac{1}{T_d(z')}\int_{z'}^{z_N}\left(\frac{1}{T_0}-\frac{1}{T_1(z'')}\right)dz''\right)^{\frac{1}{2}}dz' \\ -c_{out}\displaystyle\int_{z_N}^{B}\left(\frac{1}{T_d(z')}\int_{z_N}^{z'}\left(\frac{1}{T_0}-\frac{1}{T_1(z'')}\right)dz''\right)^{\frac{1}{2}}dz' \end{array}\right\}.$$

Step 2: Once the neutral plane is located, calculate the mass flow rates in and out of the compartment plane via inputting the measurements from the three thermocouple trees into Steckler's equations.

Mass flow rate in:

$$\boxed{\dot{m}_{in} = 1561ac_{in}\int_{0}^{z_N}\left(\frac{1}{T_d(z')}\int_{z'}^{z_N}\left(\frac{1}{T_0}-\frac{1}{T_1(z'')}\right)dz''\right)^{\frac{1}{2}}dz'}$$

Mass flow rate out:

$$\boxed{\dot{m}_{out} = 1561ac_{out}\int_{z_N}^{B}\left(\frac{1}{T_d(z')}\int_{z_N}^{z'}\left(\frac{1}{T_0}-\frac{1}{T_1(z'')}\right)dz''\right)^{\frac{1}{2}}dz'}$$

Chapter 7

Summary and Conclusions

Summary

The purpose of this research was to provide the basis upon which the University of Canterbury could design a full-scale fire laboratory.

This document consists of a review of full-scale room fire experimental methods. The role of other scales of fire test methods are discussed. Fundamental principles of room fire behaviour are examined. In particular, detailed analysis' of the properties of heat release and fire induced vent flows are presented.

Conclusions and Recommendations

Fire Properties

To successfully investigate fire properties in a full-scale environment requires an integrated knowledge of many areas of science and engineering. Of particular importance are; mathematics, numerical methods, physics, chemistry, fluid mechanics and fire dynamics. It is notable that although some of the principles of room fire behaviour are well documented, actual physical measurement of these properties is difficult. The case in point from this study is obviously the measurement of rate of heat release using the oxygen consumption principle.

Experiments are an essential part of fire safety science. In particular full-scale experiments are crucial in validating fire models.

The most important of the room fire properties is the heat release rate.

It is not meaningful to measure the heat release rate indirectly as was done prior to 1980. The principal of oxygen consumption allows direct measurement of the heat release rate.

The methodology presented in this document for the measurement of; heat release rates, mass flow rates of exhaust products and fire induced vertical vent flows, should be adopted.

The calculations presented in this document for determining the; heat release rates, mass flow rates of exhaust products and fire induced vertical vent flows, should be adopted. In addition to the fire properties examined in this document, other characteristics need also be examined. (See future research recommendations).

The Facility

It is desirable to standardise full-scale room fire experiments into appropriate test methods.

The fire experiment facility at the University of Canterbury should be at least equivalent to the ISO 9705 room. This will also allow the University to become part of the world wide round robin testing program for building materials.

The fire laboratory at the University of Canterbury, should be constructed as closely to the ASTM E-603 guide as finances allow.

The compartment should be constructed so as to be able to be removed. This will allow the exhaust system to be used on its own, as a furniture calorimeter.

The hood and extract system should be designed with reference to Cooper (1994)

Future Research

Extensive research is required in developing reliable models of full-scale room behaviour from full-scale open air experimental results and ultimately bench-scale results.

It should be recognised that this document is only the beginning. Several fire properties such as; heat release of a fuel of a known composition via a carbon balance technique, compartment temperatures, layer interface heights, heat flux, ignition source, ignition location, fire spread, smoke production and toxicity, have not been considered here.

It is recommended that these properties be prioritised and examined in future projects.

References

Ames, S. A. Babrauskas, V. and Parker, W. J. 1992 "Upholstered Furniture: Prediction by Correlations" *Heat Release in Fires*. (Babrauskas, V. and Grayson, S. J. Ed.). Elsevier Science Publishers ltd. London. p519-543.

ASTM. 1977. "Standard Guide for Room Fire Experiments" *Annual Book of ASTM Standards*. ASTM E 603-77 (1994). American Society for Testing Materials. Philadelphia.

ASTM. 1982. "Proposed Standard Method for Room Fire Test of Wall and Ceiling Materials and Assemblies" *Annual Book of ASTM Standards.* American Society for Testing Materials. Philadelphia.

ASTM. 1992. "Standard Test Method for Heat and Visible Smoke Release Rates for Materials and Products Using an Oxygen Consumption Calorimeter" *Annual Book of ASTM Standards*. ASTM E 1354-90 (1992). American Society for Testing Materials. Philadelphia.

Babrauskas, V. 1982. "Development of the Cone Calorimeter - A Bench-scale Heat Release Rate Apparatus Based on Oxygen Consumption" NBSIR 82-2611. National Bureau of Standards. Gaithersburg, MD.

Babrauskas, V. 1983. "Fire Engineering Test Development: Bench-scale Tests to Predict Full-scale Behaviour" 7th UJNR Panel on Fire Research and Safety. p369-391.

Babrauskas, V. 1984. "Development of the Cone Calorimeter - A Bench-scale Heat Release Rate Apparatus Based on Oxygen Consumption" *Fire and Materials* v8 No.2 p81-95.

Babrauskas, V. 1985. "Fire Engineering Test Development: Bench-scale Tests to Predict Full-scale Behaviour" 7th Joint Panel Meeting on Fire Research and Safety, October 24-28 1983 p369-392 (also NBSIR 85-3118) National Bureau of Standards Gaithersburg, MD.

Babrauskas, V. 1987. "Use of the Cone Calorimeter for Smoke Production Measurements" Technical Papers, Regional Technical Conference, 'PVC: The Issues' p41-64. Society of Plastic Engineers Incorporated. Atlantic City, NJ.

Babrauskas, V. 1991. "Effective Measurement Techniques for Heat, Smoke and Toxic Fire Gases" *Fire Safety Journal.* v17 p13-26.

Babrauskas, V. 1992a. "From Bunsen Burner to Heat Release Rate Calorimeter" *Heat Release in Fires.* (Babrauskas, V. and Grayson, S. J. Ed.). Elsevier Science Publishers ltd. London. p7-29.

Babrauskas, V. 1992b. "Full-scale Heat Release Rate Measurements" *Heat Release in Fires.* (Babrauskas, V. and Grayson, S. J. Ed.). Elsevier Science Publishers ltd. London. p93-111.

Babrauskas, V. and Wickstrom, U. 1989. "The Rational Development of Bench-scale Fire Tests for Full-scale Fire Prediction" *Fire Safety Science - Proceedings of the Second International Symposium.* p813-822.

Beitel, J. J. jr. 1990. "ISR's Fire Research Project" *ASTM Standardisation News.* v18, p36-38.

Breese, J. and Peacock, R. 1986. "A User's Guide for RAPID, Reduction Algorithms for the Presentation of Incremental Fire Data" Special Publication 722, National Bureau of Standards Gaithersburg, MD.

Cooper, C. Y. 1994. "Some Factors Affecting the Design of a Calorimeter Hood and Exhaust" *Journal of Fire Protection Engineering.* v6, No. 3, p99-111.

Dietenberger, M. A. 1988. "Improved Furniture Modelling Within 'FAST': HEMFAST-2" NBS-GCR-88-545 National Bureau of Standards Gaithersburg, MD.

Dietenberger, M. A. 1992. "Upholstered Furniture: Detailed Model" *Heat Release in Fires*. (Babrauskas, V. and Grayson, S. J. Ed.). Elsevier Science Publishers ltd. London. p479-518.

Emmons, H. 1968 *Fire Research Abstracts and Reviews*. v10 p133.

Fitzgerald, W. E. 1978 "Quantification of Fires: 1. Energy Kinetics of Burning in a Dynamic Room Size Calorimeter" *Journal of Fire and Flammability*. v9 p510-525.

Fleischmann, C. Bolliger, I. and Millar, D. 1995. "Exploratory Experiments on Backdraft in a Full Residential Scale Compartment" To be published. University of Canterbury, Christchurch, New Zealand.

Huggett, C. 1980. "Estimation of Rate of Heat Release by Means of Oxygen Consumption" *Fire and Materials*. v4 No. 2 p61-65.

ISO. 1986. "Fire Tests - Reaction to Fire - Ignitability of Building Materials" ISO 5657:1986(E). International Standards Organisation.

ISO. 1992. "Fire Tests - Reaction to Fire - Rate of Heat Release from Building Products" ISO 5660:1992. International Standards Organisation.

ISO. 1993. "Fire Tests - Full-scale Room Test for Surface Products" ISO 9705:1993(E). International Standards Organisation.

Janssens, M. 1991(a). "Measuring Rate of Heat Release by Oxygen Consumption" *Fire Technology*. v27 p234-49.

Janssens, M. 1991(b). "Fundamental Thermophysical Characteristics of Wood and their Role in Enclosure Fire Growth" Dissertation for the Degree of Doctor of Philosophy, University of Ghent (Belgium), National Forest Products Association, Washington, DC.

Janssens, M. 1992 "Room Fire Models" *Heat Release in Fires*. (Babrauskas, V. and Grayson, S. J. Ed.). Elsevier Science Publishers ltd. London. p113-157.

Jones, W. and Peacock, R. 1989. "Technical Reference Guide to FAST 18" Technical Note 1262, National Institute of Standards and Technology, Gaithersburg, MD.

Kawagoe, K. 1958. "Fire Behaviour in Rooms" Building Research Institute, Ministry of Construction, Tokyo.

McCaffrey, B. and Heskestad, G. 1976. "A Robust Bi-directional Low-velocity Probe for Flame and Fire Application" *Combustion and Flame*. v26 p125-127.

NFPA. 1990. "Standard Method of Test for Heat Release Rates for Upholstered Furniture Components or Composites and Mattresses Using an Oxygen Consumption Calorimeter" ANSI/NFPA 264A 1990. National Fire Protection Association. Quincy MA.

NORDTEST. 1987. "Upholstered Furniture: Burning Behaviour - Full-scale Test" NT FIRE 032, Helsinki, Finland.

Ower, E. and Pankhurst, R. 1977. "The Measurement of Air Flow" Pergamon Press 5th Edition.

Parker, W. 1982. "Calculations of the Heat Release Rate by Oxygen Consumption for Various Applications" NBSIR 81-2427. National Bureau of Standards Gaithersburg, MD.

Quintiere, J. G. 1989. "Scaling Applications in Fire Research" *Fire Safety Journal*. v15, p3-29.

Redfern, J. P. 1990. "The Use of Rate of Heat Release in Understanding Fire." *International Journal of Materials and Product Technology*. v5 p349-366.

Rockett, J. 1976. “Fire Induced Gas Flow in an Enclosure” *Combustion Science and Technology* v12 p165-175.

Ross, D. and Robertson, J. 1951. “A Superposition Analysis of the Turbulent Boundary Layer in an Adverse Pressure Gradient” *Journal of Applied Mechanics* v18 p95-100.

Schlichting, H. 1976. “Boundary Layer Theory” McGraw-Hill Series in Mechanical Engineering.” McGraw-Hill, 7th Edition.

Shaw, J. R. and Urbas, J. 1993. “An Intermediate-scale Calorimeter for Building Materials and Assemblies” *Fire and Materials* v17 p259-263.

Steckler, K. Buam, H. and Quintiere, J. 1984. “Fire Induced Flow through Room Openings - Flow Coefficients” *Twentieth International Symposium on Combustion,* The Combustion Institute, Pittsburgh PA p1591-1600.

Steckler, K. Quintiere, J. and Rinkinen, W. 1982. “Flow Induced by Fire in a Compartment” NBS-IR 82-2520. National Bureau of Standards Gaithersburg, MD.

Thomas, P. Hinkley, P. Theobald, C. and Simms, D. 1963. “Investigations into the Flow of Hot Gases in Roof Venting.” Fire Research Technical Paper No. 7, Ministry of Technology, London.

Thornton, W. 1917. “The Relationship of Oxygen to the Heat of Combustion of Organic Compounds.” *Philosophical Magazine and Journal of Science.* v33 p196-203.

Twilley, W. H. and Babrauskas, V. 1987. “User’s Guide for the Cone Calorimeter.” NBS-SP 745. National Bureau of Standards Gaithersburg, MD.

White, J. A. jr. 1986. “Full-scale Room Fire Testing:- Does It Give Us All the Answers ?” *ASTM Standardisation News.* American Society for Testing Materials.

Williamson, R. B. and Dembsey, N. A. 1993. “Advances in Assessment Methods for Fire Safety” *Fire Safety Journal* v20 p15-38.

Winternitz, F. and Fischl, C. 1957. “A Simplified Integration Technique for Pipe Flow Measurement” *Water Power* v9 p225-234.

Bibliography

ASTM. 1987. "Standard Practice for Conducting an Interlaboratory Study to Determine the Precision of a Test Method" *Annual Book of ASTM Standards.* ASTM E 691-87. American Society for Testing Materials. Philadelphia.

ASTM. 1991. "Standard Terminology of Fire Standards" *Annual Book of ASTM Standards.* ASTM E 176-91d. American Society for Testing Materials. Philadelphia.

Babrauskas, V. 1981. "Applications of Predictive Smoke Measurements" *Journal of Fire and Flammability.* v12, p51-64.

Babrauskas, V. 1984a. "Development of the Cone Calorimeter - A Bench-scale Heat Release Rate Apparatus Based on Oxygen Consumption" *Fire and Materials.* v8, No. 2, p81-95.

Babrauskas, V. 1984b. "Bench-scale Methods for Prediction of Full-scale Fire Behaviour of Furnishings and Wall Linings" SFPE Technology Report 84-10. Society of Fire Protection Engineers, Boston.

Babrauskas, V. 1988. "Burning Rates" Chapter 2-1 *SFPE Handbook of Fire Protection Engineering.* Society of Fire Protection Engineers. Quincy, MA.

Babrauskas, V. 1990a. "The Cone Calorimeter: A New Tool for Fire Safety Engineering" *ASTM Standardisation News.* January 1990.

Babrauskas, V. 1990b. "New test Methods for Assessing Smoke, Toxic Products, Corrosive Products and Heat Release in Fires" *Flame Retardants.* p20-31.

Babrauskas, V. 1991. "Effective Measurement Techniques for Heat, Smoke and Toxic Fire Gases" *Fire Safety Journal.* v17, p13-26.

Babrauskas, V. 1992a. "Cone Calorimeter Annotated Bibliography 1982 - 1991" NIST Technical Note 1296. National Institute of Standards and Technology. Gaithersburg MD.

Babrauskas, V. 1992b. "Room Fire Models" *Heat Release in Fires*. (Babrauskas, V. and Grayson, S. J. Ed.). Elsevier Science Publishers ltd. London. p113-157.

Babrauskas, V. 1992c. "Handling Fire Data: the FDMS" *Heat Release in Fires*. (Babrauskas, V. and Grayson, S. J. Ed.). Elsevier Science Publishers ltd. London. p591-607.

Babrauskas, V. 1992d. "Gas Species Measurement" *Heat Release in Fires*. (Babrauskas, V. and Grayson, S. J. Ed.). Elsevier Science Publishers ltd. London. p251-255.

Babrauskas, V. 1992e. "Heat of Combustion and Potential Heat" *Heat Release in Fires*. (Babrauskas, V. and Grayson, S. J. Ed.). Elsevier Science Publishers ltd. London. p207-223.

Babrauskas, V. et al. 1982. "Upholstered Furniture Heat Release Rates Measured With a Furniture Calorimeter" NBSIR 82-2604. National Bureau of Standards. Gaithersburg MD.

Babrauskas, V. and Peacock, R. D. 1992. "Heat Release Rate: The Single Most Important Variable in Fire Hazard" *Fire Safety Journal*. v18, p255-272.

Babrauskas, V. Peacock, R. D. and Janssens, M. 1991. "Standardisation of Formats and Presentation of Fire Data - the FDMS" *Fire and Materials*. v15, p85-92.

Babrauskas, V. Urbas, J and Richardson, L. 1992. Non-combustibility" *Heat Release in Fires*. (Babrauskas, V. and Grayson, S. J. Ed.). Elsevier Science Publishers ltd. London. p257-264.

Belles, D. W. 1985. "Full-scale Smoke Toxicity Tests in Furnished Rooms" *Fire Journal*. v79, No. 2. p35-41 and continued p83-85.

Cooper, L. Y. 1993. "Some Factors Affecting the Design of a Furniture Calorimeter Hood and Exhaust" NISTIR 5298. National Institute of Standards and Technology. Gaithersburg MD.

Cortese, R. and Perlee, H. 1992. "Robust Bi-directional Turbulence Probe" *Review of Scientific Instruments*. v63, No. 7, p3729-3732.

Dietenberger, M. A. 1991. "Modifications to Furniture Fire Model for HAZARD System" NIST-GCR 92-601. National Institute of Standards and Technology. Gaithersburg MD.

Dingyi, H. 1987. "Evaluation of Quarter-scale Compartment Fire Modeling for Constant and Stepped Heat Inputs" *Fire and Materials*. v11, No. 4, p179-190.

Drysdale, D. D. 1985. "An Introduction to Fire Dynamics" A Wiley-Interscience Publication. St Edmundsbury Press, Suffolk, UK.

Emmons, H. W. 1988. "Vent Flows" Chapter 1-8 *SFPE Handbook of Fire Protection Engineering*. Society of Fire Protection Engineers. Quincy, MA.

Harmanthy, T. Z. and Mehaffey, J. R. 1982. "Normalised Heat Load: A Key Parameter in Fire Safety Design" *Fire and Materials*. v6, No. 1, 1982.

ISO. 1975. "Fire Tests - Vocabulary" ISO 3261. International Standards Organisation.

Janssens, M. 1992 "Determining Flame Spread Properties from Cone Calorimeter Measurements" *Heat Release in Fires*. (Babrauskas, V. and Grayson, S. J. Ed.). Elsevier Science Publishers ltd. London. p265-306.

Janssens, M. and Parker, W. J. 1992 "Oxygen Consumption Calorimetry" *Heat Release in Fires*. (Babrauskas, V. and Grayson, S. J. Ed.). Elsevier Science Publishers ltd. London. p31-59.

Jones, W. and Peacock, R. D. 1989. "Refinement and Experimental Verification of a Model for Fire Growth and Smoke Transport" *Fire Safety Science - Proceedings of the Second International Symposium*. p897-906.

Karlsson, B. and Magnusson, S. 1992 "An Example Room Fire Model" *Heat Release in Fires*. (Babrauskas, V. and Grayson, S. J. Ed.). Elsevier Science Publishers ltd. London. p159-171.

Kent, L. A. and Schneider, M. E. 19??. "The Design and Application of Bi-directional Velocity Probes for Measurements in Large Pool Fires" *ISA Transactions*. v26, No. 4, p25-32.

Lawson, J. R. Walton, W. D. and Twilley, W. H. 1983. "Fire Performance of Furnishings as Measured in the NBS Furniture Calorimeter" Part 1. NBSIR 83-2787. National Bureau of Standards. Gaithersburg MD.

Lee, B. T. 1985. "Standard Room Fire Test Development at the National Bureau of Standards" *Fire Safety: Science and Engineering*, ASTM STP 882. (T. Z. Harmanthy, Ed.). American Society for Testing and Materials, Philadelphia. p29-44.

Magnusson, S. E. and Sundstrom, B. 1985. "Combustible Linings and Room Fire Growth - A First Analysis" *Fire Safety: Science and Engineering* ASTM STP 882. (T. Z. Harmanthy, Ed.). American Society for Testing and Materials, Philadelphia. p45-69.

McCaffrey, B. J. Quintiere, J. G. and Harkleroad, M. F. 1981. "Estimating Room Temperatures and the Likelihood of Flashover Using Fire Test Data Correlations" *Fire Technology*. v17, No. 2, p98-119.

McCaffrey, B. J. and Rockett, J. A. 1977 "Static Pressure Measurements of Enclosure Fires" Journal of Research of the National Bureau of Standards, v82, No. 2 p107-117. National Bureau of Standards. Gaithersburg, MD.

Mikkola, E. 1992 "Ignitibility of Solid Materials" (Babrauskas, V. and Grayson, S. J. Ed.). Elsevier Science Publishers ltd. London. p225-232.

Mowrer, F. W. 1994. "Development of the Fire Data Management System" NIST-GCR-94-639. National Institute of Standards and Technology. Gaithersburg, MD.

Mowrer, F. W. and Williamson, R. B. 1987. "Estimating Room Temperatures from Fires along Walls and in Corners" *Fire Technology*. v23, p133-145.

Mulholland, G. W. 1988. "Smoke Production and Properties" Chapter 1-25 *SFPE Handbook of Fire Protection Engineering*. Society of Fire Protection Engineers. Quincy, MA.

Mulholland, G. W. Henzel, V. and Babrauskas, V. 1989. "The Effect of Scale on Smoke Emission" *Fire Safety Science - Proceedings of the Second International Symposium*. p347-357.

Nakaya, I. Tanaka, T. and Yoshida, M. 1986, "Doorway Flow Induced by a Propane Fire" *Fire Safety Journal*. v10, p185-195.

NBS. 1988. "Construction Drawings for the Cone Calorimeter" Revision C. Unpublished Data. National Bureau of Standards. Gaithersburg, MD.

Newmann, J. S. and Croce, P. A. 1979 "A Simple Aspirated Thermocouple for Use in Fires" *Journal of Fire and Flammability*. v10, p326-337.

Nussbaum, R. 1987. "Oxygen Consumption Measurements in the Cone Calorimeter: A Direct Comparison Between a Paramagnetic Cell and a High Temperature Cell" *Fire and Materials*. v11, p201-203.

Ostman, B. 1992. "Smoke and Soot" *Heat Release in Fires*. (Babrauskas, V. and Grayson, S. J. Ed.). Elsevier Science Publishers ltd. London. p233-250.

Ostmann, B. Svensson, I. and Blomqvist, J. 1985. "Comparison of Three Test Methods for Measuring Rate of Heat Release" *Fire and Materials*. v9, p176-184.

Ostmann, B. and Tsantaridis, L. D. 1991. "Smoke Production in the Cone Calorimeter and the Room Fire Test" *Fire Safety Journal*. v17 p27-43.

Parker, W. J. 1989. "Prediction of the Heat Release Rate of Douglas Fir" *Fire Safety Science - Proceedings of the Second International Symposium*. p337-346.

Paul, K. T. 1994. "Cone Calorimeter: Initial Experiences of Calibration and Use" *Fire Safety Journal*. v22, p67-87.

Prahl, J. and Emmons, H. W. 1975. "Fire Induced Flow Through an Opening" *Combustion and Flame*. v25, p369-385.

Quintiere, J. G. McCaffrey, B. J. and Den Braven, K. 197? "Experimental and Theoretical Analysis of Quasi-steady Small-scale Enclosure Fires" Colloquium on Fire and Explosion.

Quintiere, J.G. Steckler, K. and Corley, D. 1984. "An Assessment of Fire Induced Flows in Compartments" *Fire Science and Technology*. v4, No. 1, p1-14.

Redfern, J. P. 1989. "Rate of Heat Release Measurement Using the Cone Calorimeter" *Journal of Thermal Analysis*. v35 p1861-1877.

Sundstrom, B. 1986. "Full-scale Fire Testing of Upholstered Furniture and the Use of Test data" Elsevier Applied Science. p147-154.

Tewarson, A. and Newman, J. S. 1986. "Scale Effects on Fire Properties of Materials" *Fire Safety Science - Proceedings of the First International Symposium*. p451-471.

Umberto, F. 1989. "Large-scale Fire Reaction Tests" *Polymer Degradation and Stability*. v23, No. 4, p385-396.

Wickstrom, U. and Goransson, U. 1992. “Full-scale / Bench-scale Correlations of Wall and Ceiling Linings” *Fire and Materials*. v16, p15-22.

Wickstrom, U. Sundstrom, B. and Holmstedt, G. 1983. “The Development of a Full-scale Room Fire Test” *Fire Safety Journal*. v5, p191-197.

You, H. and Kung, H. 1984. “Strong Buoyant Plumes of Growing Rack Storage Fires” *Twentieth International Symposium on Combustion*. p1547-1554.

FIRE ENGINEERING RESEARCH REPORTS

95/1	**Full Residential Scale Backdraft**	**I. B. Bolliger**
95/2	**A Study of Full Scale Room Fire Experiments**	**P. A. Enright**
95/3	**Design of Load-bearing Light Steel Frame Walls for Fire Resistance**	**J. T. Gerlich**
95/4	**Full Scale Limited Ventilation Fire Experiments**	**D. J. Millar**
95/5	**An Analysis of Domestic Sprinkler Systems for Use in New Zealand**	**F. Rahmanian**

School of Engineering
University of Canterbury
Private Bag 4800
Christchurch, New Zealand

Phone 643 366-7001
Fax 643 364-2758