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Summary Because of the lack of valid information regarding the influence of bends on the thermal and hydraulic performance of flues, a series of preliminary experiments were conducted using a full-scale Perspex rig. Analysis of the results and observations enabled the development of a methodology capable of determining the reduction of flow rate arising from the presence of bends. The analysis has also shown that existing quasi-steady-state methods used for the design of these flue systems are valid. Consequently, it is possible to determine flue gas and flue wall temperatures with respect to height within a flue system containing bends operating under low burn rate conditions.

Preliminary experimental investigation into the influence of bends on the thermal and hydraulic performance of domestic flues

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List of symbols

$\theta_g(z)$ Flue gas excess temperature at location z (°C)
$\theta_{gi}(z)$ Flue gas excess temperature at the fire inlet (°C)
M Mass flow rate of flue gas (kg s^{-1})
ΔM Change in mass flow rate of flue gas (kg s^{-1})
R Thermal resistance of flue (m K W^{-1})
C Specific heat capacity of flue gas (J kg^{-1} K^{-1})
Re Reynolds number for flue gas
Δp Thermally induced stack pressure (Pa)
ρ Flue gas density (kg m^{-3})
T Excess flue gas temperature (K)
f Friction factor for flue surfaces
l Length of the flue (m)
u Flue gas velocity (m s^{-1})
g Gravitational acceleration (m s^{-2})
k_n Pressure loss factor
β Pressure loss enhancement coefficient (equation 13)
g Slope factor = $1/MRC$ (see equation 1)
$G(z)$ Flue gas temperature reduction factor

1 Introduction

At present there are few sources of valid information on the thermophysical processes within domestic type flue systems that include bends. The existence of non-isothermal heat flow regimes within straight flues, characterised by low mass flow rates, was reported by Daws[1]. It has been concluded from this source that if steady-state conditions are assumed to hold within a straight flue, then good estimates for flue gas temperatures, flue wall temperatures and stack pressures can be evaluated using existing theory. Thus, in investigating fluid flow, hydraulic and thermal performance in straight flues as well as those containing bends, existing theory for straight flues required was examined to determine whether it could be used for flues that included bends. The following paragraphs discuss the experimental approach adopted and the analysis of the results that were employed to evaluate the influence of the bends on the performance of the flues operating under low burn rate conditions typical of normal domestic operating conditions.

2 Background

Losses due to friction and bends are known to occur for upward-flowing gas–air mixtures within a flue system. These losses and temperature variations that occur in a straight flue have been investigated in some detail for full-scale straight flue systems where heated air has been used to simulate the flue gas mixtures[1].

The main conclusion from the analysis of these results for straight flues was that the variation in the excess bulk gas–air temperature $\theta_g(z)$ with respect to height z from the flue entrance can be expressed as

$$\theta_g(z) = \theta_{gi} \exp(-z/MRC) \qquad (1)$$

where θ_{gi} is the excess gas–air temperature (°C) when $z = 0$, M is the bulk mass flow rate of gas–air (kg s^{-1}), R is the thermal resistance of the flue per unit length, and C is the specific heat capacity of gas–air (J kg^{-1} K^{-1}).

To preserve the dimensional consistency of equation 1, the MRC product must have the dimensions of length. Also, it has been assumed that the specific heat capacity C of the gas–air mixture can be approximated to that of air. This can be justified by comparing specific heat capacity values for air and typical flue gas from solid-, oil- or gas-fuelled boilers using data obtained from reference 2.

The authors concede that equation 1 is intended only for steady-state periods of flue operation and cannot be used to predict gas temperatures during the transient burn-down and start-up periods of operation. In this regard there exists a model reported by Siddell and Pearson[3] that predicts gas temperatures within a chimney system during the transient mode of operation. However, owing to the difference in operation (on–off) of a gas-fired boiler, the heat output of which is constant during operation, and the fact that the model of Siddell and Pearson has been validated only for a 2 m length of flue, it follows that such a model would not represent the non-steady-state behaviour of domestic flue systems attached to solid-fuel-burning appliances. After a review of literature relevant to gas flow in vertical flues, it was decided to investigate the ability of equation 1 to predict gas temperatures in flue systems that included bend offsets typical of those found in domestic flue systems.

The decision to use equation 1 was the result of conclusions from an extensive study and analysis of data for full-scale straight flues of different diameters serving operational solid-fuel devices that included a situation in which one of the straight flues was modified to include a 30° bend offset[4].

3 Theoretical considerations

Whenever a fluid flowing in a straight pipe of uniform cross section meets a change of direction, the fluid within the central axial region having the highest velocity is subjected to a central force that causes the fluid to move towards the wall on the outside of the bend. This causes a secondary flow to occur at right angles to the axial flow. For pipes or flues of symmetrical cross section, this secondary flow has been shown usually to consist of a pair of helical vortices[5].

It has also been shown in the case of long bends that the axial and secondary flows become independent of the distance along the axis. This sort of bend system is rare in domestic flue designs, where short bends are common. In the case of short bends, it has been shown that the type and geometry of the bends influence the flow at and near the bend.

It is not the intention to develop a theoretical discussion on such flows round bends other than to recognise that all the information on the influence of short bends on fluid flow has by and large been obtained as the direct result of experimentation. It should also be pointed out that most of the published work has focused on turbulent and laminar flow types[5] and not on the possible transition state type of flow that exists between the laminar and turbulent regimes, i.e. within the region defined by the Reynolds number inequality $3 \times 10^3 \leq Re \leq 6 \times 10^3$, which is considered to exist in domestic flue systems operating at a low or slumbering burn rate.

As indicated in reference 6, the mathematical analysis of fluid flow around short bends is complex and cannot accommodate the direct effects of viscosity. Therefore, the data and information regarding such flow have been almost entirely experimental; in fact, most of the investigations have involved turbulent type flows rather than laminar flow. Such studies are not wholely related to the transition type and near-transition types of flow, and consequently there are few or no experimental data on this type of fluid flow behaviour.

As previously reported[1], the flow of flue gases within straight flues can be of a transitional nature where an intermediate type of flow behaviour is to be expected. This flow behaviour is also expected in flues of the same vertical height and operating conditions that incorporate bend offsets.

In the following an attempt is made to modify simple theoretical expressions that are compatible with physical parameters that can be measured experimentally. Such expressions will then be used along with equation 1 to estimate the influence of bends on the mass flow rate of flue gases during transitional and near-transitional type flow regimes.

4 The influence of bends

Assuming a quasi-steady-state or near-steady-state flow regime to operate within a flue, it can be shown[1] that the thermally induced stack pressure Δp_s within a vertical flue or one that includes a bend offset can be written as

$$\Delta p_s = \frac{\rho_N T_N g l}{T_0}\left[\frac{\ln(T_{gi}/T_{gl})}{\ln(\theta_{gi}/\theta_{gl})}\right] \tag{2}$$

In equation 2 l is the vertical height of flue system above a reference point, which is taken to be where the of hot fire gas enters the flue system.

On the assumption that equation 2 is valid for all the flue systems, i.e. for those including bends, it follows that the condition $\Delta p_{straight} = \Delta p_{bend}$ is approximately true for flues of the same vertical height and opening conditions, provided the following boundary conditions are maintained for the inlet gas temperatures:

$$(T_{gi})_{straight} = (T_{gi})_{bend}$$

$$(\theta_{gi})_{straight} = (\theta_{gi})_{bend}$$

where T is in K and θ is in °C.

We use these assumptions and the assumption that the pressure drops Δp due to bends and internal friction within the flue can be expressed as

$$\Delta p = \frac{1}{2}\rho u^2\left[\frac{4fl}{gd} + \sum k\right] \tag{3}$$

or

$$\Delta p = \frac{1}{2}\rho u^2 K_n$$

where ρ is the fluid density, u is the mean fluid velocity, l is the length of the flue, d is the diameter of the flue, f is the friction factor for the internal surface of the pipe, K is the sum $\sum k$ of pressure loss factors for bends, entrance and exit of flue, and $n = 1$ for a straight flue and 2 for flues including bends.

Expressions for the reduction in mass flow rate caused by the introduction of a bend into a flue system for flow regimes of interest are now developed where the hydraulic performance of a straight flue is compared with that of a flue containing a bend of the same physical height and internal cross-sectional area that is experiencing the same stack pressure difference Δp_s. In equation 3 let K_1 relate to the sum of friction, entrance and exit shock losses within a straight flue and K_2 represent the total losses within a similar flue system containing bends.

Thus, assuming the same flue gas entry excess temperature θ_{gi}, vertical height l and available stack pressures for each flue, it follows that

$$\Delta p_1 = \Delta p_2$$

so that

$$\frac{1}{2}\rho u_1^2 K_1 = \frac{1}{2}\rho u_2^2 K_2 \tag{4}$$

This means that flue gas velocities u_1 and u_2 can be related as

$$u_2 = u_1\sqrt{\frac{K_1}{K_2}} \tag{5}$$

Further, expressing the fluid mass flow rate M as $M = \rho A u$, where ρ is the density of flue gas, the ratio of mass flow rates is

$$\frac{M_2}{M_1} = \sqrt{\frac{K_1}{K_2}} \tag{6}$$

Thus, the change in mass flow ($\Delta M = M_1 - M_2$) due to the inclusion of bends can be written as

$$\frac{\Delta M}{M_1} = 1 - \sqrt{\frac{K_1}{K_2}} \tag{7}$$

where 1 refers to straight flues and 2 refers to flues containing bends.

Provided respective values for K_1 and K_2 for the straight flue and the flue including the bend offsets are available from the literature or elsewhere, the fractional change in mass flow rates $\Delta M/M_s$ resulting from the inclusion of a bend can be evaluated. Sadly, this route is not yet available owing to the lack of information on bend losses, as discussed earlier; thus an experimental approach was considered necessary.

Using equation 1 for a straight flue and for a flue containing a bend offset of the same vertical height l and assuming that

$$(\theta_{gi})_s = (\theta_{gi})_B \tag{8}$$

it follows that a graph based on equation 1

$$\ln\left(\frac{\theta_z}{\theta_{gi}}\right) = \frac{-z}{MRC} \tag{9}$$

if valid, will be a straight line the slope g for which yields $g = 1/MRC$. In this way, values for MRC for all flue systems can be generated. Assuming the RC products to be the same for all of the flue systems investigated, M_s and M_B can be determined from g_s and g_B, respectively, i.e. $M_s/M_B = g_B/g_s$ and

$$\left(1 - \frac{g_s}{g_B}\right) = \left(1 - \frac{M_B}{M_s}\right) = \frac{\Delta M}{M_s} \tag{10}$$

From equation 7 it follows that

$$\frac{\Delta M}{M_s} = \left(1 - \frac{g_s}{g_B}\right) = 1 - \sqrt{\frac{K_s}{K_B}}$$

and

$$\frac{K_s}{K_B} = \left(1 - \frac{\Delta M}{M_s}\right)^2 \tag{11}$$

Equation 11 can now be rewritten as

$$K_B = \frac{K_s}{[1 - (\Delta M/M_s)]^2} \tag{12}$$

Provided a suitable validation for equation 1 can be obtained for all the flue systems, values for K_B can be determined with respect to K_s for flue systems of the same vertical height and excess gas temperature entry conditions.

Assuming that

$$K_B = \beta K_s \tag{13}$$

and using equation 1 for both flue systems,

$$\theta_s(z) = \theta_{gi} \exp\left(-\frac{z}{M_s RC}\right)$$

$$\theta_B(z) = \theta_{gi} \exp\left(-\frac{z}{M_B RC}\right) \tag{14}$$

Using equation 6 it can be shown that

$$\frac{M_B}{M_s} = \sqrt{\frac{K_s}{K_B}} = \sqrt{\frac{1}{\beta}} \tag{15}$$

$$M_B = \frac{1}{\sqrt{\beta}} M_s$$

Thus, equation 15 can be arranged to give equation 16:

$$\frac{\theta_B(z)}{\theta_s(z)} = \exp\left(-\frac{(\sqrt{\beta} - 1)z}{M_s RC}\right) = \exp[-\sqrt{\beta} - 1)g_s z] \tag{16}$$

therefore

$$\theta_B(z) = \theta_s(z) \exp[-(\sqrt{\beta} - 1)g_s z] \tag{17}$$

where

$$g_s = \frac{1}{M_s RC}$$

In this way, provided g_s and $\theta_s(z)$ are known for the straight flue, values for equivalent temperatures $\theta_B(z)$ within flues containing bend offsets at the same vertical height above the entry to flue can be determined.

It must be noted that equation 17 can now be rewritten as

$$\theta_B(z) = \theta_s(z) G(z) \tag{18}$$

where

$$G(z) = \exp[-(\sqrt{\beta} - 1)g_s z]$$

With knowledge of values for g_s and β, graphs relating $G(z)$ to z for various combinations of g_s and β can be developed to enable a quick estimate of $\theta_B(z)$ to be determined using equation 18.

5 Experimental programme

5.1 Preliminary experiments

Prior to the planning and design of the main set of experiments, a series of flow visualization experiments were conducted to determine the most effective locations for the instrumentation. In the rig used, a sealed appliance was attached to a 140 mm internal diameter transparent plastic pipe system that included a variety of bend offsets. The gas flow and stack pressure were induced using a propane burner located within the fire box. The presence of smoke was simulated using a non-toxic smoke. The propane gas burner was set to produce an inlet excess gas temperature to the flue of ~100°C above ambient, which was considered to be fairly representative of a real fire burning at a very low or slumbering rate. Also, such gas temperatures were low enough not to cause damage to the plastic flue system.

Smoke was injected into the system for each of the flue bend configurations, so that a visual record of the flow patterns could be obtained. From these observations it was clear that the bend influenced the flows, causing skewing towards the outer side of the bend. There was also some limited visual evidence of weak secondary flows in the regions close to and at the bends.

It was concluded from these preliminary experiments that single gas bulk temperatures taken at a fixed location would not give an accurate record of the thermal conditions within the flue.

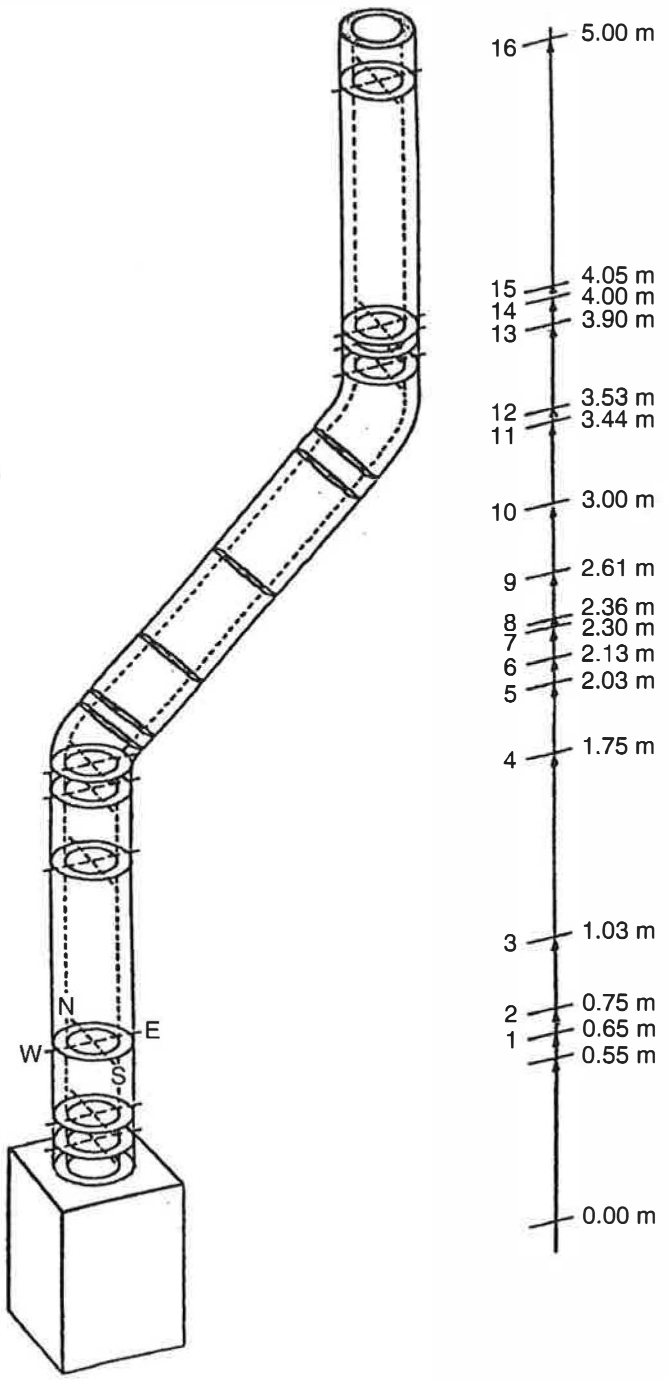

Figure 1 The locations of thermocouples used to analyse the flue with the 45° bend configuration

5.2 *Main programme*

For determination in a complete, pragmatic and economic manner of the influence of bend offsets on the hydraulic and thermal behaviour of the flue gases, a sequence of experiments were conducted on full-scale flue systems fabricated from 140 mm diameter plastic pipes, bend offsets and flexible piping of the same diameter. Figure 1 details the flue system including the 45° bend system, and Figure 2 shows the bend offsets chosen to represent the range of bend offsets found in practice.

On the basis of the observations and results obtained during the preliminary experiments, air was heated by a propane burner located within the fire box of a domestic appliance and used to simulate the hot combustion products. This assumption is based on the fact that the relevant thermal physical parameters such as thermal conductivity, viscosity and density of fire gases are similar to those of air[2] over a wide temperature range.

For simulation of the behaviour of flue gases for low burn rates, the excess gas inlet temperature was set at approximately 100°C above ambient so that, according to equation 8, all of the flue systems of same vertical height were operated under the same thermally induced hydraulic stack pressures according to equation 4. The position of the movable thermocouple used to collect flue gas temperature data at all levels is given in Figure 3. In the experiments an error of ±2 mm could be associated with location of sampling ports and ±2°C was the error recorded for temperatures.

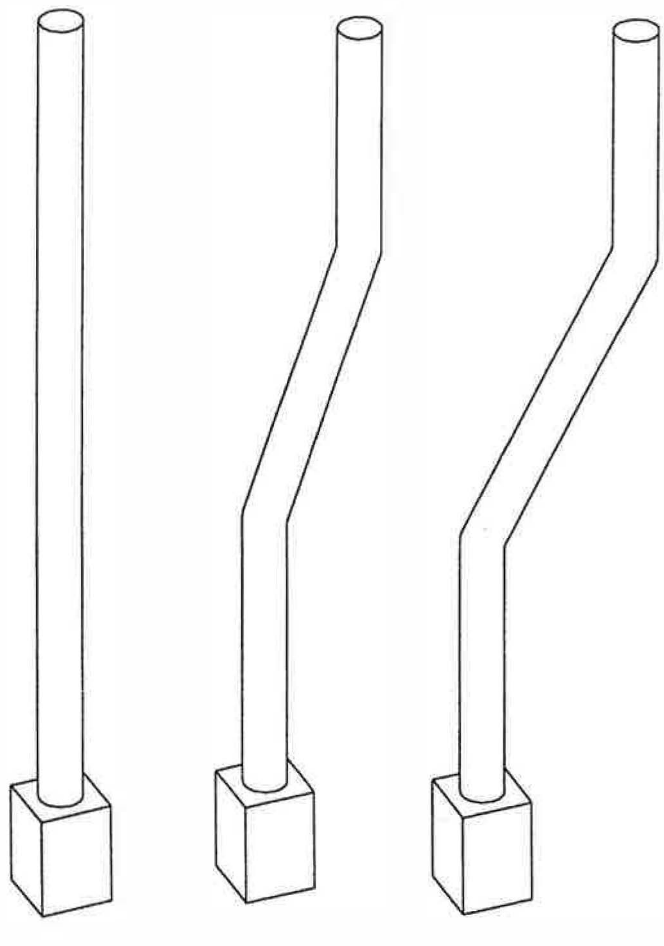

Figure 2 Other bend configurations used in the experimental analysis: (a) straight flue; (b) 22.5° bend arrangement; (c) 30° bend

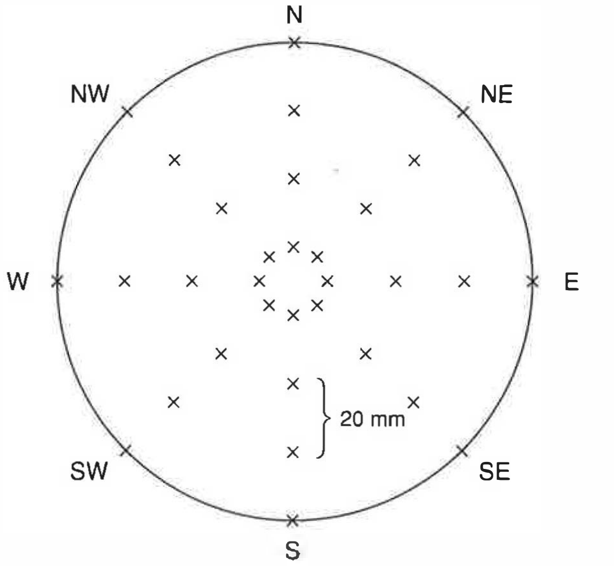

Figure 3 Cross-sectional view of temperature and velocity sampling ports

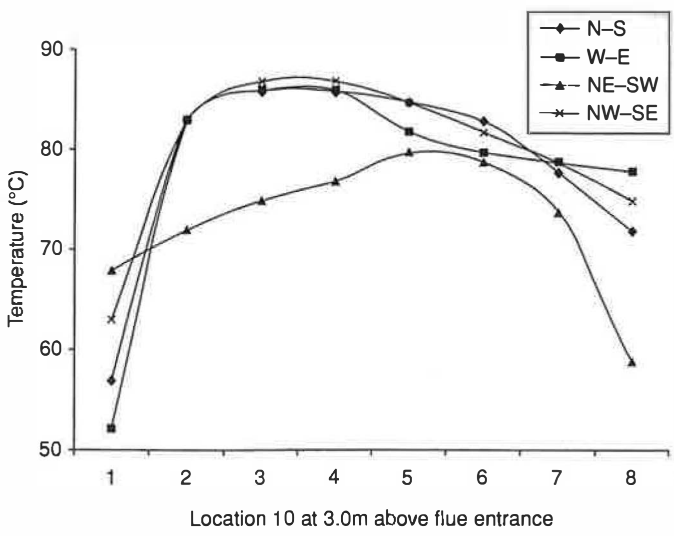

Figure 5 Temperature variation within the flue at location 10 in the 45° bend flue

In all of the experiments conducted, the data were collected manually over a period of 20 minutes during which it was assumed that the boundary conditions for all the flue systems under test remained constant.

6 Analysis of results

One of the main objectives of the experiments was to validate a simple analytical expression as given by equation 1, in order that data from flues containing bend offsets could be further analysed. In earlier experiments by Dick[7] and Daws[1], average bulk gas temperatures were used to validate equation 1. However, owing to the complex flow patterns observed and the wide variation of the flue gas temperature data noted in all of the full-scale rig tests, it was decided to explore other possible gas temperature parameters. Such variations in flue gas temperatures across the flue (see Figure 3) are clearly demonstrated in the thermal fields depicted in Figures 4, 5, 6 and 7 for a flue system containing a 45° bend offset (Figure 1) at four vertical locations (locations 5, 10, 15 and 16) within the flue. Such complex flow behaviour required a representative temperature that would satisfy the requirements imposed by

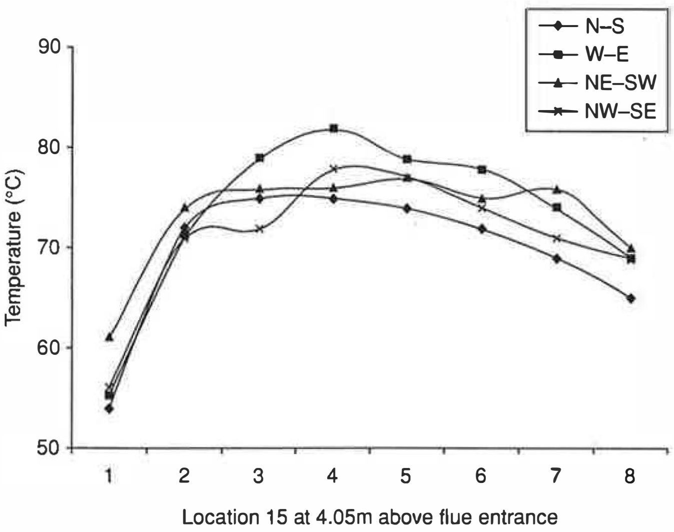

Figure 6 Temperature variation within the flue at location 15 in the 45° bend flue

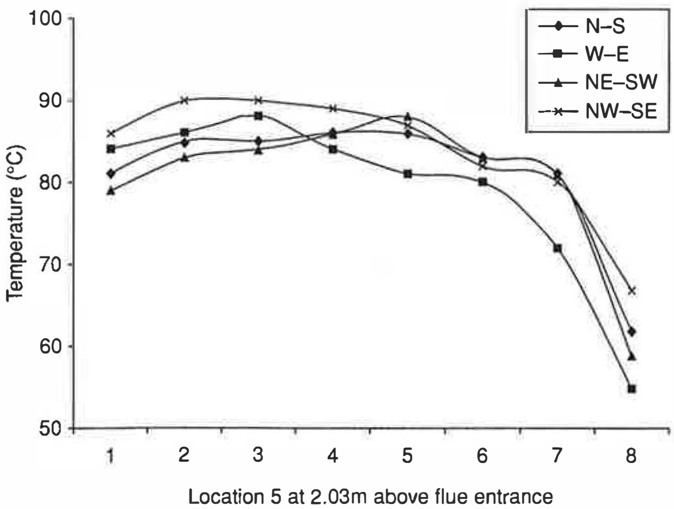

Figure 4 Temperature variation within the flue at location 5 in the 45° bend flue

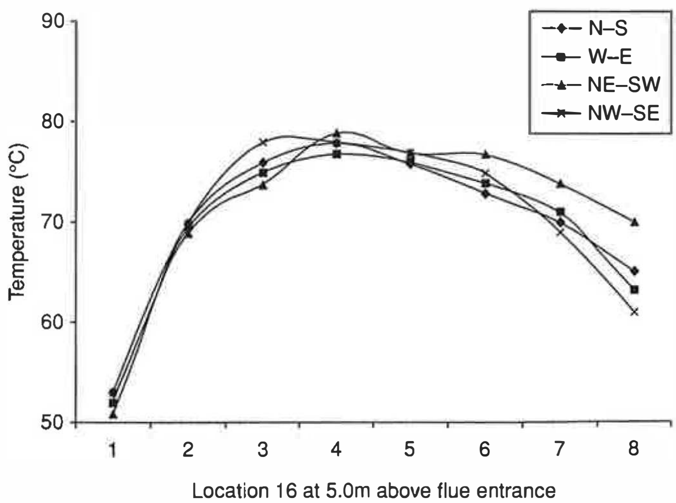

Figure 7 Temperature variation within the flue at location 16 in the 45° bend flue

equation 1. Consequently, two possible temperatures were chosen to represent the excess flue gas temperatures, i.e. the maximum and the average centre line temperatures.

Using average centre temperatures that are represented by the average of the eight centrally located thermocouples and maximum gas temperatures, two series of graphs based on equation 9 were drawn for each of the flues tested. In each case the excess gas temperature recorded at each location was divided by the inlet gas temperature. Also, it must be noted that the origin for the data is 0.55 m above floor level. Consequently, comparing Figures 8, 9, 10 and 11, where the maximum gas excess temperature data were used, with those using the average excess centre line temperature (Figures 12, 13, 14 and 15) indicates that the former would appear to fit more completely the temperature variation predicted by equation 1, which is a straight line passing through an origin at 0.55 mm. Note that the entrance to the flue system is located on a plane 0.55 m above ground level (Figure 1). Thus, if maximum gas temperatures are employed these can be satisfactorily analysed by equation 1.

Using the slopes of the straight lines given in Figures 8, 9, 10 and 11, respectively, values for the *MRC* products were obtained for each of the flue systems that include bend offsets.

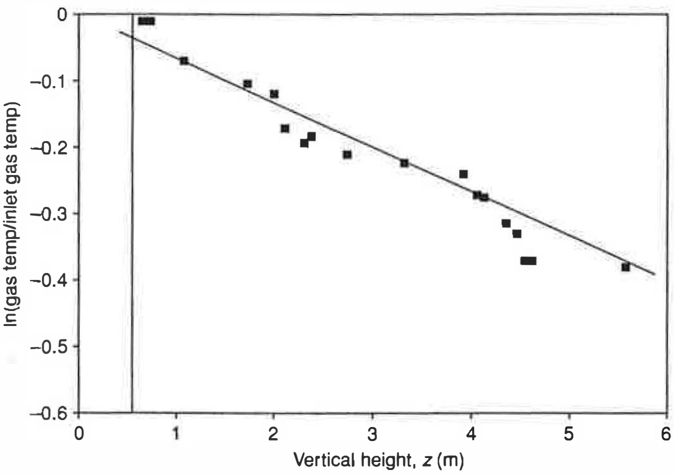

Figure 8 Temperature versus height graph for determining *MRC* value for a straight flue using maximum gas temperature: *MRC* = −*z*/[ln(gas temperature/inlet gas temperature)]

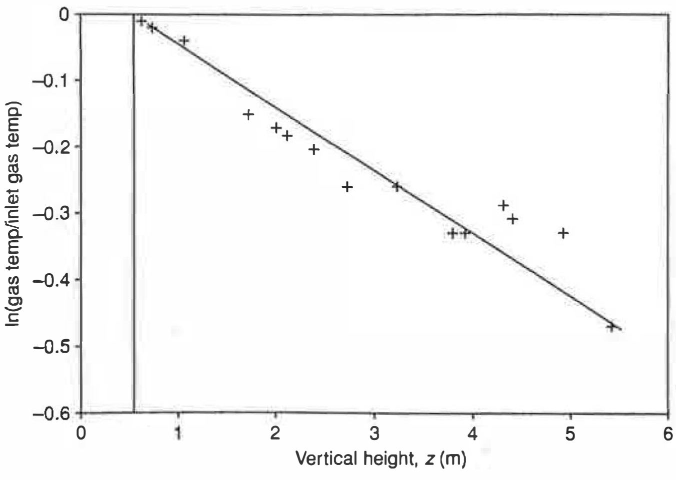

Figure 9 Temperature versus height graph for determining *MRC* value for a flue with 22.5° bend using maximum gas temperature: *MRC* = −*z*[ln(gas temperature/inlet gas temperature)]

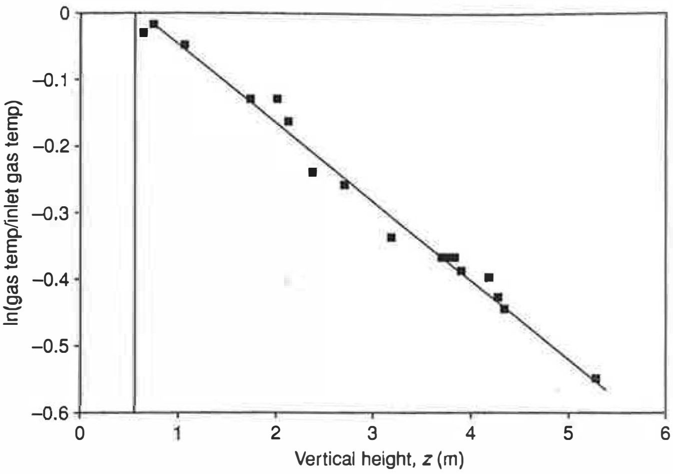

Figure 10 Temperature versus height graph for determining *MRC* value for a flue with 30° bend using maximum gas temperature: *MRC* = −*z*/[ln(gas temperature/inlet gas temperature)]

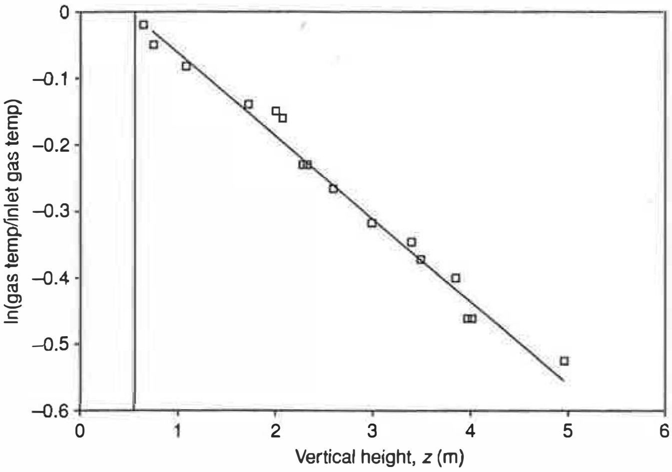

Figure 11 Temperature versus height graph for determining *MRC* value for a flue with 45° bend using maximum gas temperature: *MRC* = −*z*/[ln(gas temperature/inlet gas temperature)]

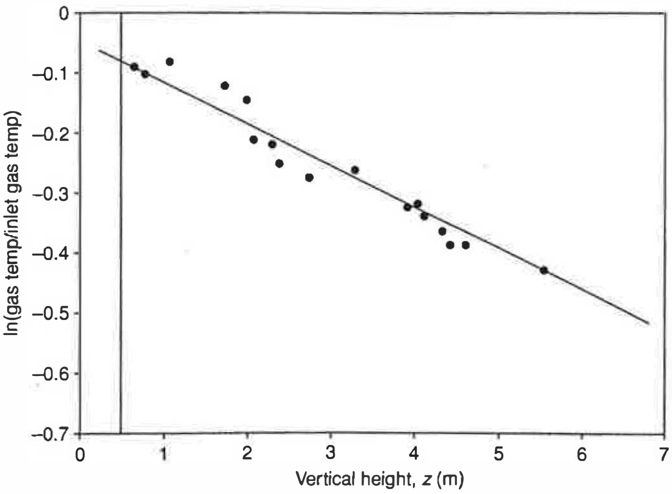

Figure 12 Temperature versus height graph for determining *MRC* value for a straight flue using centre line temperature: *MRC* = −*z*/[ln(gas temperature/inlet gas temperature)]

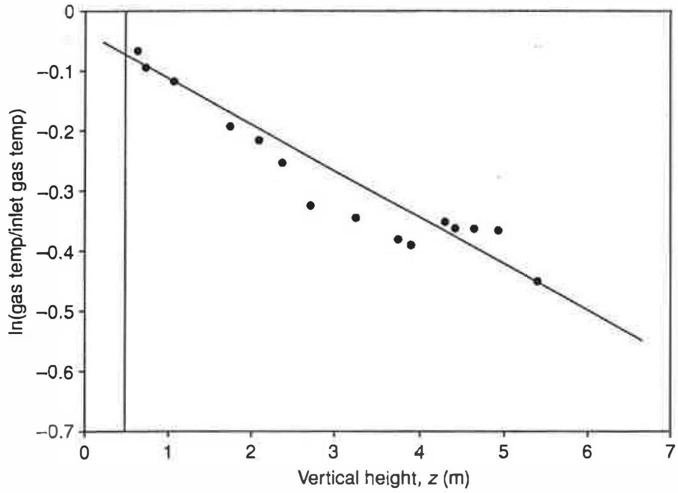

Figure 13 Temperature versus height graph for determining *MRC* value for a flue with 22.5° bend using maximum gas temperature: $MRC = -z/[\ln(\text{gas temperature/inlet gas temperature})]$

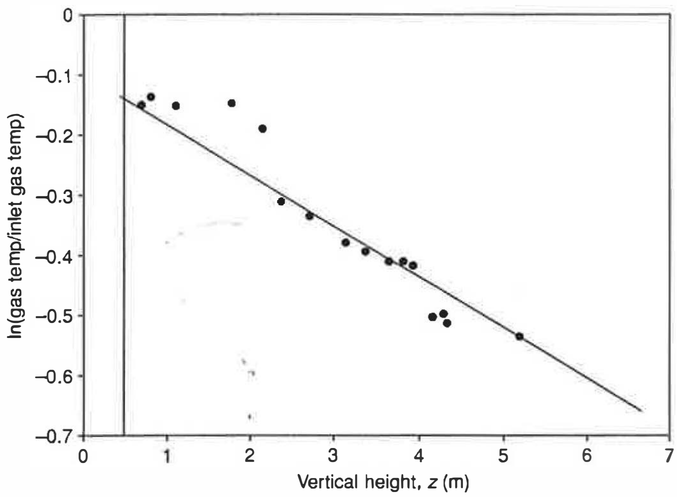

Figure 14 Temperature versus height graph for determining *MRC* value for a flue with 30° bend using maximum gas temperature: $MRC = -z/[\ln(\text{gas temperature/inlet gas temperature})]$

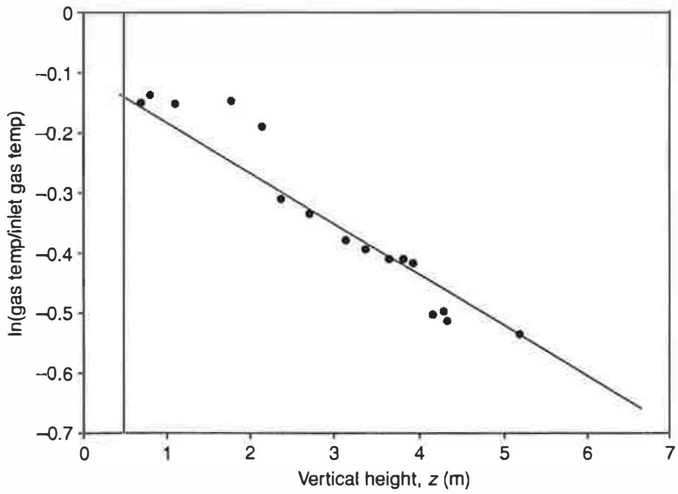

Figure 15 Temperature versus height graph for determining *MRC* value for a flue with 45° bend using maximum gas temperature: $MRC = -z/[\ln(\text{gas temperature/inlet gas temperature})]$

Table 1 *MRC* values using maximum gas temperatures

Type of flue	g (m^{-1})	*MRC* value (m)	Percentage reduction in *MRC* value, $(\Delta m/m_s) \times 100$	K_B	β
Straight	0.071	14.00			
22½° arrangement	0.096	10.75	23.0	$1.7K_s$	1.7
30° arrangement	0.118	8.47	39.5	$2.8K_s$	2.8
45° arrangement	0.124	8.06	42.4	$3.0K_s$	3.0

The values for the *MRC* product and g for each of the flue systems are given in Table 1. Also included in Table 1 is the traditional reduction in mass flow rate based on equation 10.

Values for K_B based on equation 12 are also given in Table 1 with respect to K_s for each of the flue systems tested. It must be noted that these values are for flues having a smooth internal surface attached to appliances operating at very low burn rates.

Using the information given in Table 1 and assuming that $K_B = \beta K_s$, as has been shown earlier in the text, provided maximum gas temperature information $\theta_s(z)_s$ exists or can be evaluated for the straight flue of the same vertical height as the flue system containing bends, then the maximum temperature $\theta_B(z)$ at any height (z) above the entrance for the flue containing a bend can be expressed as follows from equation 17:

$$\theta_B(z) = \theta_s(z)\exp[-(\sqrt{\beta} - 1)g_s z]$$

In equation 16, $\exp[-(\sqrt{\beta} - 1)g_s z]$ is the component $G(z)$ that reduces the maximum gas temperatures at location (z) within a straight flue of the same vertical height to the corresponding temperature $\theta_B(z)$ for the flue containing the bend offsets.

Using data for β given in the text or taken from elsewhere[2], the variation of $G(z)$ can be determined. For example, for a β value of 2 and a range of typical g_s, values of relevant $G(z)$ are as shown in Figure 16.

6.1 Sensitivity of $G(z)$ and $\theta_B(z)$

It is essential to determine the sensitivity of the function given by equation 17 to the range of experimental errors related to the key parameters g_s, β and z. Using a simple traditional approach, equations were determined that express the fractional experimental error expected for each function.

6.1.1 The $G(z)$ function

The following expression was determined for the fractional error in $G(z)$:

$$\frac{\Delta G(z)}{G(z)} = (\sqrt{\beta} - 1)g_s z \left\{\left[\left(\frac{1}{2}\frac{\Delta\beta}{\beta}\right)^2 + \left(\frac{\Delta g_s}{g_s}\right)^2 + \left(\frac{\Delta z}{z}\right)^2\right]^{1/2}\right\} \tag{19}$$

Assuming that $\Delta\beta/\beta = 0.1$, $\Delta g_s/g_s = 0.1$ and $\Delta z/z = 0.1$ are conservative values for the expected fractional errors when $g_s = 0.07\ \text{m}^{-1}$, $\beta = 3$ and $z = 5$ m, it can be shown that $\Delta G(z)/G(z) = 0.04$.

This implies that the function $G(z)$ is relatively insensitive to the expected experimental errors in the g_s, β and z parameters.

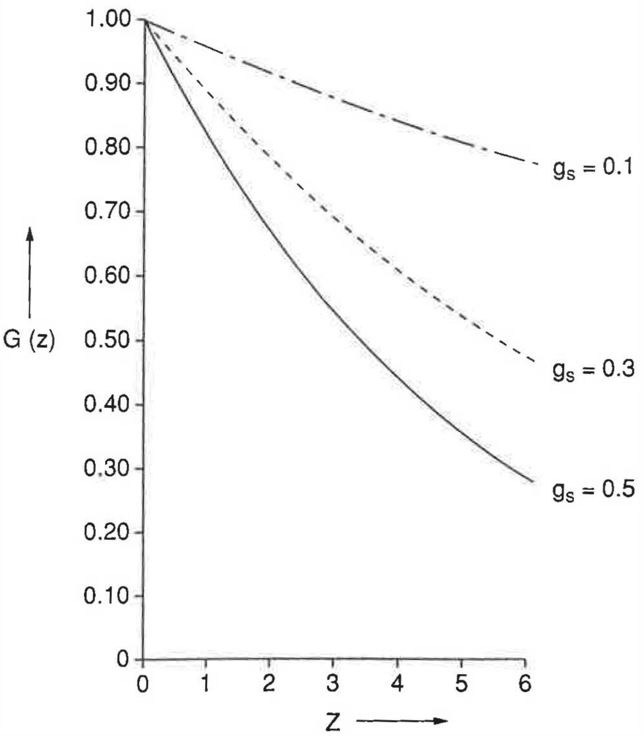

Figure 16 *G(z)* temperature function

6.1.2 The $\theta_B(z)$ function

Referring to equation 17 for $\theta_B(z)$, the fractional error $\Delta\theta_B/\theta_B(z)$ can be expressed as

$$\frac{\Delta\theta_B(z)}{\theta_B(z)} = \left[\left(\frac{\Delta\theta_s(z)}{\theta_s}\right)^2 + \left(\frac{\Delta G(z)}{G(z)}\right)^2\right]^{1/2} \tag{20}$$

Assuming that the error in $\theta_s(z)$ is ±2°C for the thermocouples used and that, for example, $\theta_s(z) = 60$°C, using these values and equation 20 it can be shown that

$$\frac{\Delta\theta_s(z)}{\theta_s(z)} = 0.033 \text{ and } \left(\frac{\Delta\theta_B(z)}{\theta_s(z)} \simeq 0.05\right)$$

Thus the expected overall experimental error in $\theta_B(z)$ will have a maximum value of approximately 6%.

7 Conclusions

— From this preliminary investigation, it has been possible to validate equation 1 for flues containing bends based on maximum excess flue gas temperatures.

— Making the assumption that low burn rates were in operation, it has been possible through the use of expressions based on equation 1, 3 and 4 to relate the total loss factor for a flue containing a bend to that of a straight flue of the same height operating under the same thermal boundary conditions.

— Using equation 1 and assuming that $K_B = \beta K_s$, it has been possible to relate $\theta_B(z)$ for the flue containing a bend to the equivalent gas temperature in a straight flue of the same vertical height and operating under the same thermal conditions via equation 17.

— Using experimental errors of ±2°C for temperature and ±2 mm for the z location, an overall maximum error of $\theta_s(z)$ of 6% can be expected.

8 Recommendations

— Since the experiments described above were preliminary, it is recommended that these studies be extended to include higher burn rate and other offset bends variations.

— In the meantime, for design purposes equation 1 and equation 17 based on maximum gas temperature should be used to determine $\theta_B(z)$ from $\theta_s(z)$ values obtained using a methodology described in reference 2 and the β values given in Table 1 or taken from elsewhere for the burning conditions utilized in the test scenarios.

— It is recommended that the vast database associated with this experimental investigation, which is available in reference 4, be utilized to validate emerging computational fluid dynamics analysis developed to simulate the thermal performance of vertical flue systems that include bends.

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