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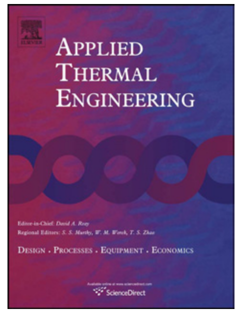

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Estimation of Radiation Losses from Sheathed Thermocouples

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Abstract

Thermocouples are often used for temperature measurements in heat exchangers. However if the radiation losses from a thermocouple in a high temperature gas flow to colder surroundings are ignored significant errors can occur. Even at moderate temperature differences, these can be significant. Prediction of radiation losses from theory can be problematic, especially in situations where there are large variations in the measured temperatures as the emissivity and radiative heat transfer coefficient of the thermocouple are not constant. The following approach combines experimental results with established empirical relationships to estimate losses due to radiation in an annular heat exchanger at temperatures up to 950°C.

Keywords: thermocouple, radiation loss, annular heat exchanger, steam boiler

Nomenclature

a	Dimensionless constant	
A	Cross sectional area	m²
c_p	Specific heat capacity	J/kg K
d	Hydraulic diameter / characteristic length	m
h	Local convective heat transfer coefficient	W/m² K
k	Thermal conductivity	W/m K
m	Mass flow rate	kg/s
p	Pressure	N/m²
q	Heat transfer rate	W
T	Temperature	K
U	Overall heat transfer coefficient	W/m² K
v	Velocity	m/s
ε	Emissivity	
μ	Dynamic viscosity	kg / ms
ρ	Density	kg/m³
σ	Stefan-Boltzman constant	W/m² K^4

Subscripts

d	Hydraulic diameter / characteristic length
S	Sheathed thermocouple
T	Thermocouple, measured temperature

U Unsheathed thermocouple
W Evaluation at wall condition
∞ Evaluation at free stream condition

Dimensionless Groups

Nu	Nusselt Number	$\left(\frac{h\,d}{k}\right)$
Pr	Prandtl Number	$\left(\frac{c_p\,\mu}{k}\right)$
Re	Reynolds Number	$\left(\frac{\rho\,d\,v}{\mu}\right)$

1 Introduction

The authors undertook an extensive experimental investigation into the performance of shell and tube steam boilers as part of a Knowledge Transfer Partnership between the University of Leeds and Dennis Baldwin & Sons Ltd (1), a manufacturer of such boilers. One element of the investigation was the determination of the heat transfer properties of an individual boiler tube, unpublished due to reasons of commercial confidentiality. To investigate this, the authors employed an annular test rig with hot gases flowing through the centre tube, being cooled by water flowing through the annular space. The authors recognised that the thermocouples used to measure the temperature of the hot gases would under-record the gas temperature due to radiation losses from the thermocouples to the cold wall of the tube. Using the same test rig with two different types of thermocouple, an experiment to assess these losses was undertaken. The results of this experiment were combined with other experimental data from the test rig in order to refine the assessment of the losses. They were then validated by comparing them with the results obtained by Roderick et al. (2).

The gas temperatures at inlet and outlet were measured using K-type thermocouples supplied by Endress and Hauser Ltd. These consisted of a hot junction encased in a magnesium oxide insulation material packed into a 6mm Inconel sheath. Their measurement accuracy conformed to BS EN 60584-2:1993 Class 2 (3).

There were large temperature differences (up to ~900°C) between the gas stream and the wall of the tube which caused large radiation losses from the thermocouples. This resulted in the measured gas temperature being lower than the true gas temperature. One method of estimating the radiation losses is by deriving the heat transfer coefficient over the thermocouple using an empirical correlation which can then be used in a convection gain radiation balance equation. Silvani and Morandini (4) when undertaking temperature measurements in wildland fires, using 250 μm diameter sheathed thermocouples, employed this method, modelling the thermocouple as a tube in cross flow.

McAdams (5) proposed that the error could be estimated by measuring the apparent gas temperature using several different sized thermocouples and extrapolating to zero. Brohez et al. (6) employed two thermocouples with different sized beads and observed reductions of up to 200°C when investigating losses from unshielded thermocouples in compartment fires with true gas temperatures of 927°C. Lönnermark et al. (7) also used this method when comparing temperature measurements from thermocouples with those of a fibre Bragg grating sensor. Additionally they noted that the choice of correction method is a balance of several factors including operating environment, robustness (of the thermocouple), complexity, and cost, amongst other factors. Furthermore, as Bradley and Matthews (8) note, the Reynolds number over the surface of the thermocouples also affects the convective heat transfer coefficient. In this experiment employing several thermocouples at the same time was impractical due to the design of the test rig and the variable flow conditions inside the tube. It was therefore decided to undertake a calibration exercise in order to develop a mathematical expression for compensation using two thermocouples of different diameter. Tagawa and Ohta (9), investigating temperature measurement in combustion using a two thermocouple probe, point out the necessity to have a large enough difference in size to generate a sufficient difference in response.

The present programme employed a 0.75 mm bead thermocouple, to be used as reference, supplied by the University of Oxford, via Rolls Royce plc. This was a K type thermocouple of similar initial construction to the Endress and Hauser unit. However, the end of the sheath had been machined off and the junction re-welded resulting in a thermocouple with a small bead.

2 Test Equipment

2.1 Heat Exchanger Description

The test rig consisted of an annular parallel flow heat exchanger. A drawing of the test rig is shown in Figure 1 (extraneous features omitted) and the legend in Table 1. The inner tube was a 44.3 mm bore tube with a 3.2 mm wall standard boiler tube. This was jacketed by a section of 200 mm nominal bore pipe to form the annulus. Hot exhaust gases flowed through the tube and water through the annular space. The working length was 2.905 m.

Figure 1: Isometric View of the Test Rig Working Section and Instrumentation

Table 1: Legend for Figure 1

1	Hot Gases from Furnace
2	K-type Thermocouple for Gas Temperature at Inlet
3	Mounting Point for Differential Pressure at Inlet
4	Water Inlets, 3 off, Equispaced
5	Mounting Point for Water Temperature at Inlet
6	Mounting Points for Wall Temperature at Inlet, 3 off, Equispaced
7	View Ports
8	Mounting Point for Water Temperature at Outlet
9	Mounting Points for Wall Temperature at Outlet, 3 off, Equispaced
10	Water Outlets, 3 off, Equispaced
11	Mounting Point for Differential Pressure at Outlet
12	K-type Thermocouple for Gas Temperature at Outlet
13	Gate Type Flow Control Valve
14	K-type Thermocouples for Mass Flow Rate
15	Orifice Plate for Mass Flow Measurement
16	Mounting Point for Oxygen Sensor
17	To Induced Draft Fan and Exhaust

2.2 The Gas Side Circuit

The exhaust gases were generated by a Nuway CL372H/L burner firing light fuel oil (35 second) into a refractory lined furnace. They were then drawn through the heat exchanger by a centrifugal fan located as the last element in the circuit prior to atmospheric exhaust. The mass flow rate through the rig was controlled using a gate valve and measured by a calibrated orifice plate. The gate valve and orifice plate were mounted downstream of the test rig working section.

2.3 The Water Side Circuit

Hot water at approximately 80°C was taken from the boilerhouse hotwell. The water temperature within the hotwell was maintained by a Spirax Sarco SA128 steam injector. It was pumped through the rig using a Grundfos CR2 positive displacement pump and returned to the tank. The water circuit was designed and sized to limit the temperature rise of the water through the rig to <10°C. The pump circulated approximately 2,000kg/hr of water. The mass of water in the hotwell was sufficient to absorb the added heat from the test rig without affecting the control of the steam injector.

2.4 Instrumentation:

The gas temperatures at inlet and outlet of the heat exchanger working section were measured using the previously described K-type thermocouples. These were mounted in the centre of the tube.

The thermocouples measuring the boiler tube wall temperature were mounted externally (on the water side) of the tube using Johnson Matthey Silverflo 55 silver solder. Three thermocouples were mounted at each end of the test length, two at 60° to the vertical and one at the bottom. They were mounted in the same plane as the gas thermocouples. The arrangement is shown in Figure 2. The inlet and outlet temperatures of the water were also measured using K-type thermocouples. All thermocouples were read using a Kane May KM330 digitally compensated reader.

Figure 2: Thermocouple Arrangement at Inlet to the Test Rig

The mass flow rate of the exhaust gases was measured using a calibrated corner tapped orifice plate manufactured in accordance with BS EN ISO 5167-2:2003 (10) coupled to an Endress and Hauser Deltabar PMD235 differential pressure cell ranged 0-100mbar. The gas temperature was measured just upstream and downstream of the orifice plate using sheathed K-type thermocouples mounted in the centre of the flow path. The average temperature was used in the density calculation. The orifice plate and housing were heavily insulated to minimise heat loss and therefore radiation losses between the thermocouples and the wall were neglected. The upstream absolute pressure was measured from the corner tapping using an Endress and Hauser Cerabar M pressure transmitter.

2.5 Modification for the Experiment

The rig was modified by replacing the sheathed thermocouple measuring the exhaust gas temperature at the entrance of the rig with the unsheathed one supplied by the University of Oxford. The two thermocouples measuring the exhaust gas temperatures (at inlet and outlet) were connected to an existing Iconics / Endress + Hauser SCADA package with a data collection rate of one second.

2.6 Test Method

The test rig was preset such that the mass flow rate of exhaust gas through the rig was relatively high (set to 50 kg/hr at an inlet temperature of 700°C) and the oxygen content at 4.0-4.5%. No settings were altered on the rig throughout the duration of the tests. This was to ensure as far as possible that the flow conditions at any given temperature would be the same for each test run. The same sheathed thermocouple was used to measure the exit temperature in both runs and was not moved between runs. The unsheathed thermocouple was used to measure the entry temperature for the first run.

The burner was fired and the data collection started. The rig was then left untouched until near steady state conditions were attained. The burner was then switched off and air drawn through the furnace and the rig. The temperature data were collected when the burner was firing and also when air was drawn through the rig. The test was then repeated but with the unsheathed thermocouple at entry replaced by a sheathed unit.

3 Results.

The test duration was slightly over three hours in both cases. The time plots (Figures 3 & 4) demonstrate a very similar profile for both test runs with a very rapid rise in temperature at the start of the test which then slowed to zero as the rig approached steady state. At this point the burner was switched off and a very rapid fall in temperature followed. This rate of change then reduced as the furnace was cooled by the air being drawn through it. Both thermocouples indicated the same temperature, 80°C, as the water jacket prior to the furnace being fired.

Figure 3: Time Plot of Unsheathed / Sheathed Thermocouples.

Figure 4: Time Plot of Sheathed / Sheathed Thermocouples.

The unsheathed thermocouple was more responsive to changes in temperature than the sheathed thermocouple. To avoid results being unduly affected by lag, all data below 530°C were removed prior to processing on the heating part of the curve and all data above 600°C on the cooling part. Some lag effects are still evident on the first test run (unsheathed thermocouple) in the range 500-600°C. However, it was considered more consistent to draw the comparison based on data sets of the same range.
The exhaust gas entry versus exit temperature was plotted for each data set and a polynomial curve was fitted to each data set. These are shown in Figures 5 & 6. The data were normalised to the wall temperature, 80°C at which point the radiation losses were taken as zero. A second order polynomial equation was found to be the best fit to both data sets.

Figure 5: Exit and Entry Temperatures, Sheathed / Unsheathed Thermocouples.

Figure 6: Exit and Entry Temperatures, Sheathed / Sheathed Thermocouples.

By comparing the inlet temperatures of the sheathed and unsheathed thermocouples for a given exit temperature, an initial estimate of the radiation losses may be determined (Table 2). Plotting the difference between the two regression lines as a function of the inlet temperature indicated by the sheathed thermocouple, yields a further curve from which the losses may be determined as a function of the inlet temperature as measured by the sheathed thermocouple. This is shown in Figure 7.

Table 2: Calculated Losses.

Poly. Fit:	T_U = -2.8648x10^{-04}(T-80)^2 + 1.6912(T-80) + 80	T_S = -5.2697x10^{-04}(T-80)^2 + 1.6896(T-80) + 80	
Exit Temperature, (/ ℃)	Calculated Inlet Temperature, (/ ℃) Unsheathed Thermocouple (Fig. 5)	Calculated Inlet Temperature, (/ ℃) Sheathed Thermocouple (Fig 6)	Loss Calculated for Sheathed Thermocouple (/ ℃)
80	80.0	80.0	0.0
132	167.2	166.4	0.8
200	278.8	275.2	3.6
300	438.2	426.2	12.0
400	591.8	566.7	25.1
500	739.7	696.7	43.0
590	867.9	804.6	63.3

Figure 7: Calculated Radiation Loss.

3.1 Losses from the Unsheathed Thermocouple

The above represents a simplistic estimation of the radiative loss by comparison only. The unsheathed thermocouple also suffered losses and these can be estimated using standard theory. The convection-radiation energy balance for a thermocouple may be stated as follows:

$$h\,A\,(T_\infty - T_T) = \sigma\,A\,\varepsilon\,(T_T^4 - T_W^4) \qquad (1)$$

By rearranging Equation 1, it can be seen that the energy balance for both cases may be directly compared, Equation 2. Thus the total losses from the sheathed thermocouple are the sum of the difference between the measured temperatures of the sheathed and unsheathed thermocouples and the additional loss from the unsheathed thermocouple.

$$T_\infty = \frac{\sigma\,\varepsilon_U\,(T_{TU}^4 - T_W^4)}{h_U} + T_{TU} = \frac{\sigma\,\varepsilon_S\,(T_{TS}^4 - T_W^4)}{h_S} + T_{TS} \qquad (2)$$

The convective heat transfer coefficient for the unsheathed thermocouple can be estimated using Whitaker's (11) correlation for the Nusselt number for gases and liquids flowing past spheres, as the bead of the unsheathed thermocouple closely resembled a sphere. Hence, the hydraulic diameter of the unsheathed thermocouple was taken as the diameter of the thermocouple bead. Whitaker's correlation was also used by Brohez et al. (6) in their study of radiation losses from thermocouples in the same manner.

$$Nu_U = 2 + (0.4\,Re_{dU}^{0.5} + 0.6\,Re_{dU}^{2/3})\,Pr_U^{0.4}\left(\frac{\mu_\infty}{\mu_w}\right)^{0.25} \qquad (3)$$

As the temperature difference between the stream and the wall of the thermocouple is small the viscosity term tends to unity and is neglected in the calculations.

Using Equation 3, with the thermophysical properties of the exhaust gas calculated at the temperatures in Table 2, the local heat transfer coefficient for the unsheathed thermocouple can be determined. Combining this with an emissivity of 0.8 the additional temperature loss for the unsheathed thermocouple can be calculated using Equation 1. The value of the emissivity was taken as that of lampblack, Holman (12) as the thermocouples were found to be covered in a thin layer of soot when removed from the test rig. The results of these calculations are shown in Table 3.

Table 3: Additional Loss due to the Unsheathed Thermocouple.

Mass Flow Rate, kg/s	0.0139				
Reference Temperature, Unsheathed Thermocouple, °C	278.8	438.2	591.8	739.7	867.9
Density, kg/m³	0.639	0.496	0.408	0.348	0.309
Dynamic Viscosity, kg/m s	2.71E-05	3.26E-05	3.75E-05	4.18E-05	4.54E-05
Thermal Conductivity, W/m K	0.0403	0.0497	0.0582	0.0659	0.0723
Velocity, m/s	14.10	18.18	22.10	25.88	29.15
Specific Heat Capacity, J/kg K	1,117	1,164	1,208	1,248	1,278
Tube Internal Diameter, m	0.0443				
Hydraulic Diameter, Unsheathed Thermocouple, m	0.00075				
Emissivity, Unsheathed Thermocouple	0.8				
Local Reynolds Number, Tube	14,715	12,228	10,646	9,546	8,802
Prandtl Number at Reference Temperature	0.75	0.76	0.78	0.79	0.80
Re, referenced to Unsheathed Thermocouple	249.1	207.0	180.2	161.6	149.0
Nu referenced to Unsheathed Thermocouple	28.8	26.0	24.2	22.8	21.9
Local Heat Transfer Coefficient, Unsheathed Thermocouple, W/m² K	1,549	1,725	1,876	2,008	2,112
δT, Unsheathed Thermocouple, K	2.3	6.3	13.2	23.4	36.1
Equivalent Sheathed Thermocouple Temperature, °C	275.2	426.2	566.7	696.7	804.6
Total Temperature Loss, K	5.9	18.3	38.3	66.4	99.4

The results for this calibration exercise were obtained using a constant mass flow rate. However, during the experimental test runs, the mass flow rate was varied in order to obtain a range of heat transfer results. Any change in the mass flow rate will cause a change in the local Reynolds number over the thermocouple and therefore change the local heat transfer coefficient and thus the radiation losses from the thermocouple, Bradley and Matthews (8). No data were available for the local heat transfer coefficients with respect to sheathed thermocouples, either from the literature or from the manufacturer. However it was possible to estimate the effect of changes in the mass flow rate via consideration of the ratio of the emissivity and local heat transfer coefficient over the thermocouple (which were both unknown) in conjunction with Equation 2. This ratio was calculated by determination of the true gas temperature using the results from the unsheathed thermocouple (with its radiation losses accounted for via Whittaker's (11) approximation. From this the effect of the change in the local Reynolds number over the sheathed thermocouple may be isolated and estimated.

Figure 8 plots the local heat transfer coefficient as a function of the local tube Reynolds number with a notional emissivity of 0.6. The data were taken from the test results for the same 0.0443m internal diameter tube that this test was conducted on. The curves represent the measured temperatures, i.e. before adjustment for losses. As can be seen, the coefficients vary as a function of temperature. In order to normalise the curves, the value of the emissivity was altered to account for its change with temperature. Once the emissivity of the sheathed thermocouple has been changed in this way, the data collapse onto a single curve. The results of this are shown in Figure 9 and the emissivities in Table 4. Whilst this exercise was performed for the sheathed thermocouple, it was not for the unsheathed. The change in emissivity of the sheathed thermocouple is less than ±5% of the middle value. If replicated on the unsheathed thermocouple this would lead to an error of less than ±2°C at the highest temperatures and has been neglected.

Table 4: Variation of notional emissivity with temperature

Measured Temperature / ℃	Emissivity
800	0.58
700	0.60
600	0.63
500	0.65

Figure 8: Local heat transfer coefficient as a function of the local Reynolds number

Figure 9: Local heat transfer as a function of the local Reynolds number and emissivity

4 Validation

In order to test the validity of the above method, comparison was made with measurements from a shell boiler. Tests were conducted on a Dennis Baldwin & Sons Ltd, Yorkshireman 2,000 kg/hr F&A100°C three pass wetback shell and tube steam boiler rated at 10.3 bar g fitted with a Dunphy TD35 Hi Lo dual fuel burner. The boiler is referred to subsequently by its serial number YSX2000/15.

4.1 Test Method

Thermocouples were placed inside the smokeboxes and reversal cell and were used to measure the gas temperatures at inlet and outlet of the firetubes. They were placed at the entrance and exit to selected tubes, such that they were in the centre of the tube and in the same plane as the tube plate outer face. The boiler was started and run up to pressure. Steam was then vented to atmosphere at a steady rate such that the burner held its firing position at the desired boiler pressure. The firing rates reflect the set up of the burner at commissioning. Once steady state had been achieved, the data collection was started. The tests were run for ½ to 1 hour. The thermocouples were moved from tube to tube between each set of tests so that the gas temperatures at the tube plates were mapped and then averaged.

4.2 Results

The measured average temperatures, compensated for radiation losses at entry and exit to the tube plates are stated in Table 5. The temperatures (high fire) at inlet to the tubes in the reversal cell are lower than the design temperatures of 980°C and 1,017°C, for natural gas and diesel oil respectively, corrected to the actual fuel input, calculated in accordance with BS EN 12953-1:2002 Section 6.1 (c) (13). The measured temperatures were compensated for radiation losses using the experimentally derived calculation method, described above. The calculated increases in temperature for natural gas and diesel oil at high fire were 99°C and 116°C respectively.
Roderick et al. (2) undertook a series of measurements of gas temperatures at the tube plates of a coal fired economic boiler. For these measurements they used a suction pyrometer and commented that in the temperature range that they were working in 1,600-1,800°F (870-980°C), the difference between "no suction" and "full suction" was approximately 200°F (111°C). The closeness of the results of the two methods indicates that there can be reasonable confidence in the compensation method used. The measured exhaust temperatures

also agree well with the predicted temperatures calculated by Dennis Baldwin and Sons Ltd (14) for the actual conditions. The greatest difference being less than 12°C.
The heat transfer rates were calculated in accordance with Equations 4, 5 & 6. The value of the specific heat capacity was determined using the gas composition with the values for the individual components taken from Rogers and Mayhew (15).

$$U = \frac{q}{A\,\delta T_{LM}} \qquad (4)$$

$$q = m\,(c_{pi} T_i - c_{po} T_o) \qquad (5)$$

$$c_p = fn(\%CO_2) + fn(\%H_2O) + fn(\%O_2) + fn(\%N_2) \qquad (6)$$

Table 5: Heat Transfer Results, Boiler Serial Number YSX2000/15

	Gas, hi fire	Oil, hi fire
Heat transferred in the first convective pass		
Temperature at inlet to the first bank of tubes (°C)	905.1	967.9
Temperature at outlet to the first bank of tubes (°C)	515.0	565.1
Heat Transfer Rate (kW)	298.3	339.8
Heat transferred in the second convective pass		
Temperature at inlet to the second bank of tubes (°C)	445.9	482.8
Temperature at outlet to the second bank of tubes (°C)	270.4	288.4
Heat Transfer Rate (kW)	118.7	146.7
Exhaust temperature (°C)	245.7	259.8
Predicted Exhaust Temperature (°C) (11)	247.8	248.5

With the exception of the exhaust temperatures, all the temperatures stated in Table 5 have been compensated for radiation losses using the method previously described. The exhaust temperatures are stated "as measured" as this location was considered to approximate to isothermal conditions.

Table 6: Heat Transfer Results for Both Tube Banks

	Gas, hi fire	Oil, hi fire
First convective pass, 0.0443 m internal diameter tubes		
Overall Heat Transfer Coefficient (W/m²K)	45.5	46.9
Overall HTC (Roderick et al) (W/m² K)	46.2	51.0
Heat Transfer Rate (W)	7,563	8,617
Mean Temperature (°C)	685.6	742.9
Reynolds Number	8,972	9,730
Prandtl Number	0.80	0.79
Nusselt Number by Dittus Boelter	31.3	33.3
Gas Side HTC by Dittus Boelter	45.2	49.7
Second convective pass, 0.0443 m internal diameter tubes		
Overall Heat Transfer Coefficient (W/m²K)	45.7	49.7
Overall HTC (Roderick et al) (W/m² K)	44.2	48.6
Heat Transfer Rate (W)	3,258	4,025
Mean Temperature (°C)	343.5	369.8
Reynolds Number	13,405	14,568
Prandtl Number	0.77	0.76
Nusselt Number by Dittus Boelter	42.6	45.4
Gas Side HTC by Dittus Boelter	42.7	46.8

Table 6 presents heat transfer data for 0.0443m bore tubes. The overall heat transfer coefficient was calculated between the saturation temperature of the water in the boiler and the mean gas temperature calculated as the sum of the saturation temperature plus the log mean temperature difference. Roderick et al. (2) calculated the overall heat transfer coefficient in the same fashion.

Roderick et al. (2) proposed that the overall heat transfer coefficient could be predicted using Equation 7 where the constant a has the value 0.022 with a 95% confidence limit of 0.002 based on 67 experimental runs with three different tube diameters. The values for the current work fall within these limits.

$$U = \frac{ak}{d} Re^{0.8} \qquad (7)$$

As Roderick states: "when the tubes are water cooled, the resistance to heat transfer of the metal itself and the boiling film on the water side are very small compared to the film on the gas side, so that [the gas side heat transfer coefficient] can be used as an overall coefficient between gas and water with negligible error". The Dittus-Boelter correlation (Equation 8) is quoted in BS EN 12953-3:2002 (13) for the determination of expected heat transfer rates. Comparing the values of the gas side and overall heat transfer rates, it can be seen that the gas side coefficients derived from the Dittus-Boelter expression are within 10% of the experimental, overall heat transfer coefficient values. Holman (16) states that the Dittus Boelter approximation usually agrees with actual data to within ±25%.

$$Nu = 0.023 Re^{0.8} Pr^{0.3} \qquad (8)$$

5 Discussion

This paper results from rigorous experimentation with data from well developed equipment. However, it is recognised that assumptions have had to be made, on which the outcome of the investigation depended. Of these, possibly the most important being the use of empirical correlations. While these have been employed widely over many years, in some cases, it is recognised that they are empirical and that their accuracy is limited. Blevins and Pitts (17) ascribe an uncertainty of ±25% to Nusselt numbers derived using Whitaker's (11) approximation. That said this additional uncertainty is less than 1% of the calculated true gas temperature in the current work.

A further assumption was the emissivity of the unsheathed thermocouple, (0.8); this is an assumed value. The actual emissivity at high temperature will be different from that at ambient temperature and moreover will progressively change with temperature. Whilst the uncertainty resulting from a lack of knowledge of the actual emissivity is small, determination of it would improve the accuracy of the results. As the emissivity of the thermocouples also changes over time due to oxidation and fouling when used in the rig, the emissivity would need to be determined immediately prior to the calibration exercise.

The ratio of the emissivity to the local heat transfer coefficient over the sheathed thermocouple was used to determine the effect of the variation in the mass flow rate on the radiation losses which occurred during the experimental programme. Knowledge of the absolute values for the emissivity and local heat transfer coefficient were not required since it was the ratio of the two that was under consideration. For this reason it was possible to use a notional value of the emissivity to demonstrate the effect of the change in mass flow rate and gas inlet temperature.

The thermocouples used were Class 2; these have an uncertainty of ±0.75% within the temperature range experienced. The use of Class 1 thermocouples would have reduced this to ±0.4%.

A further method of improving the accuracy of the calibration would be to employ McAdam's (5) extrapolation to zero method and perform three or more experimental runs with different sized thermocouples. However the length of time required for each run (>4 hours) makes this difficult to achieve in practice: the rig would have had to be shut down overnight, and the burner settings would have to have been changed in order to re-light it. Whilst it would have been preferable to have mounted the sheathed and the unsheathed thermocouples together in the same test run, this was not possible due to the design of the thermocouple mounts. Moreover this was considered undesirable due to the size of the thermocouple bodies (3 mm) relative to the tube bore (44.3 mm). To have changed the thermocouple mounts and fitted both thermocouples at the same time, for the calibration exercise, might have altered the thermal or fluid dynamic conditions. That said the development of a sheathed thermocouple with more than one sensing element is possible and would be of benefit. However, since it is the sheath which radiates heat, the thermocouples would need to be separately sheathed.

The positioning of the thermocouples is crucial in order to obtain accurate measurements. If the thermocouple is positioned away from the centre of the tube it will sense a lower temperature. The experimental exercise was performed after a series of modifications to the thermocouple mounting to ensure that the thermocouple was correctly positioned in the flow.

6 Conclusions

For thermocouples in a high temperature gas flow through a cooled tube, significant radiation losses (>10%) have been demonstrated in both a test rig and a shell boiler. A method of determining these losses has been proposed and validated using experimental data. This is further confirmed using comparable data taken from the scientific literature. Sources of error and methods for its attenuation have been discussed. Although the detail of this paper is specific to the conditions under consideration, the authors are confident that the methodology employed is applicable to other situations, in particular the approach taken to estimate the variation in radiation losses due to changes in the local Reynolds number over the sheathed thermocouple.

7 Acknowledgement

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References:

1 Dennis Baldwin & Sons Ltd, www.byworth.co.uk

2 Roderick, D.J.I., Murray, M.V, Wall, A.G. Heat Transfer and Draught Loss in the Tube Banks of Shell Boilers, .Inst. Fuel. Volume 32 No 225 1959, pp450-461.

3 BS EN 60584-2:1993 Thermocouples. Tolerances ISBN 0 580 20398 0

4 Silvani, X. Morandini, F. Fire spread experiments in the field: Temperature and heat flux measurements. Fire Safety Journal, Volume 44, 2009, pp279-285

5 McAdams, W.H. Heat Transmission, 3rd Ed.1954 McGraw Hill pp262-263.

6 Brohez, S. Delvosalle, C. Marlair, G. A two-thermocouples probe for radiation corrections of measured temperatures in compartment fires. Fire Safety Journal, Volume 39 2004, pp399-411

7 Lönnermark, A. Hedekvist, P.O. Ingason, H. Gas temperature measurements using fibre Bragg grating during experiments in a tunnel. Fire Safety Journal, Volume 43 2008, pp119-126

8 Bradley, D. and Matthews, K.J. Measurement of high gas temperatures with fine wire thermocouples. J. Mechanical Engineering Science, Volume 10 No 4 1968. pp299-305.

9 Tagawa, M. and Ohta, Y. Two-Thermocouple Probe for Fluctuating Temperature Measurement in Combustion--Rational Estimation of Mean and Fluctuating Time Constants. Combustion and Flame, Volume 109, Issue 4, June 1997, pp549-560

10 BS EN ISO 5167-2:2003 Measurement of fluid flow by means of pressure differential devices inserted in circular cross-section conduits running full - Part 2: Orifice plates (ISO 5167-2:2003) ISBN 0 580 41456 6

11 Whitaker S. Forced convection heat transfer correlations for flow in pipes, past flat plates, single cylinders, single spheres, and for flow in packed beds and tube bundles. AIChE J 1972 Vol 18 No 2:361–71

12 Holman, J.R. Heat Transfer, Fifth Edition, 1981, McGraw Hill, ISBN 0 07 029618-9. Table A-10 p548.

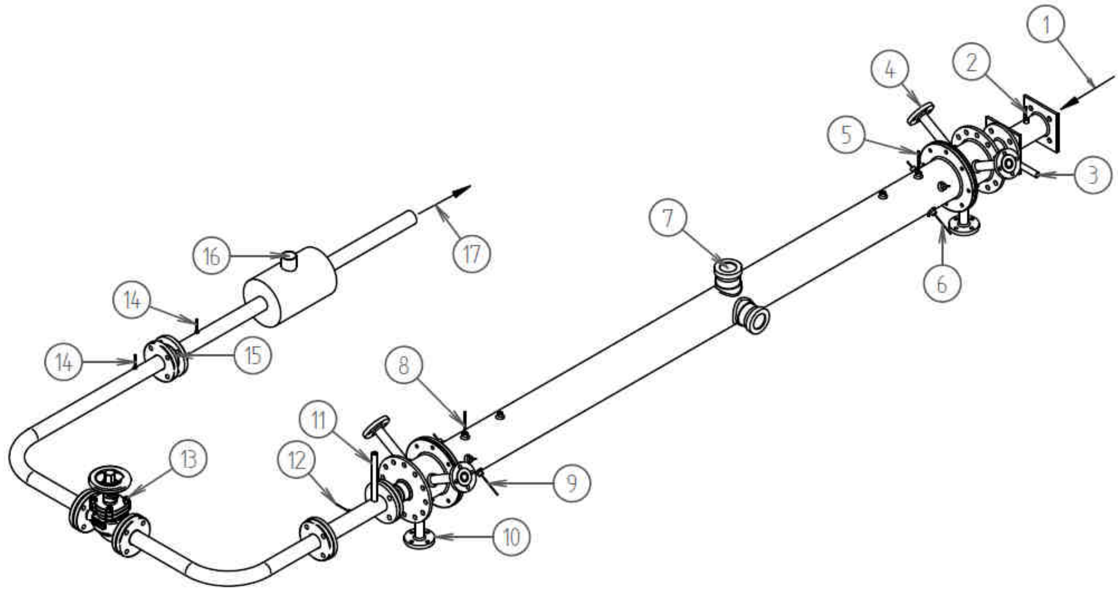
13 BS EN 12953-3:2002: Shell boilers - Part 3: Design and calculation for pressure parts. ISBN 0 580 39851 X. Section 6.1 (c), p9.

14 Dennis Baldwin & Sons Ltd Technical Department, Private communication.

15 G.F.C Rogers and Y.R. Mayhew: Thermodynamic and Transport Properties of Fluids, Fourth Edition 1987, Basil Blackwell. ISBN 0 631 90265 1 p15

16 Holman, J.R. Heat Transfer, Fifth Edition, 1981, McGraw Hill, ISBN 0 07 029618-9. p227.

17 Blevins, L.G. Pitts, W.M. Modeling of bare and aspirated thermocouples in compartment fires. Fire Safety Journal, Volume 33, 1999, pp239-259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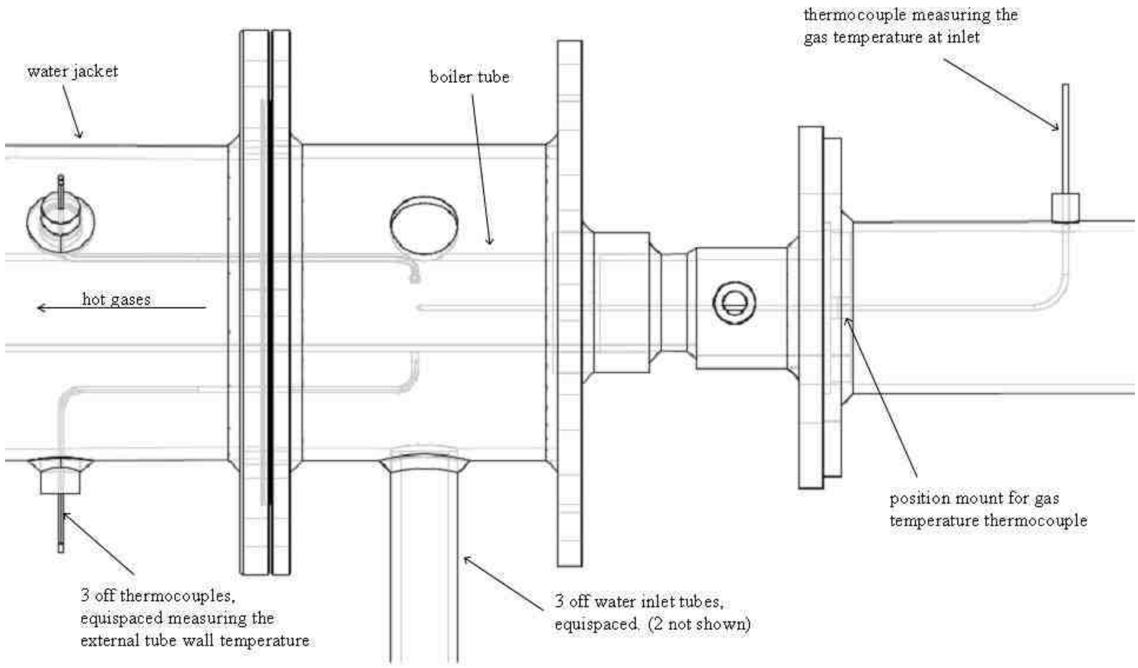
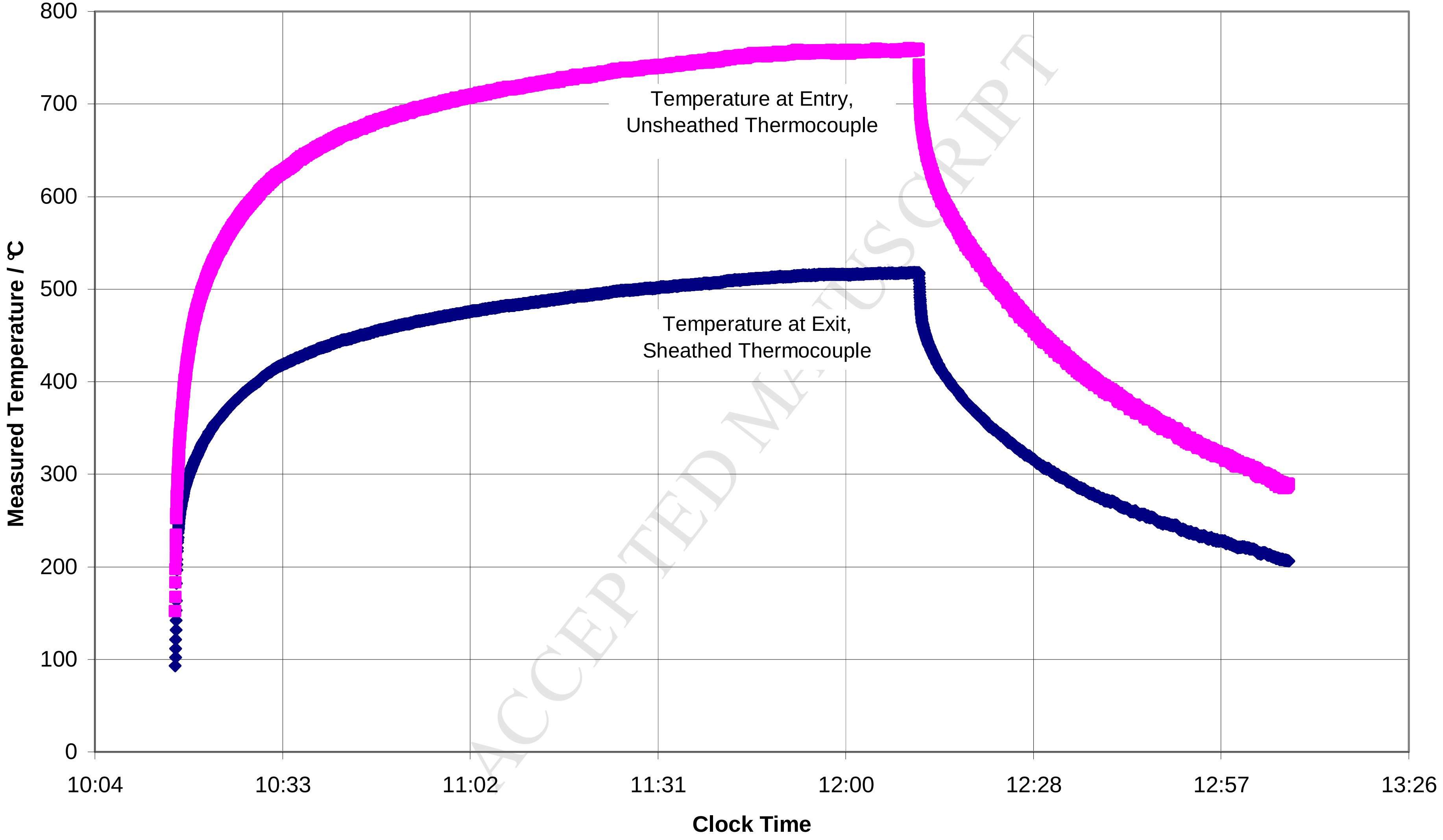
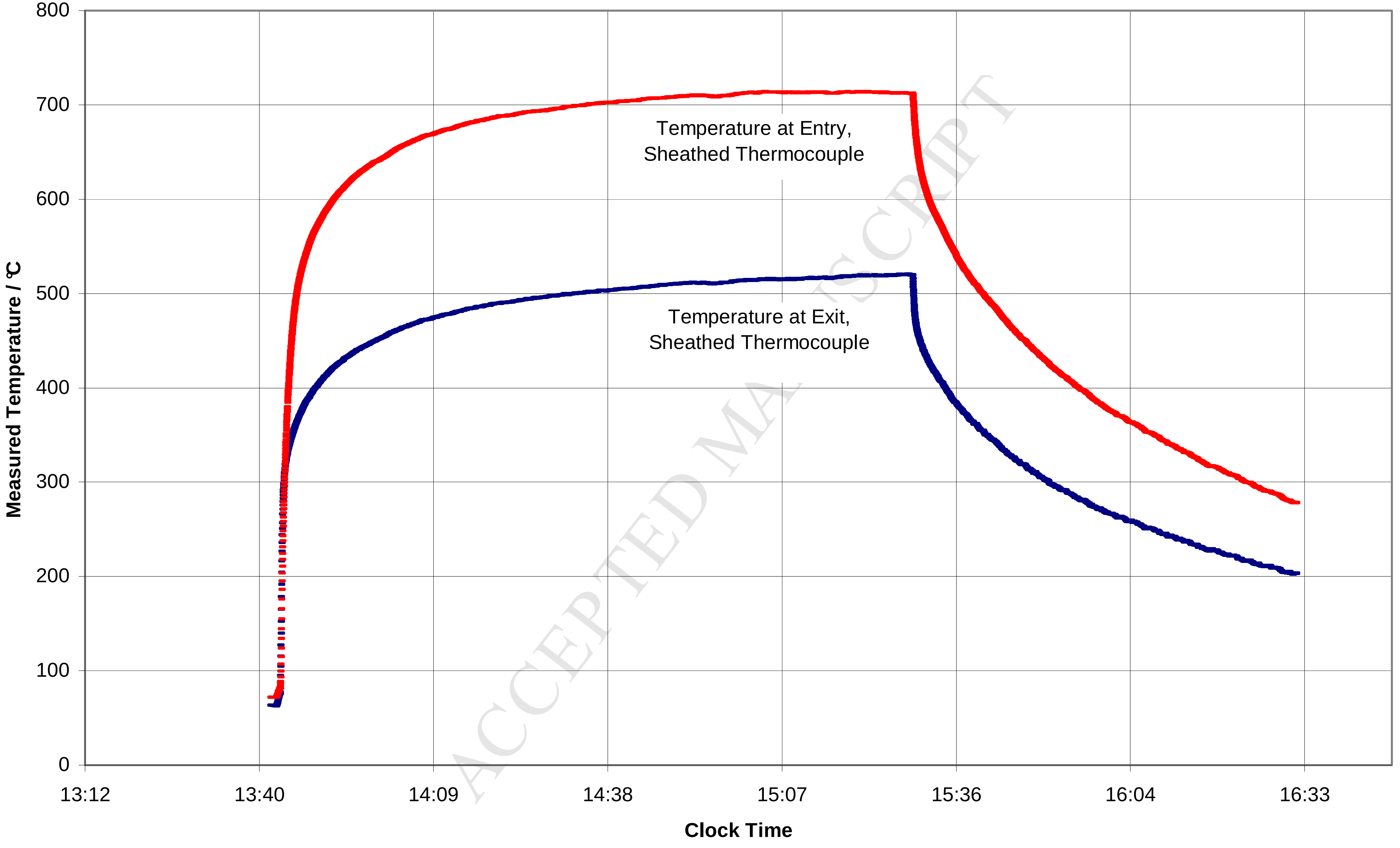


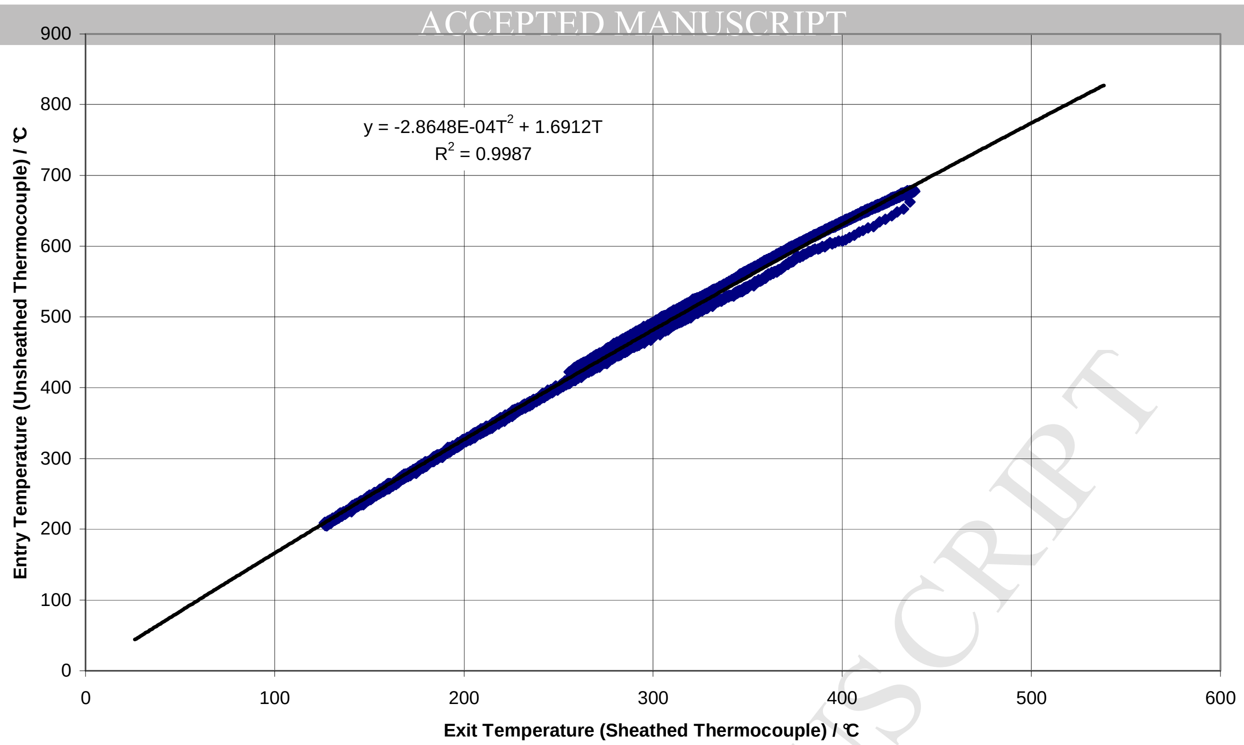
water jacket
boiler tube
thermocouple measuring the gas temperature at inlet
hot gases
position mount for gas temperature thermocouple
3 off thermocouples, equispaced measuring the external tube wall temperature
3 off water inlet tubes, equispaced. (2 not shown)

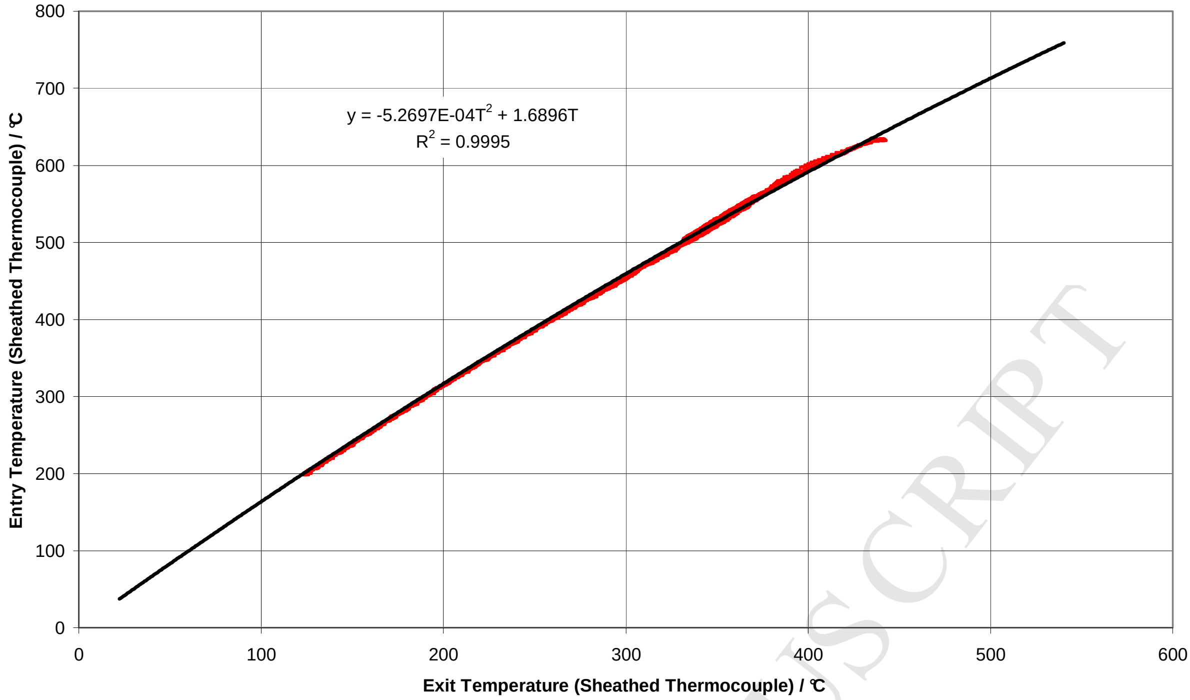


800
700
600
500
400
300
200
100
0
Measured Temperature / °C
10:04
10:33
11:02
11:31
12:00
12:28
12:57
13:26
Clock Time
Temperature at Entry,
Unsheathed Thermocouple
Temperature at Exit,
Sheathed Thermocouple

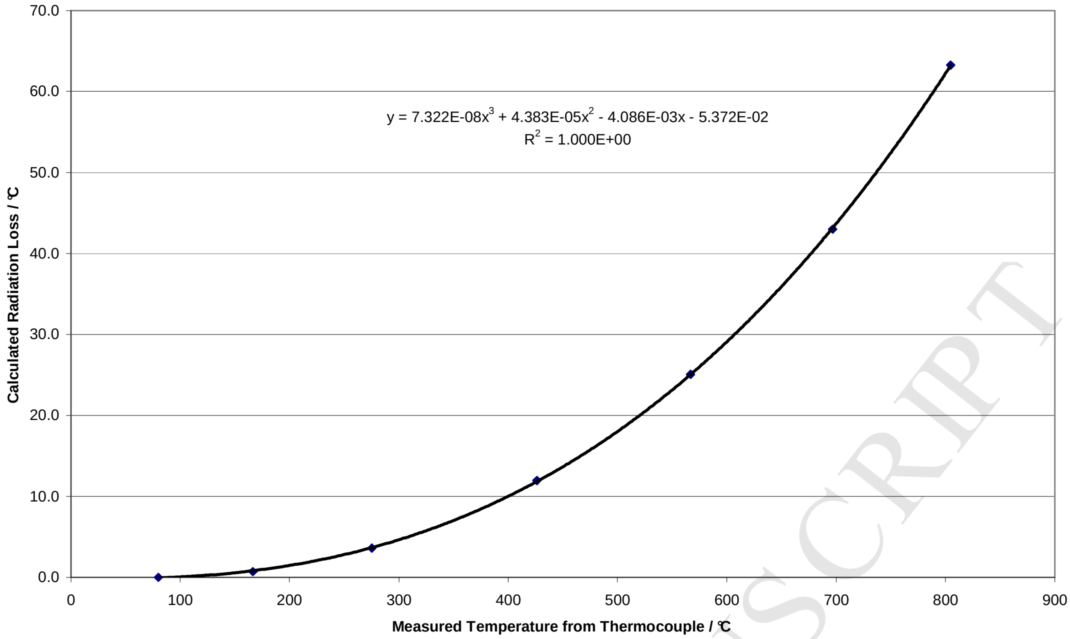


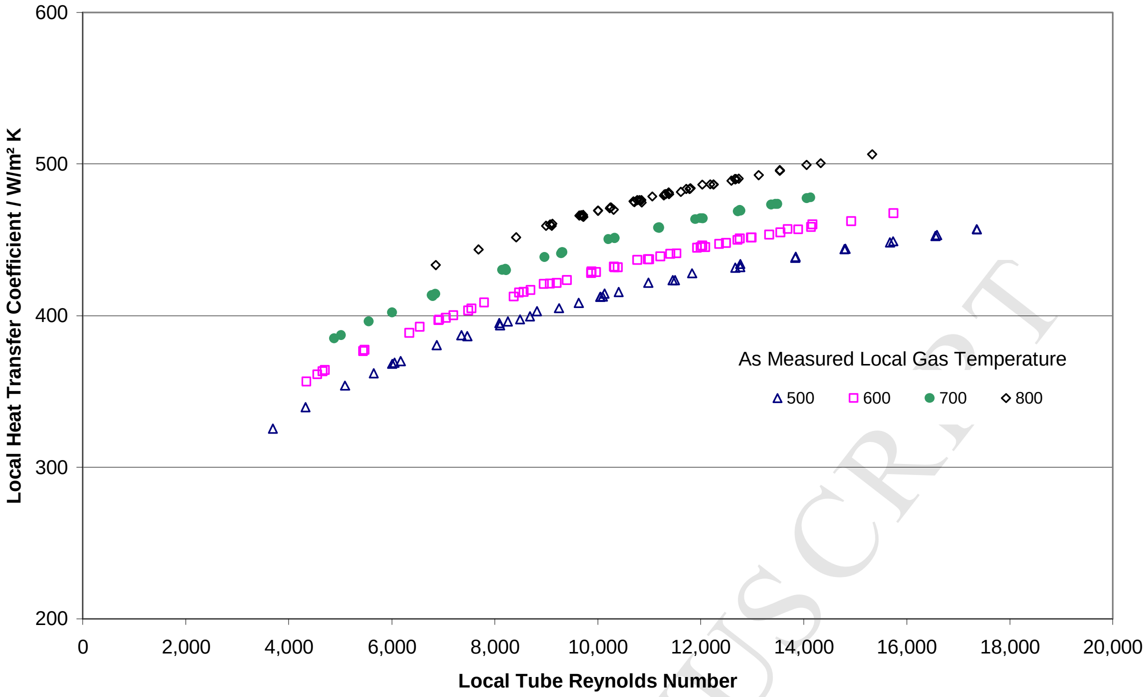
800
700
600
500
400
300
200
100
0
Measured Temperature / °C
13:12
13:40
14:09
14:38
15:07
15:36
16:04
16:33
Clock Time
Temperature at Entry, Sheathed Thermocouple
Temperature at Exit, Sheathed Thermocouple

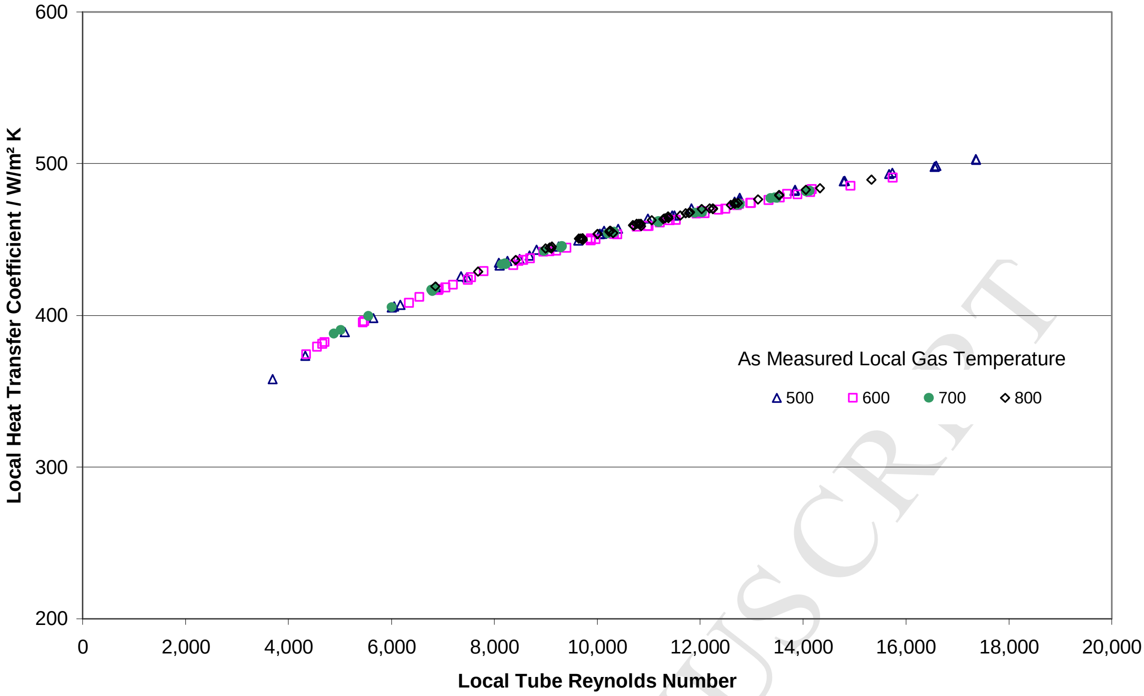

y = -2.8648E-04T^2 + 1.6912T
R^2 = 0.9987
Entry Temperature (Unsheathed Thermocouple) / ℃
Exit Temperature (Sheathed Thermocouple) / ℃
0
100
200
300
400
500
600
700
800
900
0
100
200
300
400
500
600



y = -5.2697E-04T² + 1.6896T
R² = 0.9995
Entry Temperature (Sheathed Thermocouple) / ℃
Exit Temperature (Sheathed Thermocouple) / ℃
0
100
200
300
400
500
600
700
800
0
100
200
300
400
500
600


70.0
60.0
50.0
40.0
30.0
20.0
10.0
0.0
0
100
200
300
400
500
600
700
800
900
y = 7.322E-08x3 + 4.383E-05x2 - 4.086E-03x - 5.372E-02
R2 = 1.000E+00
Calculated Radiation Loss / ℃
Measured Temperature from Thermocouple / ℃


600
500
400
300
200
Local Heat Transfer Coefficient / W/m² K
0
2,000
4,000
6,000
8,000
10,000
12,000
14,000
16,000
18,000
20,000
Local Tube Reynolds Number
As Measured Local Gas Temperature
500
600
700
800


600
500
400
300
200
Local Heat Transfer Coefficient / W/m² K
0
2,000
4,000
6,000
8,000
10,000
12,000
14,000
16,000
18,000
20,000
Local Tube Reynolds Number
As Measured Local Gas Temperature
500
600
700
800