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The Effect of Wall Emissivity on Radiator Heat Output

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Summary

The variation in the heat output of panel radiators obtained by altering the emissivity of the wall behind them has been examined. This work was conducted through both experiments and Computational Fluid Dynamics (CFD).

The results indicate that the presence of a high emissivity (black, such as the usual painted or wallpapered) surface to the wall increases the mass flow rate and air velocity behind the heat source compared to a reflective material. This is due to the radiation heat transfer to the wall creating an additional convecting surface behind the radiator. The results imply that the heat transfer rate can be increased by 20% through the use of a black instead of a reflective wall. The work concentrated on the air gap behind the radiator, so these results will not be directly applicable to a normal radiator. An extrapolation indicates that the output of single bank (plate) radiator will be increased by 10% and a double by 5%. Wall surface temperature results indicate that a reflective wall does indeed decrease the heat loss through the wall.

The trend shown in the data obtained from the CFD analysis agreed well with the experimental results. The flow and temperature plots obtained from the CFD work help to explain the heat exchange and fluid flow processes that take place between the radiator and the wall. This understanding should lead the engineer to a better consideration of radiator placement and design.

Nomenclature

A	Area	m^2
c_p	Specific heat capacity	$J\ Kg^{-1}\ K^{-1}$
$\dot{m}$	Mass flow rate	$kg\ s^{-1}$
$\dot{Q}$	Heat transfer rate	W
T	Temperature	°C or K
U	Thermal conductivity	$W\ m^{-2}\ K^{-1}$

Subscripts

1	Inlet condition
2	Outlet condition
a	Air (a_1 is air inlet condition, a_2 air outlet)
h	Enthalpy
rad	Radiation
tot	Total
w	Water
wall	Wall

1 Introduction

Radiators are the most popular central heating emitters in the UK. Steel panel radiators, usually equipped with convection fins to improve their heat output, are common in both domestic and industrial environments. Insulated aluminium foil is sometimes fixed to the wall behind radiators to reduce heat loss to the outside to improve the overall efficiency of the heating system. If there can be an improvement in heat transfer through altering the emissivity of the wall behind radiators, this would lead to a reduction in production costs since smaller radiators could be used for the same heat output.

The first law of thermodynamics states that in steady state heat flow, all of the energy put into a system must come out again. As radiators do no work, the heat transferred out must come from the hot water passing through them. Those which are described as *more efficient* cannot actually use less energy to provide the same amount of heat. The efficiency of radiators must be ascribed to a greater heat output from either the same area or the same mass of material.

Various solutions have been implemented that reduce the large amount of wall area taken up by panel radiators. The ducted air systems used in the recent past and skirting board radiators reduce the space lost to heating systems. Reducing the size of panel radiators would open up better possibilities to the Building Engineer and Architect as less wall space would be lost to these devices. It must not be forgotten that both better insulation and the superior control of ventilation have already reduced the requirements for heating in recent years.

This project was stimulated by work carried out by one of the authors [1] who built an ultra-high temperature heat exchanger with an inner ceramic tube. It was discovered that the heat which was transferred to the process fluid was greater than that predicted by convection heat transfer alone. This added heat transfer occurred because of radiation heat transfer which heated up the metal liner which in turn convected heat to the process fluid.

It was decided to examine the effect of different walls on the heat transfer to the air in the gap between the wall and the back of the radiator. A quick calculation using the standard radiation equation [2] shows that about 400 W m^{-2} can be transferred between a 70°C radiator and a 20°C wall. This is about 20% of the heat output of a conventional single radiator.

2 Heat Transfer

The major mechanism which accounts for over 60% of the heat transfer from radiators is natural (or free) convection. The steady flow energy equation for the air flow around a radiator (Figure 1) states that the rate of heat transfer from the radiator to the control volume is equal to the product of the flow rate of the air, the specific heat capacity of air and the change in temperature of the air across the radiator.

The rest of the heat transfer is due to thermal radiation. This is based on the fourth power of the absolute temperature, and the emissivity of the surface, which is a measure of how closely it approximates to a blackbody, which has an emissivity of 1.

2.1 Radiators

Radiators, with their energy input in the form of either electricity or hot water, are used to heat most homes and offices in the UK. Although they are called radiators, most of their heat output is by natural convection.

For example a 0.7 by 1.4 m single radiator has a heat output of 1800 W and a surface area of just under 1 m^2 per side [3]. The radiation heat transfer into the room is about 400 W. Therefore just under ¼ of the heat transfer is due to radiation. If account is taken of the side facing the wall, the proportion of heat transfer due to radiation rises to 45%. For a double radiator, these proportions are roughly halved.

Cast iron radiators have been used for well over a century. However, they are expensive and cumbersome, and have been superseded by steel and aluminium radiators. Steel radiators are fabricated from light gauge pressings welded together. For the same heat output, they are smaller and lighter and as their water content is generally lower, they have a faster response. More recently, die-cast aluminium radiators, lighter and requiring less water than steel radiators have been used in central heating systems. Steel radiators are more susceptible to corrosion than cast iron or aluminium radiators, but this is usually prevented by water treatment

The thermal performance of radiators is measured in accordance with EN 442-2 [4], which specifies a standard test room subject to certain test conditions. Part of the standard states that the emissivity of the paint used in the room is to be greater than 0.9. The actual output of an emitter will however vary based on the installation and operating condition [5]. Peach [6] recommends that comparison be made based on ‘specific product heat’, the heat emitted per unit mass of material. However, a heat output per wall area may actually be a better measure for the customer.

Several aspects of radiator design affect their output.

a) The use of metallic paint finishes can reduce the radiant component of radiator heat outputs by up to 10% [6].

b) The output of radiators can be slightly increased by decreasing their height above the ground and by increasing their spacing from the wall [7].

c) Decreasing the water flow rate through a radiator can lower the heat output [8,9].

d) The attachment of fins to panel radiators increases the convection heat transfer.

e) Different connection positions can affect the performance. The most common installation being with both connectors at the bottom (BOE). However introducing the flow at the top (TBOE) can improve the temperature distribution within the radiator and is used in the standard.

f) Facing the wall adjacent to the radiator with insulated reflector can lower the heat loss through the wall by 70% [10].

3 Experimental Work

A simplified domestic hot water central heating system was constructed for this work. A diagram of the experimental setup is shown in Figure 1. The heat input device was a 3 kW immersion heater. To keep the temperature steady (to within 1/2°C per hour), a variac was fitted to the heater to match the energy input to the heater with the output from the system. The flow rate was measured using a magneto-hydrodynamic flowmeter. The water was circulated using a standard central heating pump and controlled through the use of a bypass and valves.

After the pump two standard 600 x 600 mm single plate radiators were arranged in series with each other and supported using DEXION steel work. They were both positioned 150 mm above the floor. The water temperature before and after each of the radiators was measured with a total of three K-type thermocouples attached to the surface of the 15mm diameter copper piping, which was insulated with 10mm thick foam lagging. The piping was assumed to be at the same temperature as the water. The first thermocouple in the system was attached to a PC based data acquisition system so it was possible to monitor the temperature history of the system. All of these thermocouples were checked for accuracy with an independent probe.

The radiators were insulated to a thickness of 50 mm on the side that faced away from the wall using expanding foam insulation with a thermal conductivity of 0.04 W m^{-1} K^{-1}. The surface temperature of the insulation was measured, once again using K-type thermocouples. The heat loss through the rear of a radiator of this size is approximately 0.29 W K^{-1} or 14 W.

Two walls were constructed from 8mm thick plasterboard (k = 0.4 W m^{-1} K^{-1}) backed by 18 mm of expanded polystyrene insulation (k = 0.04 W m^{-1} K^{-1}). Five K-type thermocouples were buried into each of these to measure the surface temperature of the wall (see Figure 2). The average of these five temperatures is shown in Table 1 as T_{wall1}. As can be seen from Figure 2, these wall thermocouples allow a rough calculation of the temperature gradient up the radiator and also an indication into the level of edge effects. The temperature at the back of the wall was also recorded. It is shown as T_{wall2} in Table 1. These walls were positioned with a 50 mm air gap between them and the radiators.

For the tests described here, the wall behind radiator A, which was the first in line, was covered in a shiny aluminium foil surface of low emissivity. It was decided to put this one first in order to ensure that any claims about the change in heating effects would be conservative as its water inlet temperature would be slightly higher. The wall behind the second radiator, B, was left as a dull plasterboard surface of high emissivity.

The water flow rate was set to 1.0 l min^{-1} to ensure that the temperature drop of 2 to 3 °C across each radiator was large enough to give meaningful data on the heat loss from the surface.

Test were carried out to calculate the overall heat transfer to the air, and sufficient data was acquired to enable the energy balance for the whole system to be determined, The air temperature and velocity profiles at the middle of the top of both radiators were measured using a thermistor type anemometer, which had an error of ± 0.15 m s^{-1} as specified by the manufacturers. This device was also capable of providing temperature readings. The probe was shielded from radiation, which prevented the actual instrument from being placed within 2.5 mm of the wall. This limited its use in the narrow channels, particularly in the wall area, which would have been of particular interest for boundary layer investigation.

From Figure 3, it is clear that the heat lost by the radiator must either be conducted through the wall, reflected back into the radiator or convected into the air in the channel. A small amount will also be radiated into the room, but this is ignored for the analysis. Thus it is possible indirectly to measure the heat output of the radiator to the air using equation 1, shown below.

$$\dot{Q}_a = \dot{Q}_{a\,w} - \dot{Q}_{a\,cond} \qquad 1$$

Equation 1 can be expressed in terms of temperatures, specific heats, thermal resistances and flow rates as in equation 2.

$$\dot{Q}_a = \dot{m}_w c_{pw}(T_{w1} - T_{w2}) - AU(T_{wall1} - T_{wall2}) \qquad 2$$

This calculation of $\dot{Q}_a$ involves the propagation of uncertainty in the region of approximately ±10%. The principal source of this error is uncertainty in the thermocouple readings, which is close to 0.15°C and the difference between T_{w1} and T_{w2} in the calculation is of the order of 2 to 3°C.

Experimental Results

The calculated heat output to air for a 600mm $\times$ 600mm radiator is presented in the last column of Table 1. After correction for the heat loss at the front of the radiator, the total heat output of the radiator is 198 W with a high emisivity wall and 157 W with a low emisivity wall. This indicates that the shiny wall reduces the heat output by 21%. However, the increase in heat loss though the wall actually means that the heat transferred to the air is 181 and 149 W respectively, a decrease of 18% with a shiny wall. These figures for the heat output of the radiator are comparable with the performance data published by a radiator manufacturer [3]. It will be noted that the radiator with the dull wall is at a slightly lower temperature than the other one. This means that improvements in heat output shown above can be treated as somewhat conservative.

Figure 4 shows the temperature results measured at the top of the radiator. It will be seen that the temperature close to the radiator is almost independent of the wall emissivity. This is not surprising, as this will depend on the radiator temperature. On the wall side however the temperature increases from 30°C with a shiny surface to almost 50°C with a dull one. The velocity plots are shown in Figure 6. It will be seen that near the radiator, the results are similar for both wall finishes, but the dull surface increases the velocity in the middle of the gap by about 30%. At the wall, the measured velocity appears to be very similar, but the coarseness of the velocity readings may lose some of the velocity information near the boundary layer. Problems in measuring temperatures and in particular the low velocities encountered in this study are inherent in the nature of free convection. It is difficult to measure the velocities accurately without affecting the flow itself by the insertion of the measuring device.

Three wall temperatures were recorded halfway along the radiator, and these are shown in Figure 6. It will be seen that the maximum temperatures occur in the middle of the radiator. This is because the top and bottom thermocouples are able to *see* the room as well as the radiator. It will be seen that a shiny wall lowers the average surface temperature to 32°C from the 47°C seen with a dull one, which has major implications for the heat loss from rooms. This effect has been described by previous workers [10] and the results here support the published view that a shiny wall more than halves the heat transfer from a room.

The edge effects of the radiators were also investigated during the testing. Thermocouples placed in the plasterboard wall indicate that the surface temperature close to the edge of the surface directly opposite the radiator to around 10°C lower than the central temperature for a dull wall. This reduction in surface temperature results in lower flow rates and lower air temperatures at the exit to the channel. The effect of the edge on flow field and hence the overall heat transfer becomes more significant for the higher wall temperatures present when the wall emissivity is high.

4 CFD Modelling

To fully understand the effect of radiation on radiators, computer modelling was undertaken using the commercially available CFD package Fluent version 4.52 [11]. Two models were produced as a comparison with the experiments described above. Due to time and space constraints, both of these models were two-dimensional. 202 cells long by 178 high, almost 36000 nodes. The grid was non-linear, with a higher mesh concentration behind the radiator. The domain was 4 m long by 3 m high, to follow as far as possible EN422-2. The heat source was 60 cm high, 5 cm from wall 5 and 15 cm from the floor to reflect the experiment. The model is shown by the schematic diagram in Fig 7. A short convergence study was undertaken, which showed that the mesh is sufficient for a grid independent solution.

In this model the heat source, 7, was a constant temperature source set to 70°C. This was representative of the typical average surface temperature of domestic radiators and reproduced the experiment as far as possible. Wall zones 2 and 3 were set as walls of thermal conductivity 0.04 W m^{-1} $°C^{-1}$ to simulate the walls, ceiling and floor of a room. The surrounding wall zone 1 was set to a 20°C isothermal wall that would simulate the wall temperature of the European standard. Zones 4 and 5, the right hand edge of the radiator and the wall behind were set to an insulating, adiabatic wall. Wall zone 6 was defined as a conductor of 1800 W m^{-1} $°C^{-1}$ and emissivity 0.95. All other walls were given emissivities of 0.6. For the rest of the modelling only wall 5 was altered, its emissivity being set either to 0.05 (reflector) or 0.95 (black).

The problem involved all three modes of heat transfer and from a modelling point was an unusual one as neither pressure nor flow boundaries were specified. The entire circulation was buoyancy driven, up behind the radiator, and down at the far wall. Flow in all of the models was assumed to be turbulent and the RNG k-ε two equation turbulence model [12] was used with the default turbulence parameters. The near wall treatment used the two-layer zonal model with buoyancy terms. As radiation was a vital factor in the problem, the discrete transfer radiation model was implemented. It was difficult to get the model to converge, but it was found that starting with a laminar model until the residuals were all below 10^{-2} and then turning on the turbulence meant that the models converged in between 25 and 30,000 iterations. Decreasing the underrelaxation helped to stabilise the convergence and ultimately, all of the residuals were brought below 3×10^{-4}.

CFD Results

The heat transferred to the air from the radiator is shown in Table 2. The first column shows the heat transferred to the air, which was calculated from the difference in enthalpy times the mass flow rate between the entry and exit of the air behind the radiator. This shows that a change from a reflective to a black surface increases the heat transferred to the air from 98 to 136 W. This indicates that the shiny wall reduces the heat output by 27%. When looking at the heat transferred from the radiator Q_{tot}, the results are about the same as from the enthalpy calculations. The breakdown between radiation Q_{rad} and convection Q_{conv} is shown in Table 2. The convected heat is not altered by the wall covering, but the radiated output is reduced by three quarters, from 55 to 13 W . This is due to the fact that the high emissivity walls heat up by radiation and subsequently elevate the air temperature due to natural convection heat transfer.

The temperature profiles produced from the CFD at the top of the radiator are shown in Figure 4, along with the experimental results. It will be seen that the results, while not the same as the experiment, show the same trends, with the radiator wall temperature at about 70°C, the dull wall at about 50°C and the shiny one at around 32°C. Plots of the associated velocity profiles at the top of the radiator are found in Figure 5. It can be seen that close to the radiator wall, the velocity profile is not affected by the emissivity of the wall. However, from about halfway along the gap, the buoyancy provided by the hotter, dull wall, increases the velocity and hence mass flow rate relative to the shiny wall. Perhaps of equal importance, there is an increase of 14% in the total mass flow rate due to free convection at the back of the radiator. This is evidently due to the buoyancy on both sides of the air gap causing more air flow in the middle of the gap.

Figures 8a and 8b. show velocity profiles at various points on the radiator. It will be seen that at the bottom where the air enters via a right angle, the flow is concentrated towards the wall. On the radiator side, the hot boundary layer can be seen developing as the flow ascends. Near the dull wall, the boundary layer can also be seen to develop until, at the top of the radiator, there is a clear boundary layer at each side of the gap. Conversely, with a shiny wall, the velocity near the wall decreases and almost all of the convective heat transfer is seen to take place from the radiator.

Figure 6 shows the temperatures of both the dull and shiny walls facing the radiator. It will be seen that the shiny wall has an average temperature of about 28°C, whereas that of the dull wall is about 20°C hotter. An important feature of the temperature profile for the dull wall is the step change 5cm from the end of the radiator. This is because from this point, the wall can view the rest of the room (which is at 20°C) and its temperature is therefore depressed. It will be seen that these temperatures agree very well with the experimental ones. Indeed the CFD results show that the top and bottom thermocouples have been positioned in places where the temperatures have been lowered as the wall is not entirely covered from all angles by the hot radiator.

5 Discussion and Conclusions

The trends shown in the velocity results from the CFD are seen to agree quite well with those from the experiment (Figure 5). Near the radiator, in both the experimental and CFD cases, the velocity contours are similar for both shiny and dull surfaces. Closer to the wall, the results for the dull and shiny velocities diverge, with the dull, hotter surface imparting a higher velocity to the flow than the cooler, shiny surface. In the case of the experimental work, this additional velocity is shown as a rise in the centre of the gap, whereas the CFD sees the production of a high velocity boundary layer next to the wall. These differences arise due to the difficulties involved in modelling this sort of problem. The exact position of the laminar to turbulent breakdown point in free convection is very difficult to ascertain, and the CFD model has evidently produced this point much higher up the gap than occurs in the real situation.

There is greater agreement for the temperature difference (Figure 4). The temperatures near the wall is seen to agree well, showing the effect of radiation on wall temperature. Once again, for the reasons enumerated above, the CFD shows a steeper temperature gradient near the radiator than the experiment. However, there is no question that the emissivity of the back wall is a significant factor in the heat transfer from the rear surface of a radiator.

The experimental results in Table 1 indicate that there is a 20% increase in the heat output from the back of the radiator when the wall behind it has a higher emissivity. A linear extrapolation of this results indicates that the output of single bank (plate) radiator will be increased by 10% and a double by 5%. A low emissivity surface, results in less heat being removed from the radiator as the radiant energy is reflected straight back into the radiator. The greater increase in heat output seen in the CFD model can be put down partly to the difficulty of this form of modelling, and also the lack of a third dimension.

The process of heat transfer in the channel behind the radiator and its dependence on wall emissivity is now better understood. The CFD model indicates that the reason for the increased heat transfer is that the radiation from the heat source to the black wall causes it to be heated. This causes its surface temperature to rise above that of the incoming air, so heat is transferred by convection from the wall to the air. The hotter the wall surface is, the better the free convection becomes. These effects account both for the increase in heat transferred to the air and also the increased air velocity seen both computationally and experimentally.

The wall temperature readings indicate that if the reflector is the same size as the radiator, then this will reduce the heat loss through the wall adequately. However, if for aesthetic reasons the sheet is required to be smaller, it can (for a 50mm gap) be reduced by 60mm on each side and still adequately protect the wall from the peak temperatures. The values for the heat transfer trough the wall, are likely to be different in domestic situations, as although the wall is thicker, and its thermal conductivity less, the outside temperature will be lower than 20°C.

The fact that wall temperatures were measured allows the heat loss through the wall to be ascertained. This additional heat transfer to outside the room from wall mounted emitters is known as the back loss and is easily calculated [13]. The results described above may allow a correction to this as the equation shown in [13] uses the radiator temperature, which is correct for emitters mounted directly onto a wall, but too high for ones with an air gap where the wall temperature should be used.

It is appreciated that the quantity of heat output from radiators is not the sole design parameter. Two other important considerations are the thermal comfort that a radiator provides and also the time the emitter takes to warm up. The first of these was not examined in this study, but it is clear that a small, very hot area in a room is an undesirable effect. However, the increased velocity from a radiator that takes advantage of radiation will tend to distribute its heat better. Warm up time is important especially due to the fact that central heating systems usually cycle on and off as part of their control sequence. Domestic systems are usually turned off during the day and commercial ones at night. Smaller radiators that properly utilise radiation will decrease the warm up time by producing the same output from a smaller radiator, meaning that less water needs to be raised to the operating temperature. The fact that the radiation only really starts to be a major factor at higher temperature differences means that the output will be less than optimal unless the radiator is up to full operating temperature. This might make it seem like the system has more thermal inertia when it is at its maximum temperature and less when it is heating up; a desirable eventuality.

The fins inside modern double radiators could perhaps be replaced with a single black sheet of conducting material midway between the panels. This would in effect turn the double radiator into a triple plate radiator without increasing its size, saving on manufacturing costs. Currently in a double radiator, the fins typically cover half the radiator, meaning that some of the radiation is transmitted to the opposing bank.

Current thinking is to place a thin sheet between the wall and the radiator that is reflective on the wall side and absorbent on the radiator side. This work provides a strong indication that this is a good approach.

In conclusion, this work shows that putting a reflective sheet behind your radiator will reduce the running costs, but it will also reduce the heat output! It will certainly decrease the heat loss though the wall.

6 Acknowledgements

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List of Tables

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Emissivity	T_{w1} (ºC)	T_{w2} (ºC)	T_{wal11}(ºC)	T_{wall2} (ºC)	Q_{rad_tot} (W)	$Q_{condwall}$ (W)	Q_{tot} (W)	Q_a (W)
0.95 (dull)	70.8	67.7	47.2	25.0	59	17	198	181
0.05 (shiny)	73.3	70.8	32.4	22.3	5	8	157	149
% decrease					92	53	21	18

Table 1 Experimental Results Effect of changes in emissivity on Q_{tot} (after loss from front), the heat output of a radiator and Q_a, the heat transferred to the air.

Emissivity	Q_h (W)	$\dot{m}$ (kg s^{-1})	Q_{conv} (W)	Q_{rad} (W)	Q_{tot} (W)
0.95 (dull)	136	0.0191	77	55	132
0.05 (shiny)	98	0.0163	78	12	91
% decrease	27.4%	15%	-1.4%	77%	31%

Table 2 CFD results: Effect of changes in emissivity on the output of a radiator, Q_h (Based on enthalpy), $\dot{m}$ (mass flow rate) and Q_{conv} , the heat convected to the air,. Q_{rad} (the heat radiated) and Q_{tot} (the heat output from the radiator).

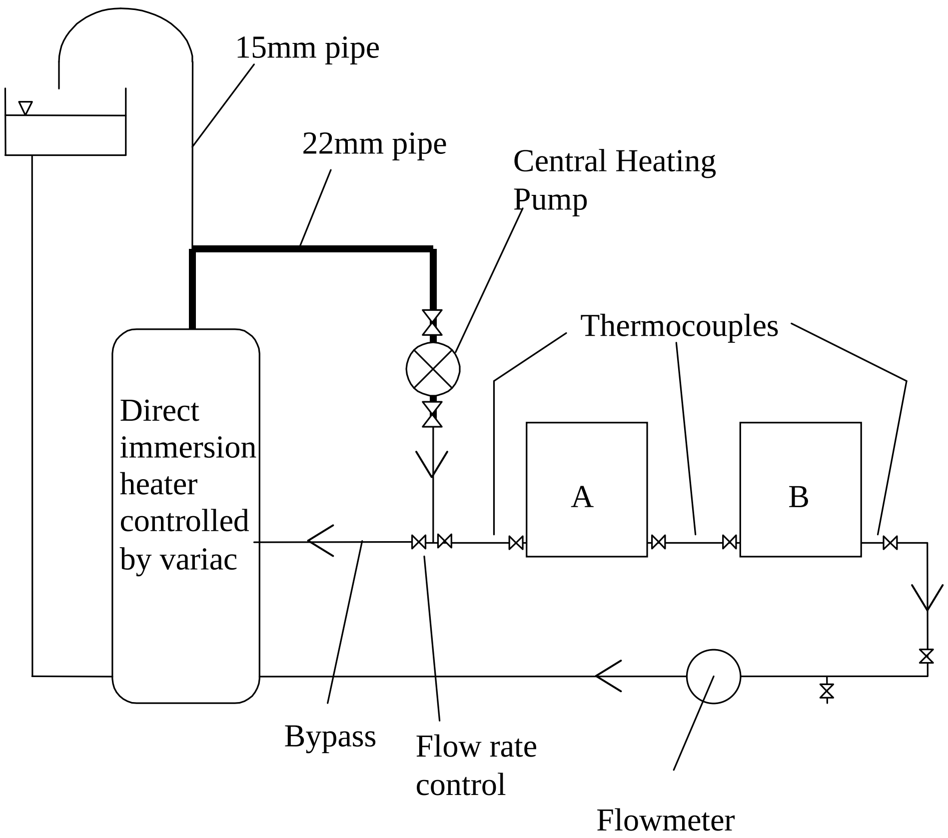
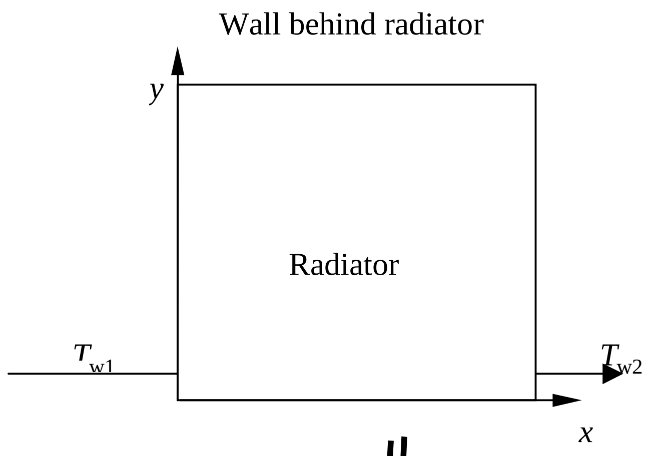
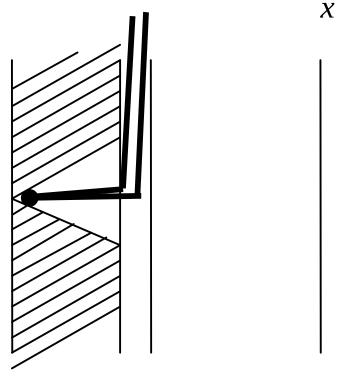


Figure 1 schematic of apparatus



Position of wall thermocouples on walls behind radiators

Thermocouple	x (mm)	y (mm)
1	300	50
2	300	300
3	300	550
4	300	800
5	550	550
6	50	550



Detail of thermocouple placement in wall

Fig 2 Thermocouple placements

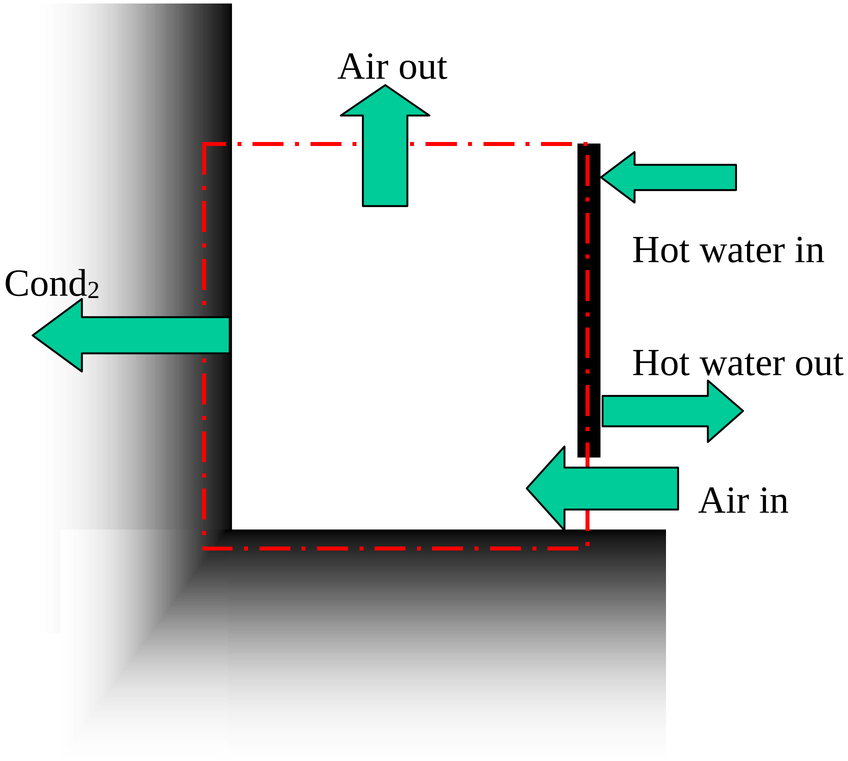


Fig 3 Heat balance for radiator

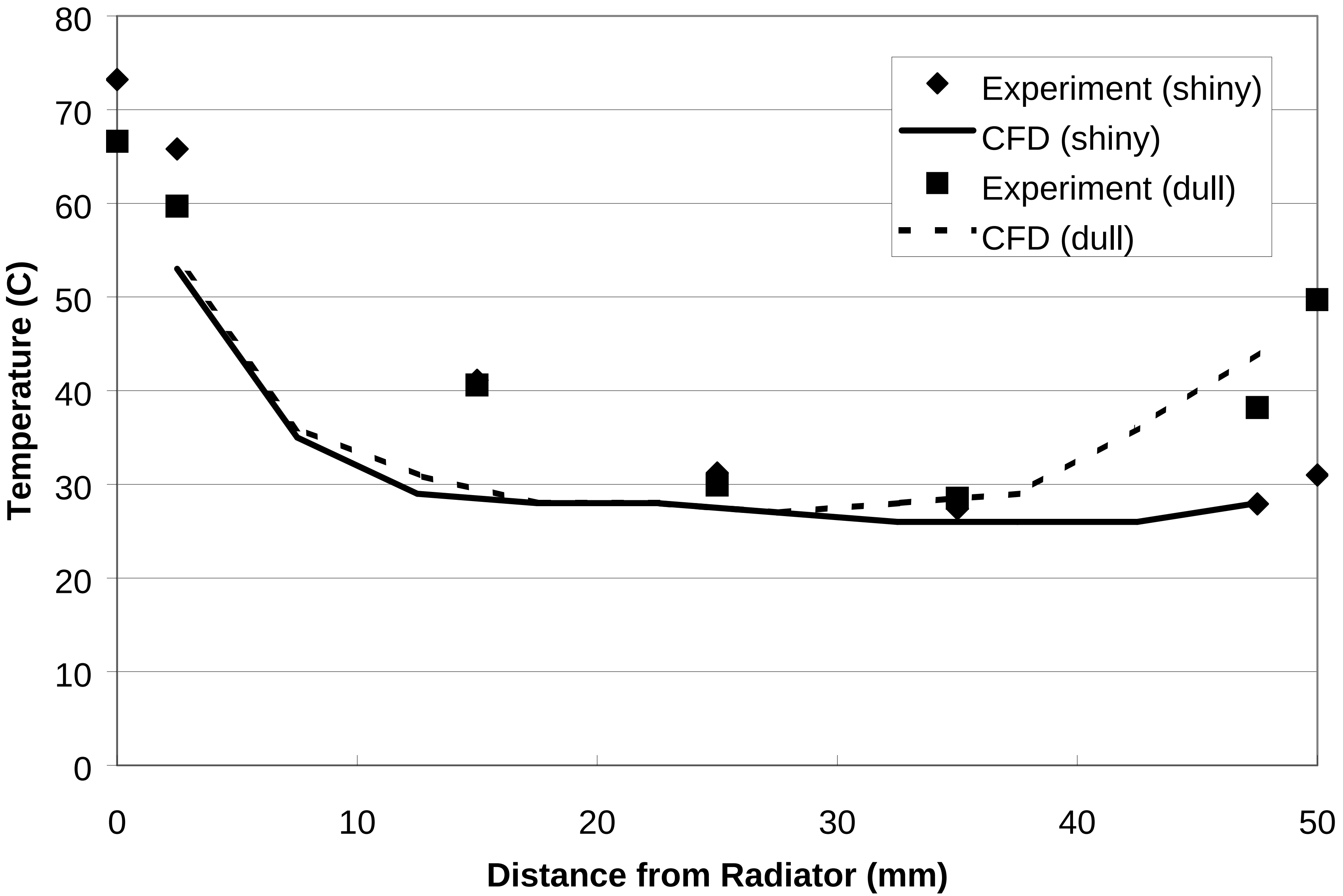


Fig 4 Temperature profiles at top of radiator (wall is at 50 mm)

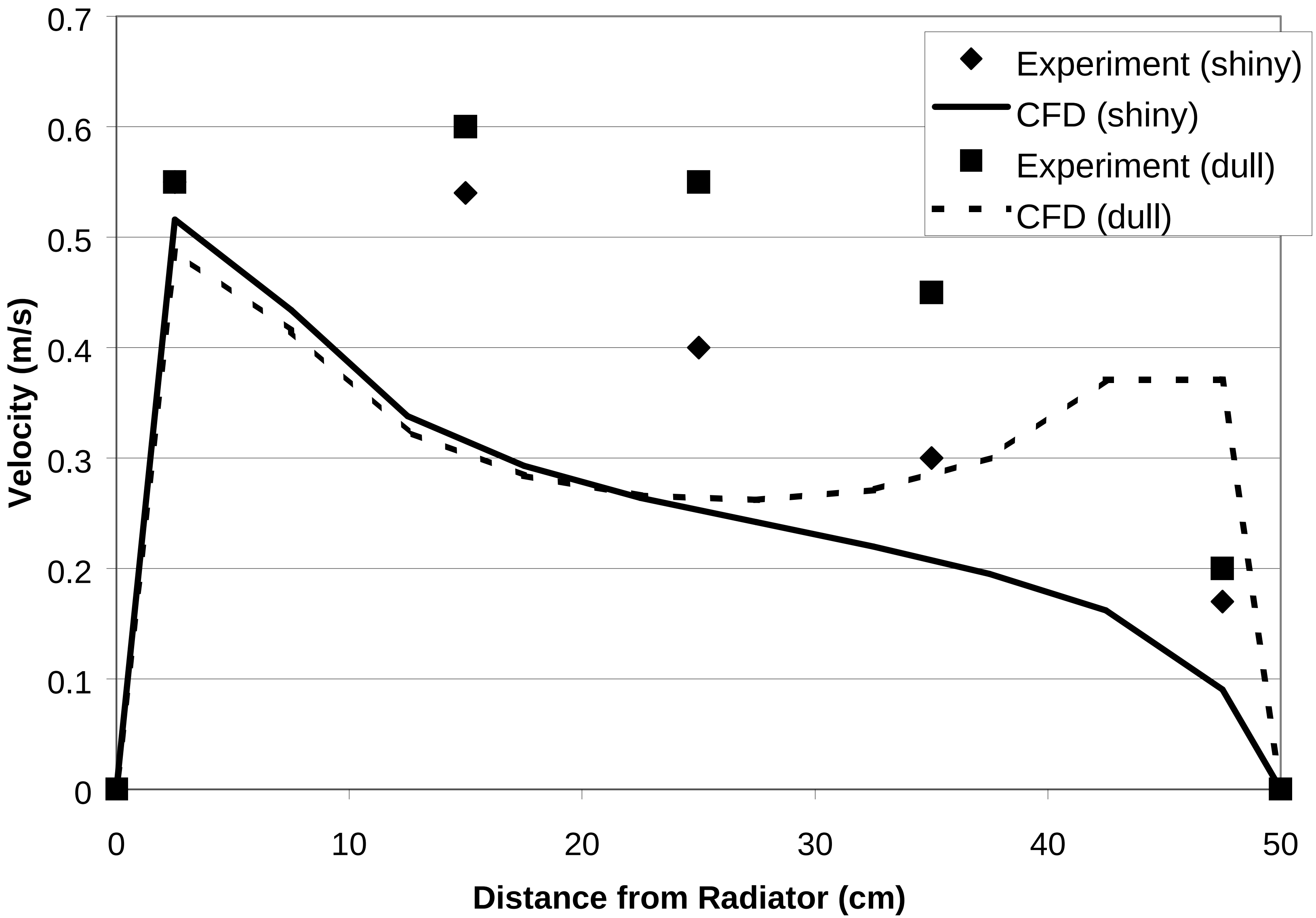


Fig 5 Velocity profiles at top of radiator (wall is at 50 mm)

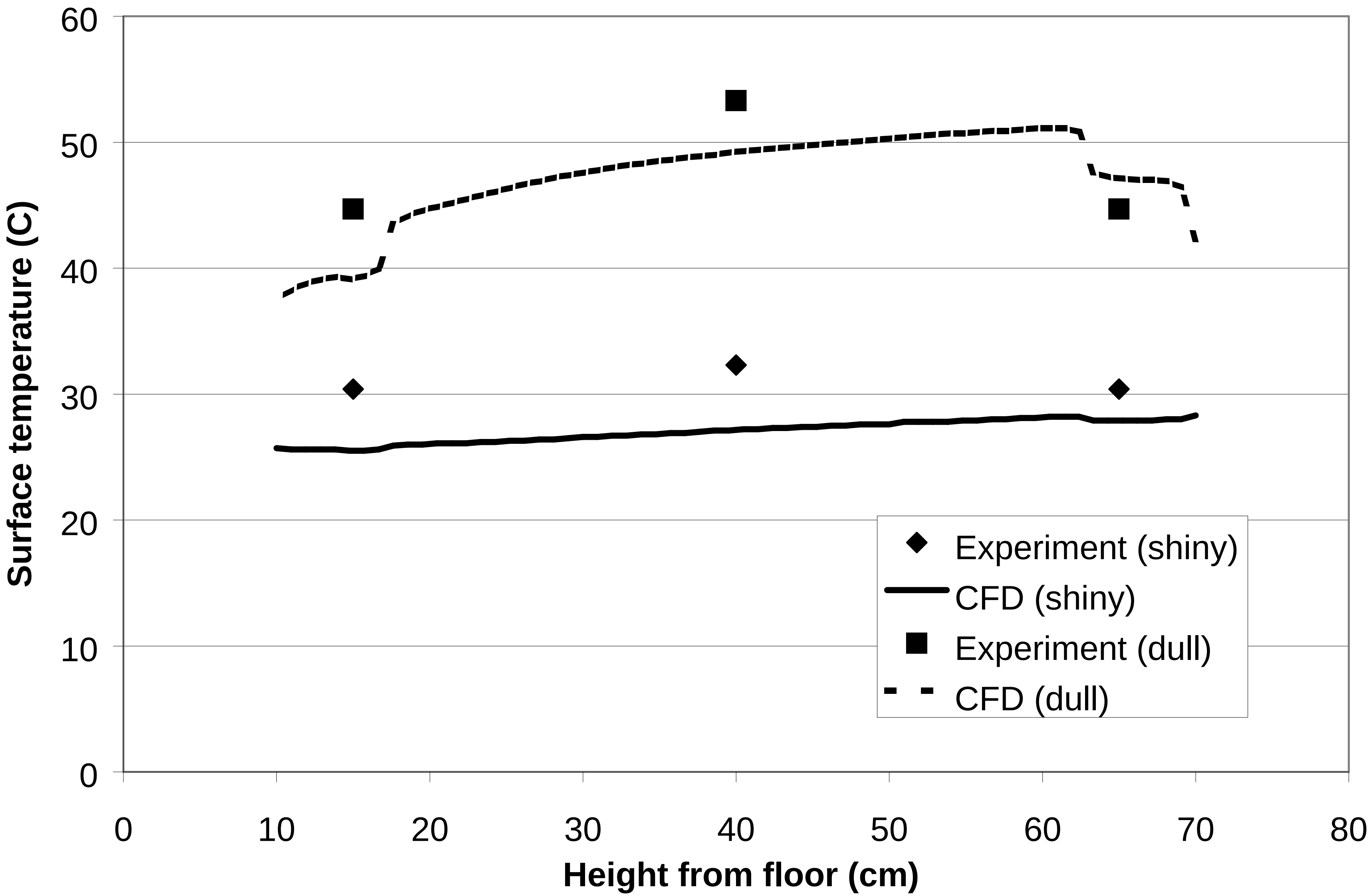


Fig 6 Wall temperature opposite radiator (radiator starts 15 cm above floor)

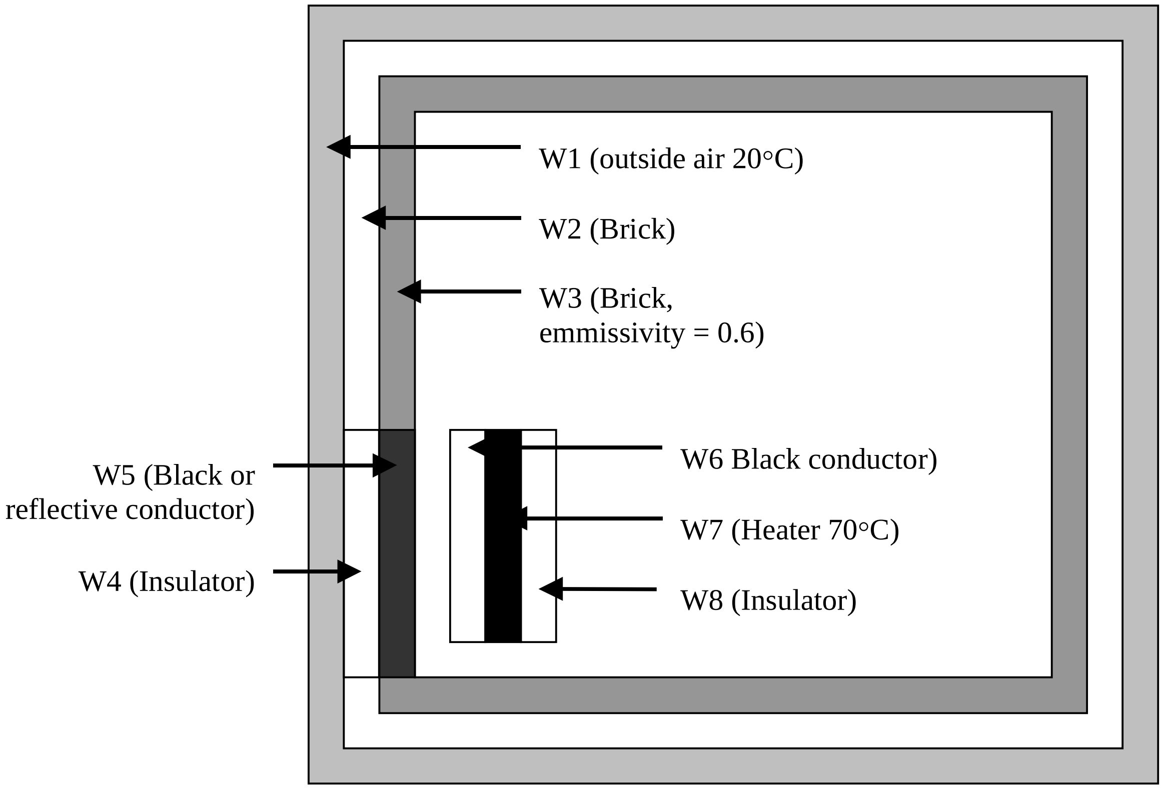


Fig 7 Diagram of CFD regions

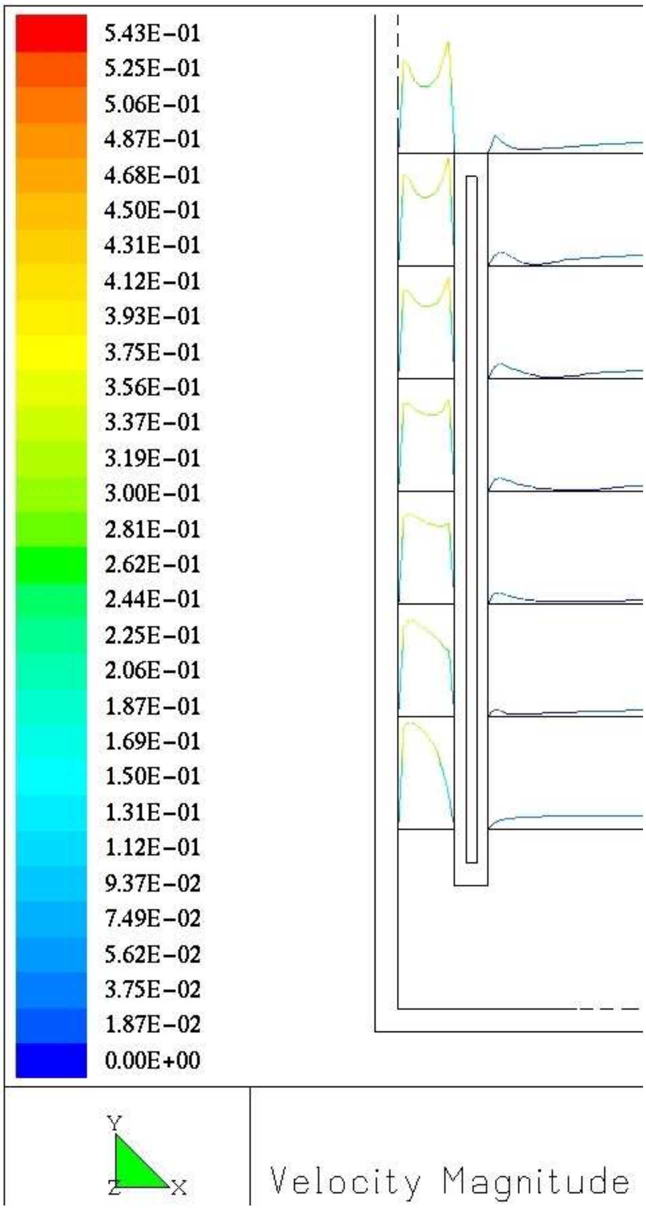
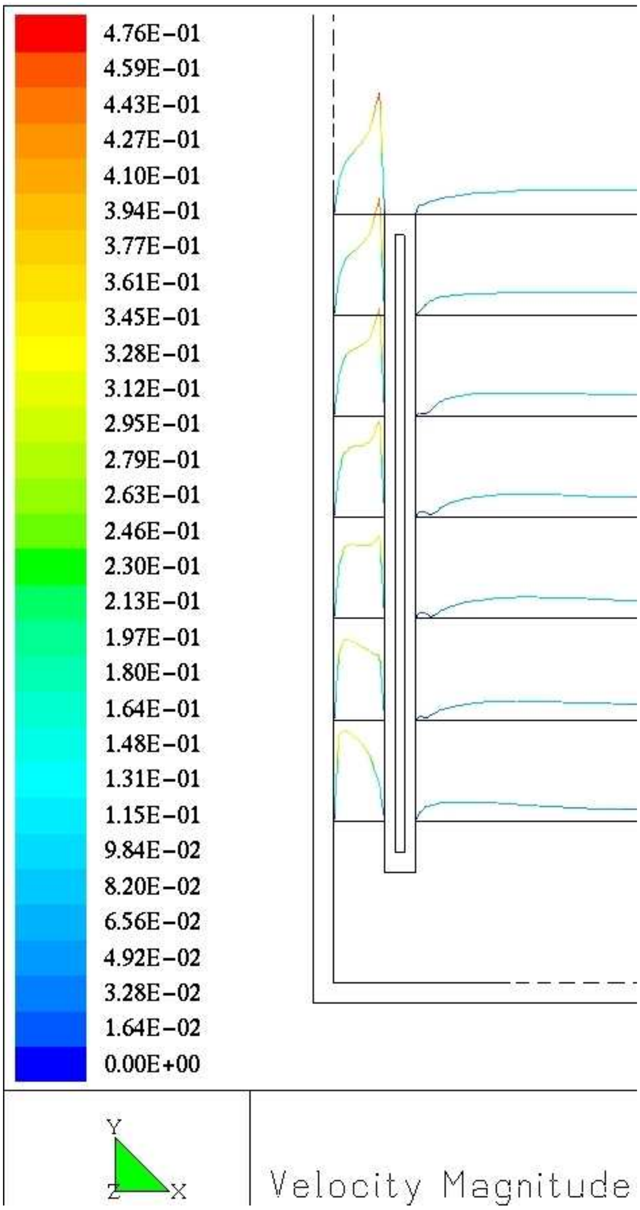
	

Fig 8a v-velocity (m/s) profiles from CFD Analysis (dull surface)	Fig 8b v-velocity (m/s) profiles from CFD Analysis (shiny surface)