University of Southern Queensland

Faculty of Engineering and Surveying

Stress Relieving Furnace Modelling and Analysis

A dissertation submitted by

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Courses ENG 4111 and 4112 Research Project

towards the degree of

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Project Abstract

This project investigates the theory behind the application of a gas fired furnace to the achievement of a certain thermal cycle. The thermal cycle is question is to achieve a stress relieving process on materials generated by processing of the material by (for example) welding.

This is achieved by exploration of available furnace technology information and then by the application of this information to the particular furnace and application under consideration. A thermal model for the furnace is then developed in an attempt to fully understand the effects of system parameter variation on the performance and efficiency of the furnace.

In conjunction with the theoretical model, a prototype furnace is also developed to enable assessment of the furnace parameters directly through measurement. Aspects of the furnace design also considered are the structural, combustion, gas control and electrical elements.

A brief life cycle assessment is also performed in order to assess the environmental impacts of the project. Alternative approaches to performing the stress relieving operation are then considered.

Finally, a summary of the project objectives is presented along with further work which is to be carried out in the future.

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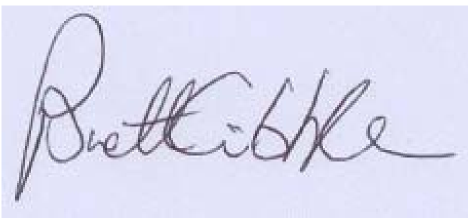

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Signature

__________31st October 2006________

Date

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Nomenclature

ΔT	Change in Temperature
β	Coefficient of Expansion
ε	Emissivity
ϕ	Equivalence Ratio
α	Absorptivity
ν	Kinematic Viscosity
σ	Stefan-Boltzmann Constant, Axial Stress
ρ	Density
τ	Shear Stress

A	Air percentage/ area
A/F_{stoic}	Air/fuel ratio stochiometric
A/F_{actual}	Air/fuel ratio actual
Bi	Biot Number
c	Centroid Distance
C	Celcius
C_3H_8	Propane
c_p	Specific heat
F	Force
Fo	Fourier Number
g	gravity
GJ	Gigajoule
Gr	Grashof Number
H_2O	Water
h_c	Convective heat transfer coefficient
H_g	Specific enthalpy
h_r	Radiative heat transfer coefficient
hr	Hour
k	Thermal Conductivity
kg	Kilogram
KW	Kilowatt
L	Length/ Thickness of Section/ Beam Length
$\dot{m}$	Mass Flow Rate
m	Mass/ Metre
M	Moment
MAT	Material Type
M_{BL}	Mean Beam Length
MJ	Megajoule
MW	Molecular weight
N_2	Nitrogen
n_i	Number of Moles
Nu	Nusselt Number
O_2	Oxygen

P	Total Pressure
p_i	Partial Pressure
Pr	Prandtl Number
P_s	Stoichiometric Product/ Fuel Ratio
q_c	Convective Heat Transfer
q_k	Conductive Heat Transfer
q_r	Radiative Heat Transfer
R	Thermal Resistance
Ra	Rayleigh Number
Re	Reynolds Number
R_s	Air/Fuel Volume Ratio
SF	Safety Factor
S_{sy}	Shear Yield Strength
S_y	Yield Strength
t	Time/ Thickness
T_e	Temperature External
T_g	Gas Temperature
T_i	Temperature Internal
U	Free Stream Velocity
V	Shear Force
V_G	Gas Volume
W	Watt
X	Excess Air Level

Chapter 1: Introduction

1.1 Project Background

Stress relieving is carried out by several large and small national and large international companies operating in Australia. Traditionally , furnaces that have been developed have been built based upon rules of thumb and historical knowledge.

Many of the operators and even builders of furnaces have little knowledge of the processes which occur within and around the furnace. This project seeks to dissect the various components of the furnace operation in able to gain understanding of the importance each plays in the successful operation of the furnace.

The question that arises at the outset of this project is what is stress relieving and why is it necessary?

Thermal stress relief of structures is conducted to remove residual stresses generally caused by welding or heat straightening operations.

According to Crane (1979) , the benefits are:

- Reduction of residual stresses
- Improve resistance to fracture in the heat affected zone of the weld
- Diffusion of hydrogen out of the weld, reducing imbrittlement.

However, Crane (1979) continues on to say that a poor stress relieving cycle can have adverse effects such as only re-distributing residual stresses , not removing them, leading to distortion of the structure.

Stress Relieving is one of the more basic types of heat treatment performed by Australian industry, and is used extensively in the power generation, oil and gas, mining and petrochemical industries on piping systems and small to large structural components. To a lesser extent it is also used to relieve stresses in machined components.

The adequacy and the temperature uniformity of the thermal cycle which are key elements of the furnace environment ensures that the stress relieving cycle is adequate for the full volume of the items being stress relieved.

There are alternative means of stress relieving components and these are:

- Induction heat treatment
- Vibratory stress relief
- Electric/ceramic element stress relief
- Furnace based stress relief

All of these methods have their merits and applications to different materials and quantities of components. The relative costs of applying each of these methods also varies considerably. These other methods are considered briefly in Chapter 7 when alternatives to the furnace method of stress relief are considered. There are certain limitations on the application of the different stress relieving methods mainly due to the size of the items to be stress relieved. The largest items that have to be stress relieved could weight several hundred tons , be hundreds of metres in length and tens of meters in diameter. An example of the type of furnace is shown in Figure 1.1 and is used to stress relief iron ore car dumper assemblies (refer Figure 1.2) which weigh around 70 tons. The fundamental principles behind stress relieving for all the methods is basically the same. With reference to the iron-carbon phase diagram shown below (Figure 1.3), the component is taken to a temperature high enough to impart sufficient energy to the atomic structure to encourage atoms to migrate to positions of lower stress without any accompanying change in phase. If the temperature is too high, the phase changes which is an undesired effect of the process, and wasteful as it is unnecessary to increase the temperature this high. Too low a temperature results in no change in the atomic positioning and is therefore a useless exercise.

Figure 1.1- Large gas fired stress relieving furnace

Figure 1.2- Iron Ore Car Dumper Body after stress relief and coating

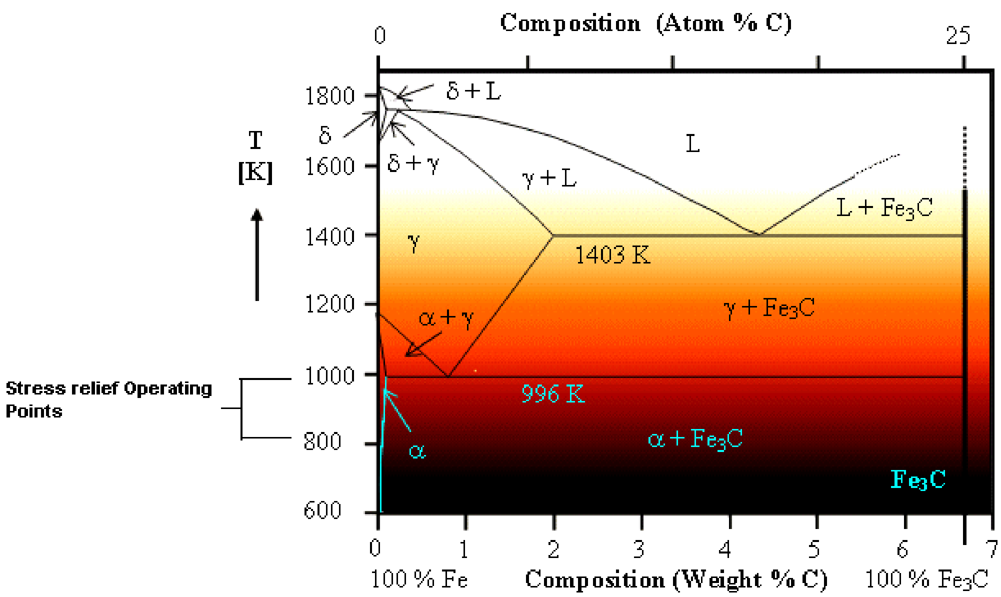


Figure 1.3 Phase diagram for steel with stress relief range

1.2 The Furnace Environment

According to Gilchrist (1963) “ the ideal furnace is a refractory box in which the maximum amount of fuel can be burned as nearly as possible to the theoretical flame temperature and the products of combustion removed with the lowest possible residual heat content”

Further to this the author concludes that the success of a furnace design is measured by:

- The rate at which the furnace can consume fuel
- The efficiency with which it transfers energy to the item being heated
- The longevity of the furnace (i.e how many cycles it can withstand before major refits, etc . are conducted

For the purposes of this project the simple definition for the furnace proposed is thus:

“A device for converting the chemical energy of gas and air into thermal energy in order to produce a controlled thermal cycle for metallic objects”

There are two major processes in a furnace

- Conversion of chemical to thermal energy
- Transfer of thermal energy to the load

Heat transfer can be achieved by the following means and must be allowed for in the circuit model. The modes of heat transmission that exist are:

Conduction – occurring in solids by intermolecular interaction

Convection- occurring in liquids and gases

Radiation- occurring in all bodies

According to Boyer (1984) all furnaces operating above 595°C are heated primarily by radiation. It will be assumed therefore that when the thermal circuit model is derived, this will be the dominating heat transfer mechanism at the **soak temperature.**

According to (ed. Baukal (2001)) a detailed analysis of furnace heat transfer is complex and many furnace vendors although having proprietary methods for computing the heat transferred to the process load, these are based upon empirical data and not published in any literature. In addition, furnace heat transfer seems to be the source of controversy and disagreement among engineering professionals in the field (much disagreement exists as to the relative importance of gas radiation and surface radiation for example). Khalil (1982) quotes the following:

"In turbulent combustion, the interaction of fluid mechanics, chemical
kinetics and heat transfer continues to be both a challenging and a
complicated area of research and application"

In addition , Khalil (1992) goes on to say that mathematical modelling procedures have been used as an alternative to direct experimentation for the last fifty years. Note that computer programs for the determination of these parameters have advanced somewhat since the author wrote this.

1.3 Project Aims

To design and analyse the operation of a gas fired stress relieving furnace using traditional thermal design and computer assisted techniques and the effects of the system variables on the performance of the furnace.

The project will use a design adopted by the stress relieving industry in Australia using traditional/historical rule of thumb techniques and evaluate the effectiveness of this design and the suitability to the role it performs.

In order to achieve these aims above the following issues require addressing:

- The furnace is a structure and as such requires structural integrity and adherence to structural design specifications. In addition all aspects of the furnace must comply to relevant standards
- The furnace is an energy transfer device with several energy transformations, the primary one being the combustion of gas to provide high furnace temperatures. The combustion system required analysis
- The furnace requires design of the control system, in order to allow the necessary thermal cycle(s) to be achieved.
- The furnace is a complicated thermal system and requires analysis of all the related inputs and outputs to establish a thermal circuit model which can be adopted to varying load and input energy scenarios

- Once the basic model is evaluated, to review this design and look at alternative improved design possibilities.

Chapter 2: The Furnace Model and Project Methodology

2.1 Design of Furnace Model

According to Ertas and Jones (1996) , engineering design if the process of devising a system, component or process to meet desired needs.

With respect to this project, the system has already been devised by our forefathers. As indicated previously , the development of these systems relied upon intuitive rather than analytical means. This analysis will therefore seek to improve the future design process by providing a more robust means of determining the variables associated with the furnace operation.

2.2 Furnace concept and structure

A basic diagram of the furnace design is shown in Appendix B.

The furnace consists of the following elements, which have to be dealt with respectively.

- Structural Component
- Combustion Component

- Control System Component
- Thermal Circuit / Heat Transfer Components

and within these elements are sub elements as shown in Figure 2.1.

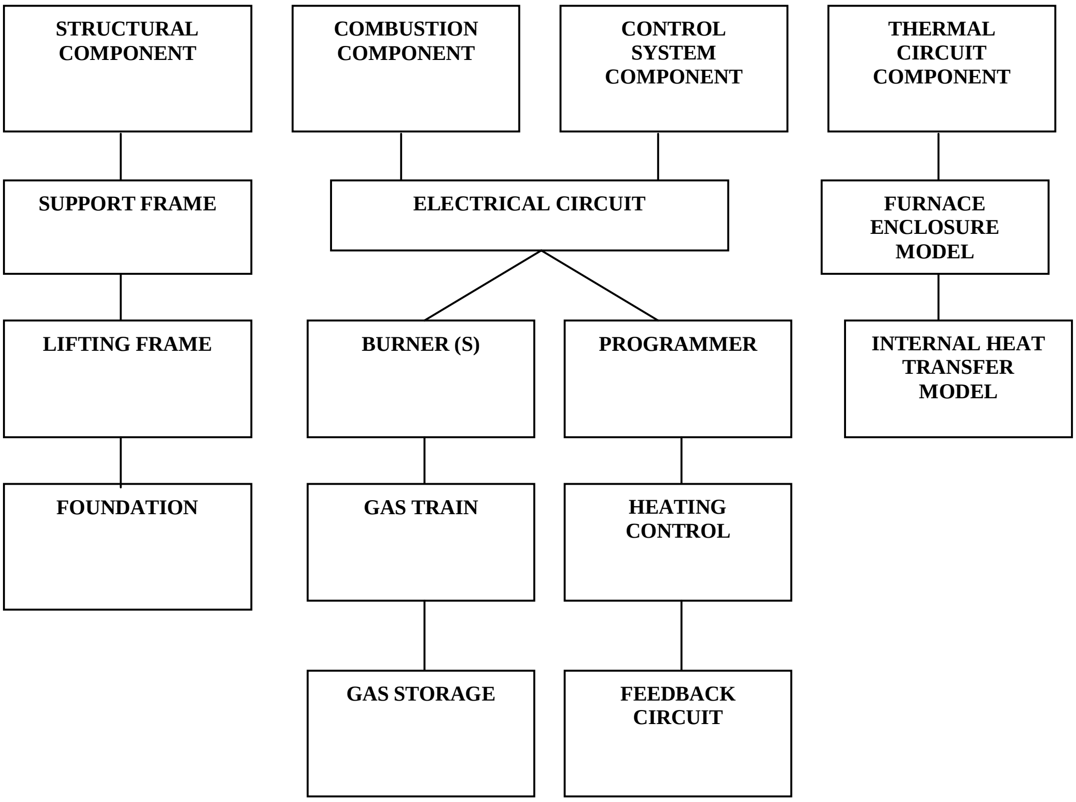


Figure 2.1- Furnace sub-elements

In addition to the core elements listed, other issues have to be taken into account, and these are:

- Consequential effects –ethical, environmental
- Risk assessment

These two issues will be dealt with in Chapter 6

2.3 Structural/Construction Component

The structural component, whilst important in its own right, is considered peripheral to the main analysis and is therefore dealt with in Chapter 5.

2.4 Combustion Component

The combustion element of the furnace is an essential part of the thermal circuit model and will be discussed in detail in Chapter 3.

2.5 Control System

The basic control system for the furnace must allow for the following cycle to be achieved with variability in the following (Refer to Figure 2.2): Explanations of terms: **Rise time-** the rate at which the temperatures ramps up from the **initial set temperature** to the **soak temperature -** the temperature at which the metallic object is held for a set **soak time** , **fall time** is the rate at which the temperature is reduced back to the **turn off temperature**

The control system for the furnace has been developed by Accutherm Pty Ltd in accordance with the Eclipse Burners Manufacturers instructions and from data supplied in accordance with Figure 2.1. The control system schematic is shown in Appendix C. A description of the control system is presented in Chapter 5.

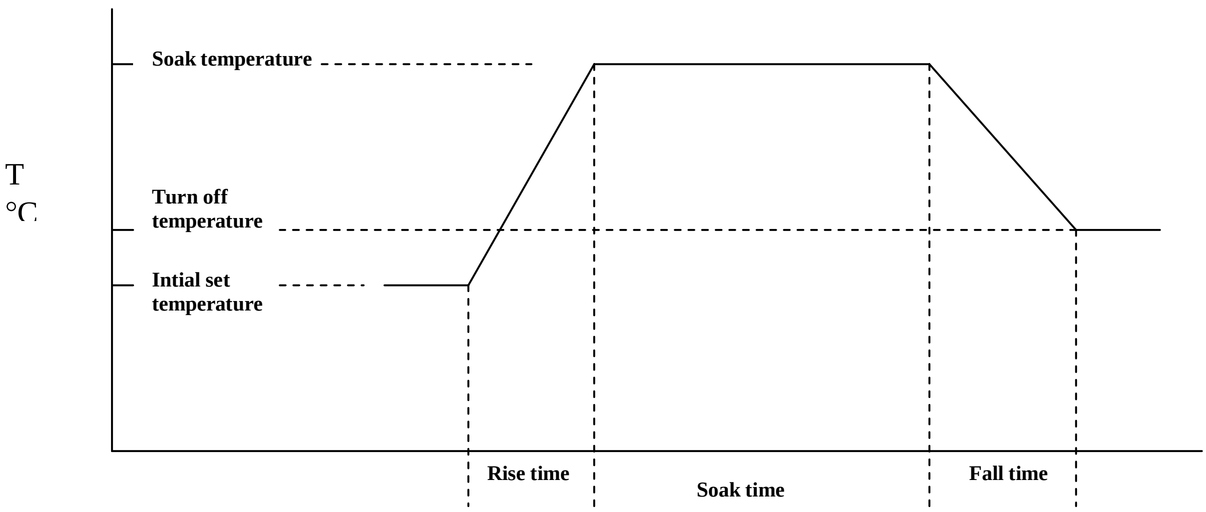


Figure 2.2- Thermal Cycle Characteristics for Furnace Load

The programmer is set to the required temperature and ramp settings and essentially the burners are fired with variable amounts of excess air until thermocouples on the furnace load indicate that they have reached the prescribed temperature levels. The control system will then adjust the heat input to the furnace to achieve the programmed cycle.

The temperatures and rates to be achieved by the furnace are as listed in Table 2.1 These temperatures are expected to cover the full range of stress relieving cycles the furnace may be required to operate.

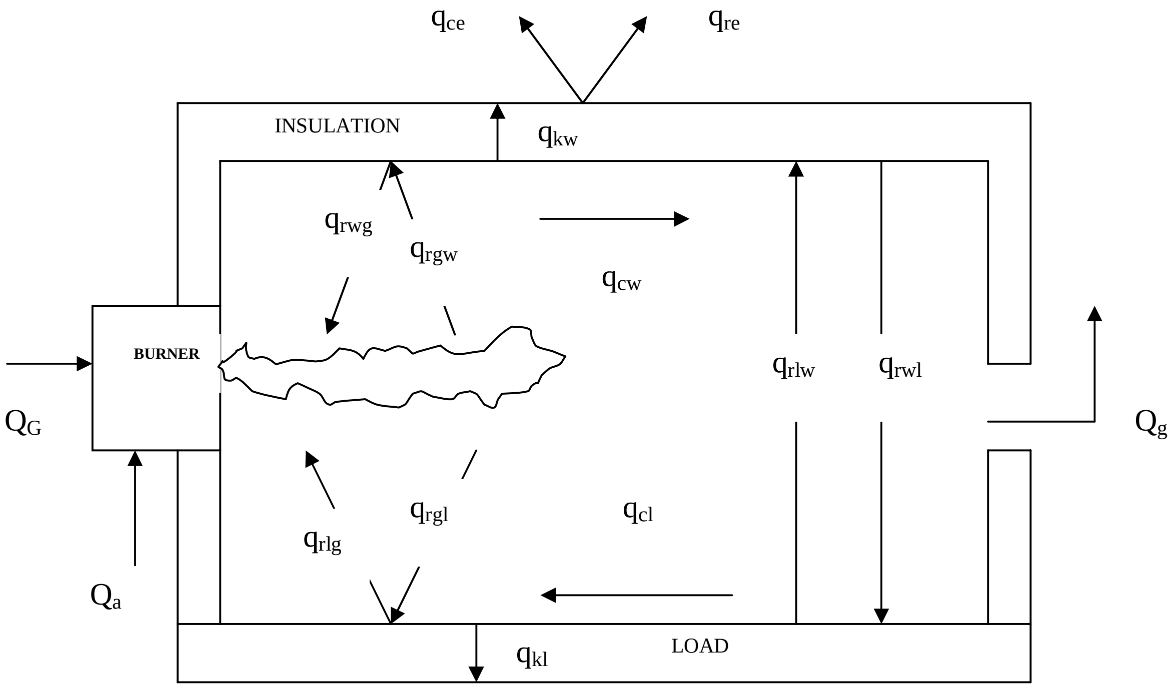
VARIABLE	MINIMUM	MAXIMUM
INITIAL TEMPERATURE (º C)	200	300
RISE TIME (º C/hr)	50	200
SOAK TEMPERATURE (º C)	500	720
SOAK TIME (hr)	1	6
TURN OFF TEMPERATURE (º C)	200	300

Table 2.1- Thermal Cycle Temperature and rate requirements

2.6 Thermal Circuit / Heat Transfer Components

The thermal circuit is essentially as shown in Figure 2.3. (From Rhine and Tucker 1990). A detailed theoretical analysis of the components is presented in Chapter 3.

The furnace design varies slightly from the Rhine and Tucker model in that the base is constructed of a sand and bitumen insulation barrier directly onto sand/earth ground material. The single gas model is the only one considered in this project as the complexity of other models, which predict spatial distribution of temperature inside the furnace would have taken considerably more time. The model is considered adequate for this particular application as the high velocity burners "stirring" of the gases should ensure reasonable temperature uniformity throughout the furnace.



Q_a- Heat content combustion air
Q_G- Heat content combustion gas
Q_g- Heat content flue/exhaust gases

q_{re}- Radiation heat loss/gain external
q_{ce}- Convective heat loss/gain external
q_{kw}- Conductive heat transfer wall
q_{kl}- Conductive heat transfer load
q_{rwg}- Radiation heat transfer wall to gas
q_{rgw}- Radiation heat transfer gas to wall
q_{rgl}- Radiation heat transfer gas to load
q_{rlg}- Radiation heat transfer load to gas
q_{cw}- Convective heat transfer gas to wall
q_{cl}- Convective heat transfer gas to load
q_{rlw}- Radiation heat transfer load to wall
q_{rwl}- Radiation heat transfer wall to load

Figure 2.3- Single Gas Model for Furnace

2.7 Project Methodology

Wherever appropriate a literature review relating to the particular system element is conducted and presented along with the presented material. For each element of the system discussed, a theoretical derivation of its value and wherever possible a physically derived value will be obtained and incorporated in the overall furnace model in accordance with Figure 2.4.

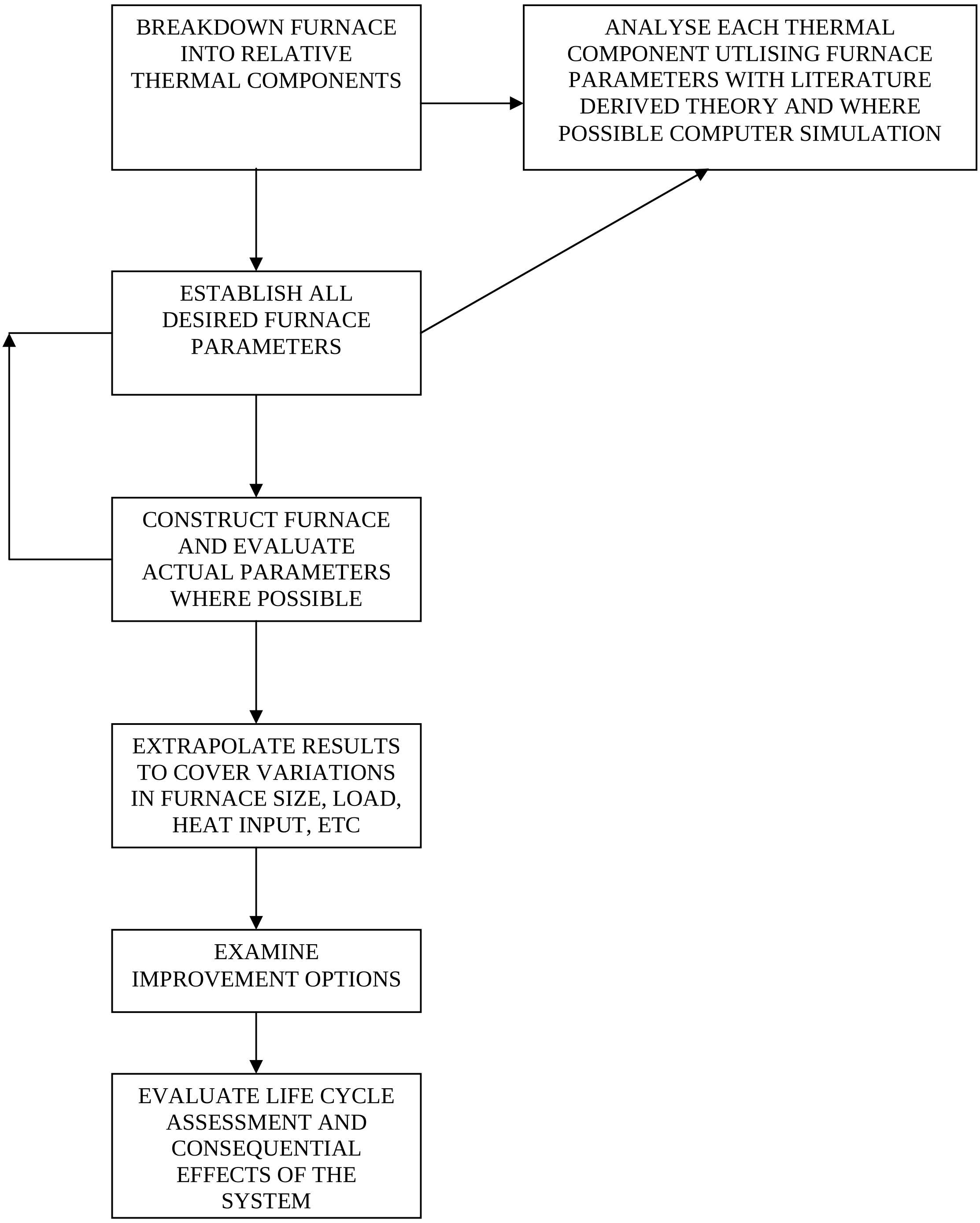


Figure 2.4 – Project Methodology Flowchart

2.7.1 Literature Review Summary

As mentioned previously, the literature review was conducted concurrently with the preparation of the particular sections of the project. The majority of the review compared the approaches taken by the various authors in solving particular heat transfer problems. Most of the texts agreed with each other however there were some differences in opinion when it came to the relative importance of a particular heat transfer mechanism in a particular situation. Some authors stated a specific instance as being factual, without much supporting evidence, whilst the other authors delat with the same subject with caution, explaining the short falls in assessing the specific phenomena.

Also of note was the absence of recent texts relating to furnace technology. Whilst the internet had a large amount of literature specifically on energy saving features of new furnace applications and new refractory material types, there were only a couple of texts written in the last ten years with more up to date information cited. The underlying message from the literature review was the extreme difficulty in isolating a particular heat transfer mechanism for evaluation when one or more other mechanisms are present. This is especially relevant in the case of gaseous heat transfer issues , where internal processes within the gas itself have to be considered in order to understand the external behaviour of the gas.

Chapter 3: Thermal Circuit Modelling

As discussed in Chapter 2, the primary components of this system are the combustion system and the thermal housing of the furnace. Evaluation of these two elements will be presented in this chapter. Referred values will be taken from the tables shown in Appendix D and any derived values from calculations in the text.

3.1 Overall Heat Balance

As this system has to obey the laws of energy conservation, it is prudent that the heat/energy balance equation for the furnace is derived first. For the furnace in question the energy balance can be written as follows (from Rhine and Tucker,1990):

$$\dot{Q}_{G,net} + \dot{Q}_a = \dot{Q}_{g,net} + \dot{Q}_L + \dot{Q}_{losses}$$

Where essentially the sum of the input energy (from the gas and air) equals the flue gas losses , increase in load energy and all other losses such as convection, conduction and radiation loss from the walls and floor.

From Serway (1990) , the energy requirement to effect the temperature change required in the load will be:

$$Q_L = mc_p \Delta T$$

This would be the ideal quantity of energy if there was some means of direct energy transfer to the load. For 3000kg load, going from 20 to 720 ºC the energy requirement will be around 1.04 GJ.

3.2 Energy Input and Losses

There are two components to the input energy side of the model and these are the gas input and the air input. To achieve the thermal turndown or heat control using the burners considered the burners operate at up to 30% excess air. This means that entrained in the combustion products are a high level of gases not involved directly in combustion which reduce the effective heat input into the furnace. The reason the burners are fired in this way, is that unless there is a more elaborate (and therefore more expensive) firing control mechanism, the burners can only be either off or on , and when they are on, if there is no other means to control the heat input, the desired thermal cycle will be impossible to control and will be exceeded quite quickly.

It is extremely important that the cycle is well controlled, because if there are large temperature differential between the internal and external surfaces of the load, distortion may occur which may not be able to be corrected without expensive rework.

3.2.1 The Combustion Process and System

According to Turns (2000) , an excess air ratio of 30% gives an equivalence ratio ϕ in accordance with the following:

$$\% \text{ excess air} = \frac{1-\phi}{\phi}.100$$

therefore giving an equivalence ratio of 0.77.

For a hydrocarbon fuel such as the one used in this project, the stoichiometric relationship for combustion is given as follows (from MEC4104 Energy Technology-USQ, 2005)

$$C_x\,H_y + a\,(O_2 + 3.76N_2) \Rightarrow x\ (CO_2 + (\text{y/2})\,H_2\,O + 3.76\,a\ N_2)$$

$$\text{where} \qquad a = \frac{x+y}{4}$$

so for propane the stoichiometric relationship for combustion is:

$$C_3\,H_8 + 5(O_2 + 3.76N_2) \Rightarrow 3CO_2 + 4\,H_2\,O + 18.8\,N_2$$

The stoichiometric air/fuel ratio is given by the following:

$$4.76\,a\,\frac{MW_{air}}{MW_{fuel}}$$

where MW_{air} = 28.85 and MW_{fuel} = 44.096

the stoichiometric air/fuel ratio A/F_{stoic} is therefore 15.57

The actual air/fuel ratio A/F_{actual} is given by

$$\frac{A/F_{stoic}}{\phi}$$

which gives 20.22

The mass flow rate of air required compared to fuel will be

$$\dot{m}_{fuel} = \frac{\dot{m}_{air}}{A/F_{actual}}$$

Therefore with the proposed burner system, 20.22 times the mass of fuel is required from the air supply system.

Each burner supplies 117kW of power to the system (Eclipse burner data- see Appendix C). Propane has an energy value of 49.93 MJ/kg. For the system to develop this power the burner has to consume 0.422 GJ per hour or 8.45kg . For four burners, the total mass requirement for propane will be 33.8kg/hr. The air supply system will then be required to supply 683.4kg of air per hour to meet the excess air requirement.

The fan proposed by the equipment suppliers (Refer Appendix C) is capable of supplying around 800m^3/hr of air at 5kPa. The density of air is 1.2 kg/m^3 at 20º C (Douglas, Gasiorek, Swaffield 1993). Therefore the fan can supply around 960kg of air to the burners per hour.

3.2.2 Effective Heat Input to Furnace

The apparent power input to the furnace will be 117kW x 4 (Four burners)

Reed (1991) claims that in combustion , approximately 33% of the heat produced in combustion is contained in the binary gases (CO_2 and H_2O) and 63% in the diatomic gases (O_2 and N_2) which are non-radiating. This is the case for a stoichiometric combustion, in the case of excess air, the diatomic gases make up a larger percentage of the gas volume and therefore there are less atoms giving up heat due to radiation.

Therefore radiation transfer of heat to the air inside the furnace and also to the walls, floor and load may be less significant than the only other transfer mode, which is convection.

The effect of using excess air on the theoretical flame temperature is also of interest as the higher the initial temperature of the products of combustion, the more effective will be the heat transfer mechanisms within the furnace.

From Davies (1970) and Gilchrist (1977) the use of excess air causes a marks reduction in the theoretical flame temperature, and therefore also of the actual flame temperature.

3.2.3 Conductive heat losses

There are a number of heat loss mechanisms to account for in the furnace and they are:

- Conductive heat loss through walls and floor
- Convective heat loss from external surfaces
- Radiative heat loss from external surfaces
- Flue gas losses

The conductive heat losses will be greatest when the temperature differential between the outer and inner surfaces are the greatest- that is , when the furnace is at the soak temperature. The opposite applies to the flue losses, in that they will be at a minimum when the soak temperature is reached, as the energy input will be dramatically cut back and the flue almost closed.

The walls and roof are fabricated from the same materials and therefore can be dealt with together. The thermal circuits for the wall can be described as shown below. There are two cases considered, the wall with only insulation and steel layers considered and also the case of considering the insulation restraining pins.

These two different models are shown in Figures 3.1 and 3.2.

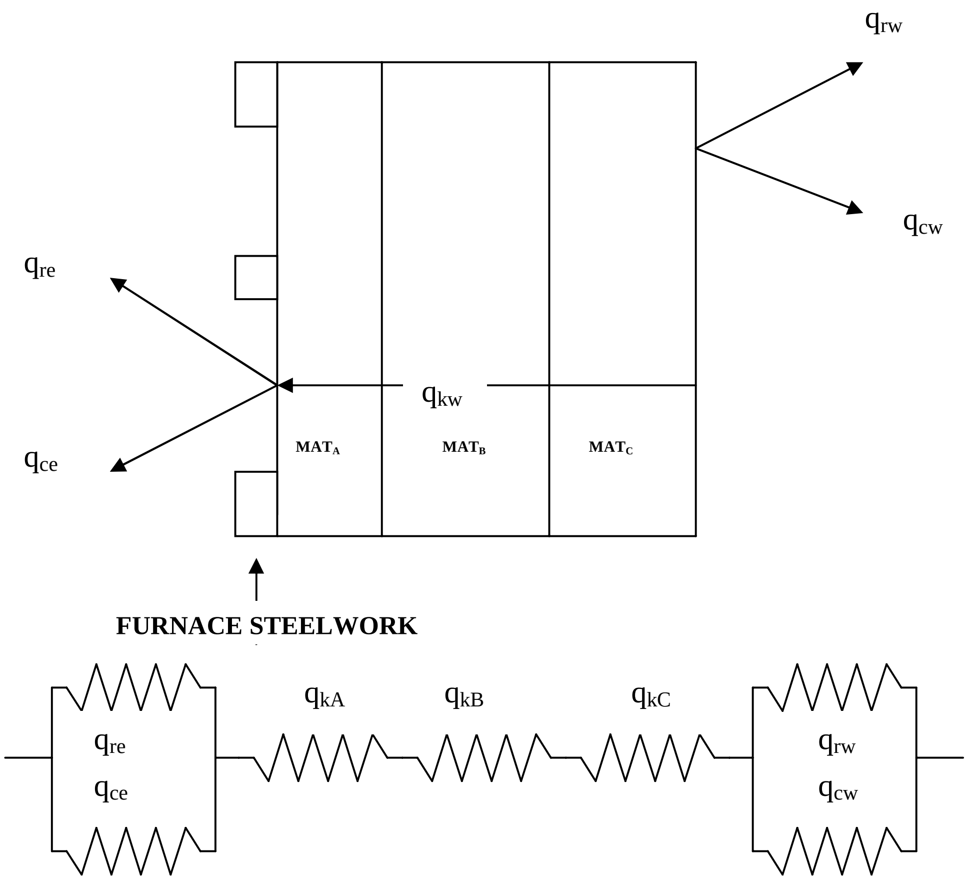


Figure 3.1 – Total furnace wall/roof model with electrical analogy

At a steady state internal temperature of 720º C , and an external temperature of $T_{e.}$ the conductive heat losses through the wall can be calculated as follows: (For now, the external and internal heat transfer modes will not be considered).

$$\frac{q_{kw}}{A} = \frac{T_i - T_e}{L_A / k_A + L_B / k_B + Lc / k_C}$$

which gives the following:

$$\frac{q_{kw}}{A} = \frac{T_i - T_e}{0.003/43 + 0.075/0.18 + 0.025/0.18}$$

therefore the steady state conductive heat loss q_{kw} through the walls is:

$$1.746\ (T_i - T_e)\ \mathrm{W/m^2}$$

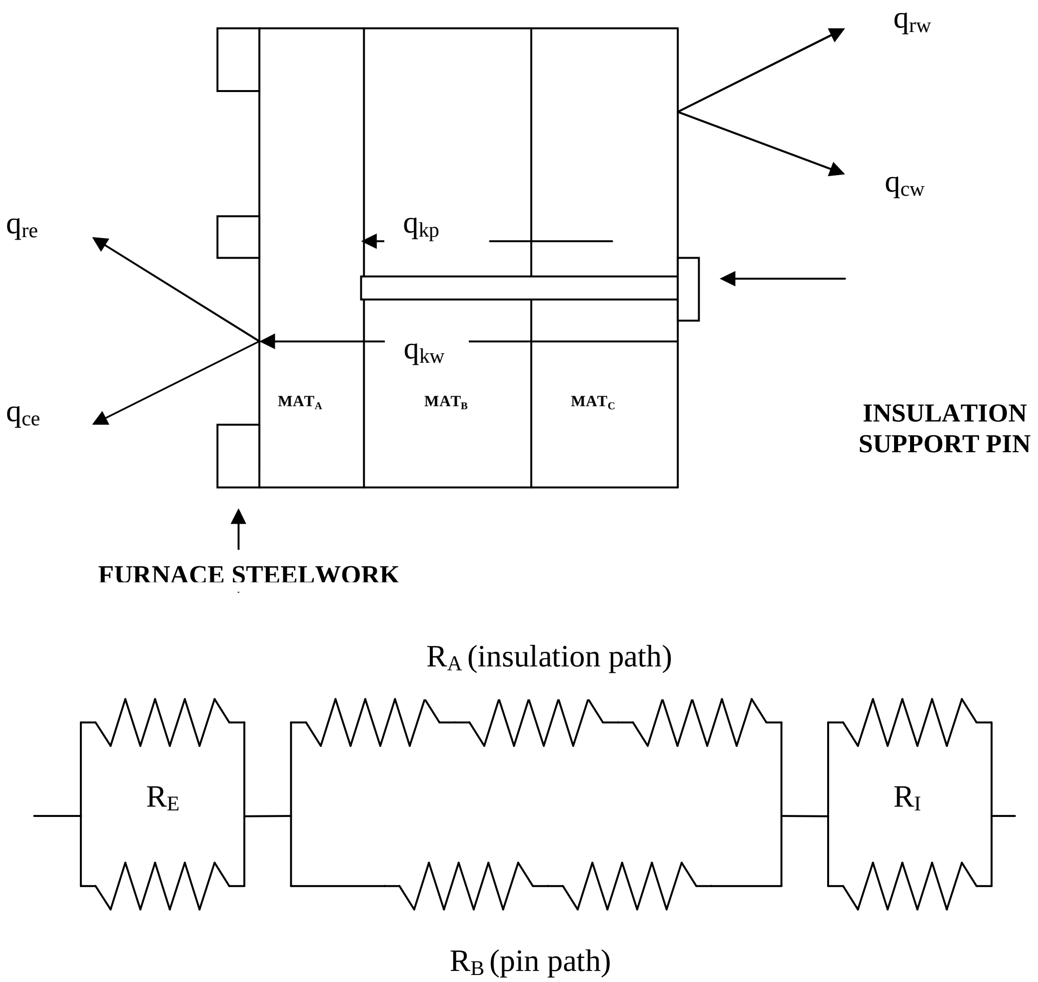


Figure 3.2 – Total furnace wall/roof model with electrical analogy and pins included

For the conductive part of the circuit shown, the following applies:

$$R_A = \frac{L_A}{k_A A} + \frac{L_B}{k_B A} + \frac{L_C}{k_C A}$$

and

$$R_B = \frac{L_P}{k_P A_P} + \frac{L_A}{k_P A_P}$$

also

$$R_T = \frac{R_A R_B}{R_A + R_B}$$

therefore

$$R_A = \frac{0.003}{43 * 0.999352} + \frac{0.075}{0.18 * 0.999352} + \frac{0.025}{0.18 * 0.999352} \quad = \quad 0.5733$$

$$R_B = \frac{0.003}{43 * 0.000648} + \frac{0.150}{14.4 * 0.000648} \quad = \quad 16.18$$

$$R_T = \frac{0.5733 * 16.18}{0.5733 + 16.18} \ = \ 0.554$$

The conductive heat transfer through the furnace wall with the pin included is therefore

$$q_{kwp} = \frac{T_i - T_e}{0.554}$$

$$q_{kwp} = 1.8(T_i - T_{e)} \text{ W/m}^2$$

For values of T_i= 720º C and T_e=15º C , q_{kwp}=1269 W/m^2

Therefore the pin model increases the conductive heat loss by around 3%. The pins only represent 0.06% of the surface area.

The floor heat transfer model can be described as shown in Figure 3.3.

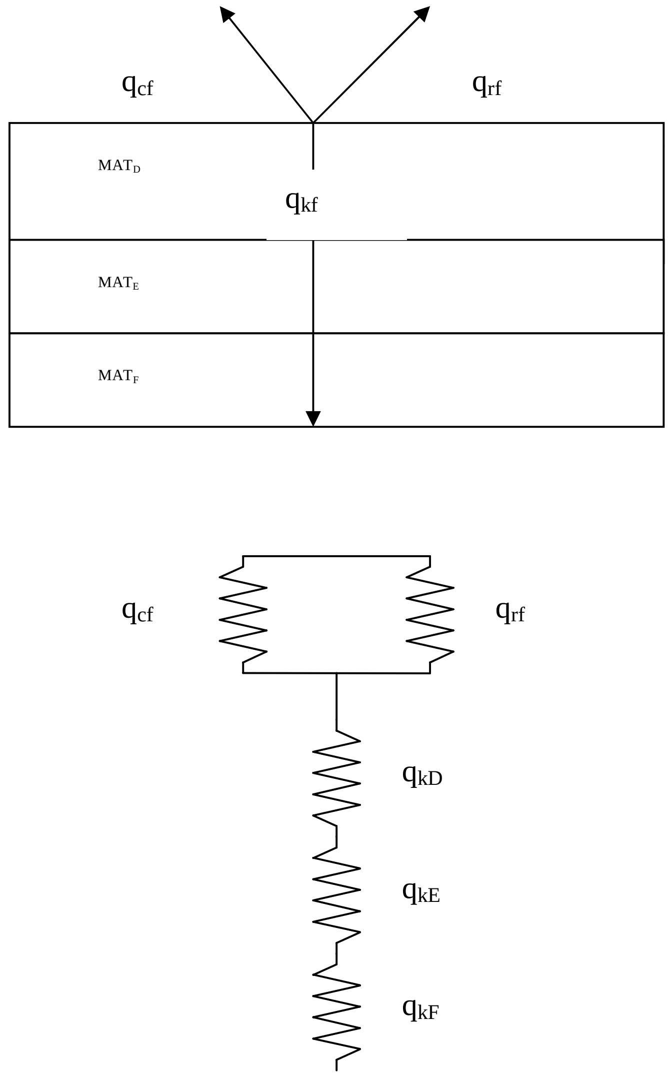


Figure 3.3– Total furnace floor model with electrical analogy

Again, the conduction component of the model will be assessed only. Note that material F (MAT_F) is the earth and has obviously infinite thickness. Therefore an interface temperature between MAT_E and MAT_F will be chosen to allow estimation of heat conduction through the floor. This is based upon a average earth temperature of 15ºC . The conductive heat loss from the floor will be as follows:

$$\frac{q_{kf}}{A} = \frac{T_i - T_f}{L_D / k_D + L_E / k_E}$$

therefore

$$q_{kf} = \frac{720 - 15}{0.15 / 0.35 + 0.02 / 0.17} = 1290.7 \text{ W/m}^2$$

3.2.4 Convective heat losses

There are two conditions which possibly exist for the furnace in terms of convective heat loss and they are either natural or forced convection. Either is possible as the furnace is based outside. Natural convection will be the least likely of the two as the air flow outside is rarely still. It will be evaluated here to compare with the forced convection values.

In the case of natural convection, there are four vertical surfaces and one horizontal. Letting the ambient temperature be 20º C (an average figure) and using the formulas found in Keith and Bohn (2001) , the Grashof number is:

$$Gr_L = \frac{g\beta(T_e - 20)L^3}{\nu^2}$$

If T_e = 100º C the film temperature is 60º C and at this temperature:

$$\frac{g\beta}{\nu^2} = 0.782$$

Therefore $Gr_L = 0.782 \times 80 \times 2.395^3 = 859.43$

Flow is therefore laminar. At 60º C the Pr number is 0.71 and k = 0.0279

$Gr_L Pr$ is therefore 610.19

The average Nusselt number (from Kreith and Bohn 2001 Fig 5.5)

$$\overline{Nu}_L = 0.555(Gr_L \Pr)^{1/4}$$

which equals $0.555(610.19)^{1/4} = 2.758$

the convection heat transfer coefficient is given by

$$\bar{h}_c = \overline{Nu}_L \frac{k}{L}$$

which equals $2.758 \times 0.0279/2.395 = 0.032 \text{W/m}^2\text{K}$

At T_e of 30º C, the film temperature is 25º C and at this temperature:

$$\frac{g\beta}{\nu^2} = 1.272$$

$$k = 0.2545$$

$$\Pr = 0.71$$

Gr_L is therefore $1.272 \times 10 \times 2.395^3 = 174.74$

$Gr_L Pr = 124.1$

$Nu_L = 0.555(124.1)^{1/4} = 1.85$

$$\bar{h}_c = \overline{Nu}_L \frac{k}{L} = 1.85 \times \frac{0.2545}{2.395} = 0.0196 \text{W/m}^2\text{K}$$

These values therefore give a maximum and minimum convective heat transfer coefficient from the vertical surfaces of 0.0196 to 0.032 W/m^{2}K.

For the first case of $\bar{h}_c$ = 0.032 and T_e =100º C

$$\frac{q_c}{A} = \bar{h}_c (T_e - T_\infty)$$

which equals $0.032(100 - 20)$

$= 2.56$ W/m^2

for the second case, $\bar{h}_c$ = 0.0196 and T_e =30º C

$$\frac{q_c}{A} = 0.0196(30 - 20)$$

$= 0.196$ W/m^2

which is around 7.6% of the first case, showing that the external temperature difference plays an important role in natural convection.

In the case of the roof, one square metre represents a characteristic length L of

$$\frac{L^2}{4L} = 0.25\text{m}$$

The Rayleigh number for the case of T_e =100º C (Mean t = 60º C) is as follows:-

$$Ra_L = \frac{g\beta\Delta T L^3}{\nu^2}\Pr$$

therefore $$Ra_L = \frac{9.8 \times 353^{-1} \times 80 \times 0.25^3 \times 0.71}{\left(1.94 \times 10^{-5}\right)^2}$$

$$= 6.5 \text{ x } 10^7$$

For Te = 30º C (Mean t = 25º C)

$$Ra_L = \frac{9.8 \times 298^{-1} \times 10 \times 0.25^3 \times 0.71}{\left(1.61 \times 10^{-5}\right)^2}$$

$$=1.4 \text{ x } 10^7$$

The Nusselt number for these cases from Icropera and Dewitt (2001) is:

$$Nu_L = 0.27 {Ra_L}^{1/4}$$

For case 1 (T_e = 100º C)

$$Nu_L = 0.27(6.5 \times 10^7)^{0.25} \qquad = \qquad 24.2$$

For case 2 (T_e = 30º C)

$$Nu_L = 0.27(1.4 \times 10^7)^{0.25} \qquad = \qquad 16.5$$

The convection heat transfer coefficient is therefore

Case 1 $$\bar{h}_c = \frac{24.2 \times 0.0279}{0.25} \qquad = 2.7 \text{ W/m}^2\text{K}$$

Case 2 $$\bar{h}_c = \frac{16.5 \times 0.02545}{0.25} \qquad = 1.7 \text{ W/m}^2\text{K}$$

It would appear that on the roof the difference between the coefficients are not so substantial. To calculate the forced convection values, an approximation of wind speed must be made. From Western Australian Bureau of Meteorology data the average wind speed in this area is 16km/hr (4.4m/s) and the prevailing wind direction is from the west.

The furnaces main axis runs north south . The wind therefore will mainly impinge on the furnaces west face, while the east face will be in a protected position. (Refer Figure 3.4) Assuming that the north, south and west faces and the roof all have the same convective heat loss and have a characteristic length of 2.395m , the losses can be determined from the following:

At 20º C , air has the following properties

$$v^2 = 1.57 \times 10^{-5} m^2 / s$$

$$\rho = 1.16 kg / m^3$$

$$c_p = 1012 J / kgK$$

$$\mathrm{Pr}=0.71$$

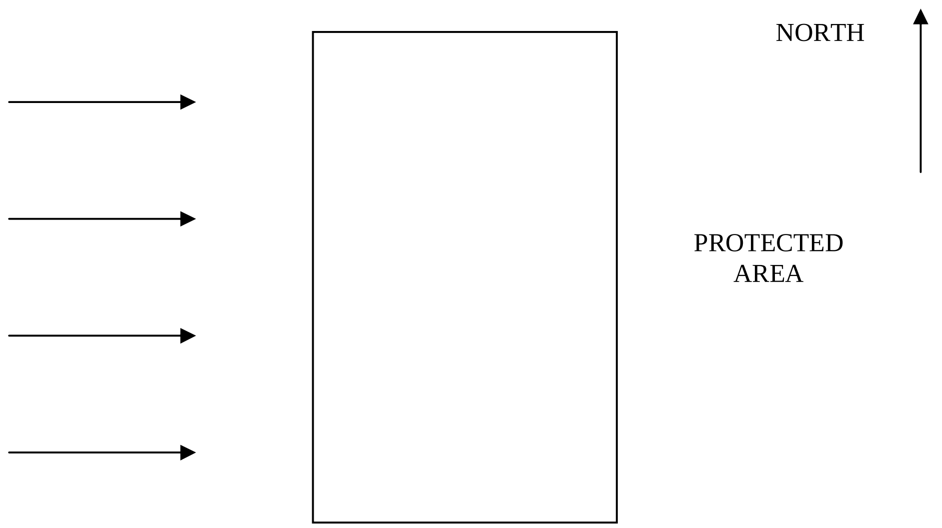


Figure 3.4 Direction of wind loading on furnace

The Reynolds number is given by:

$$\mathrm{Re}_L = \frac{U\infty L}{\nu}$$

$$= \frac{4.4 \times 2.395}{1.57 \times 10^{-5}} = 671210$$

Also $$(\bar{h}_c / c_p \rho U\infty) \Pr^{2/3} = 0.930 \, \mathrm{Re}_L^{-1/2} = 0.00113$$

Rearranging $$\bar{h}_c = \frac{0.00113 \times 1012 \times 1.16 \times 4.4}{0.71^{2/3}} = 7.3 \mathrm{W/m^2K}$$

Heat loss for case 1 (T_e =100º C)

$$\frac{q}{A} = \bar{h}_c \times \Delta T$$

$$= 7.3 \text{ x } 80 = 584 \text{ W/m}^2$$

Heat loss for case 2 (T_e =30º C)

$$\frac{q}{A} = 7.3 \text{ x } 10 = 73 \text{ W/m}^2$$

3.2.5 Radiative heat losses

Radiative heat losses are the only other external heat loss mechanism. Solar radiation flux must be considered when determining radiative heat transfer under normal circumstances. In the case of the furnace there is a weather proofing roof present at around 300mm from the furnace roof. On average it could be assumed that the roof has a mean temperature equal to ambient air.

Radiation from the roof may then be taken to equal (where T_r = external roof temperature):

$$\bar{h}_r = \frac{\varepsilon_1 \sigma (T_e^{\ 4} - T_r^{\ 4})}{T_e - T_r}$$

From the walls, the incident radiation is considered to be negligible and T_r is substituted with T_s being the surrounding temperature at 288K.

Therefore from the roof, (with steel having an emissivity of 0.94) and

$\sigma = 5.67 \times 10^{-8}$ W/m^{2}K^4

For case 1 (T_e = 100º C)

$$\bar{h}_r = \frac{0.94 \times 5.67 \times 10^{-8}(373^4 - 293^4)}{373 - 293} \qquad = \qquad 7.98 \text{ W/m}^2\text{K}$$

For case 2 (T_e = 30º C)

$$\bar{h}_r = \frac{0.94 \times 5.67 \times 10^{-8}(303^4 - 293^4)}{303 - 293} \qquad = \qquad 5.64 \text{ W/m}^2\text{K}$$

From the walls:

Case 1:

$$\bar{h}_r = \frac{0.94 \times 5.67 \times 10^{-8}(373^4 - 288^4)}{373 - 288} \qquad = \qquad 7.8 \text{ W/m}^2\text{K}$$

Case 2:

$$\bar{h}_r = \frac{0.94 \times 5.67 \times 10^{-8}(303^4 - 288^4)}{303 - 288} \qquad = \qquad 5.5 \text{ W/m}^2\text{K}$$

Assuming a shape factor of 1, radiation transfer from outside the furnace is:

$$\frac{\bar{q}_r}{A} = \bar{h}_r \sigma (T_1{}^4 - T_2{}^4)$$

For roof: 338 W/m^2

For walls: 483 W/m2

3.2.6 Other losses

The other losses that are to be accounted for are the combustion losses and flue gas losses. Reed (1991) and Baukal(2001) agree that around 5% of the theoretical energy available from the combustion process is actually lost in the combustion process. They do not however elaborate on what the cause is. This means that of the 468kW initially available, 444kW is actually input to the furnace.

Trinks et al (2003) formulates a method for calculation of $\dot{Q}_{g,net}$ of the flue products as follows;

$$\dot{Q}_{g,net} = \dot{V}_G (P_s + R_s X / 100) \rho_g H_g T_g$$

As the flue gas temperature and therefore the specific enthalpies are difficult to determine it is not considered necessary to attempt this calculation. Rather, an estimate of the flue gas losses will be made in line with commonly known values.

When these values are chosen a “sensible” heat value is available from chart data to establish the useful heat that is supplies to the furnace environment. A common value in use is around 30% of the supplied energy, although this is dependant on the temperature differential between the inner and outer temperatures. Is this is taken as an average value , it should be adequate for this analysis.

In the actual furnace, measurements will be taken with thermocouples attached to the flue exit exterior to try and determine the temperature of the flue gases.

Storage by the refractory walls is only considered as a loss if the furnace is cycled down to ambient temperature and all the stored heat is lost to the atmosphere.

3.3 Thermal interactions inside the furnace

The primary interaction of interest in the furnace is that of the transfer by convection and radiation of heat from the furnace gases and radiation only from the furnace walls to the load.

Determining the relative heat transfer coefficients for either of these mechanisms would involve a project in its own right. A limited discussion will be made here.

3.3.1 Convective Heat Transfer (Internal)

The maximum convective heat transfer available can be determined assuming that the maximum gas velocity impinging on the furnace walls is close to the exit velocity of gas from the burner. Note however that the velocity profile of the gas is extremely difficult to model as there are three other burners in the system creating a very turbulent environment with possibly a number of varying velocity profiles set up throughout the furnace.

The maximum convective heat transfer coefficient can be calculated as follows:

Assuming high velocity speed of combustion products = 152 m/s or lower and also neglecting the existence of a laminar boundary layer, allows the prediction of the convective heat transfer coefficient using the respective Reynolds and Nusselt numbers as follows:

$$\mathrm{Re}_L = \frac{\rho U_\infty L}{\mu}$$

$$\overline{Nu}_L = 0.036 \,\mathrm{Pr}^{1/3} \,\mathrm{Re}_L{}^{0.8}$$

$$\bar{h}_c = \frac{\overline{Nu}_L k}{L}$$

Table 3.1 shows a range of these figures for a $U\infty$ of 152 to 2 m/s in steps of 10, assuming a combustion gas temperature of 1200º C, and using the properties for air. As previously stated, due to the turbulent nature of the furnace environment and the interaction between each burners combustion stream, the assessment of the actual coefficients are extremely difficult, but they do give an indication of what could be expected. The properties for air at 1200º C are as follows:

Density = 0.239 kg/m^3

Viscosity = 218.091 x 10^{-6} m^2/s

Thermal conductivity = 0.08058 W/mK

Specific heat = 1598.534 J/kgK

Prandtl number =0.75 (Raznjevic (1976))

$U\infty$ (m/s)	Re_L	$\overline{Nu}_L$	$\bar{h}_c$ (W/m^{2}K)
152	398941	989	33
142	372695	937	31.5
132	346449	883	29.7
122	320203	829	27.8
112	293956	775	26
92	241464	662	22.3
82	215218	604	20.3
72	188972	544	18.3
62	162726	483	16.2
52	136480	419	14.1
42	110233	354	11.9
32	83987	284	9.5
22	57741	211	7.1
12	31495	130	4.4
2	5249	31	1.0

Table 3.1- Convective heat transfer coefficients for set gas velocities

3.3.2 Radiative Heat Transfer (Internal)

Theoretical radiation transfer can be calculated in accordance with the following:

(from Siegel and Howell, 2001)

$$q_r = \sigma A_G (\varepsilon_G T_G^{\ 4} - \alpha_G T_S^{\ 4})$$

This formula assumes that the surfaces are black, which is obviously not the case for the walls of the furnace or the load, however it is considered sufficient for this exercise.

To determine the emissivity and absorptivity of the gas requires the determination of the partial pressure of the radiating gas components, namely carbon dioxide and water, as the radiation from the diatomic gases are negligible.

For an excess air requirement of 30% , the combustion process is written as follows:

$$C_3H_8 + 6.5(O_2 + 3.76N_2) = 3CO_2 + 4H_20 + 24.4N_2 + 1.5O_2$$

The partial pressure of the radiant gases are given by the following:

$$p_i = (n_i / \sum_i n_i)P$$

where P is the total pressure of the combustion gases (taken as 1 atm)

The radiant gases are CO_2 and H_2O which have molar fractions of and 0.09 and 0.12 respectively. This will also apply to their partial pressures.

To use the radiation heat transfer coefficient equation, , assumptions about the gas shape have to made in order to calculate the effective beam length (L), which is given by taking the mean beam length from tables (assuming the gas shape is rectangular, radiating to all surfaces)

$$L = \frac{M_{BL} \times V}{A}$$

where M_{BL}= mean beam length, V= Volume and A = surface area, being $12m^3$ and $26.2m^2$ respectively.

This gives:

$$L = \frac{0.891 \times 12}{26.2} = 0.41m$$

The product of partial pressure and L then becomes:

For CO_2: 0.09 * 0.41 = 0.037 atm m

For H_2O: 0.12 * 0.41 = 0.049 atm m

For these two partial pressures and for the gas temperatures shown, the emissivity of the gases can be derived from tables (refer Kreith and Bohn Chapter 8, 2001). These values are summarized in Table 3.2

Gas type	**Gas temperature ° C**												
	1300	**1200**	**1100**	**1000**	**900**	**800**	**700**	**600**	**500**	**400**	**300**	**200**	**100**
	Emissivity												
CO_2	**0.055**	**0.063**	**0.065**	**0.072**	**0.078**	**0.081**	**0.085**	**0.085**	**0.085**	**0.08**	**0.075**	**0.078**	**0.085**
H_2O	**0.19**	**0.2**	**0.22**	**0.24**	**0.25**	**0.27**	**0.28**	**0.29**	**0.3**	**0.32**	**0.33**	**0.34**	**0.35**

Table 3.2 – Emissivity values for H_2O and CO_2 at different gas temperatures

The absorptivity is given as a function of the emissivity and the temperature of the gas versus the surface temperature. An example for H_2O follows.

$$\alpha_{H_2O} = \varepsilon_{H_2O} (\frac{T_{H_20}}{T_s})^{0.45}$$

Tables 3.3 and 3.4 summarises this calculated data for various values of gas temperature versus surface temperature for CO_2 and H_2O.

	Temperature Surface							
T gas	100	200	300	400	500	600	700	800
1300	0.60	0.44	0.37	0.32	0.29	0.27	0.25	0.24
1200	0.61	0.45	0.37	0.33	0.30	0.27	0.25	0.24
1100	0.65	0.47	0.39	0.35	0.31	0.29	0.27	0.25
1000	0.68	0.50	0.41	0.36	0.33	0.30	0.28	0.27
900	0.67	0.49	0.41	0.36	0.33	0.30	0.28	0.26
800	0.69	0.50	0.42	0.37	0.33	0.31	0.29	0.27
700	0.67	0.49	0.41	0.36	0.33	0.30	0.28	0.26
600	0.65	0.48	0.40	0.35	0.31	0.29	0.27	0.25
500	0.62	0.45	0.38	0.33	0.30	0.28	0.26	0.24
400	0.60	0.44	0.36	0.32	0.29	0.27	0.25	0.23
300	0.54	0.40	0.33	0.29	0.26	0.24	0.23	0.21
200	0.46	0.34	0.28	0.25	0.23	0.21	0.19	0.18
100	0.35	0.26	0.21	0.19	0.17	0.16	0.15	0.14

Table 3.3- Absorptivity table for H_20

	Temperature Surface							
T gas	100	200	300	400	500	600	700	800
1300	0.17	0.13	0.11	0.09	0.08	0.08	0.07	0.07
1200	0.19	0.14	0.12	0.10	0.09	0.09	0.08	0.08
1100	0.19	0.14	0.12	0.10	0.09	0.09	0.08	0.08
1000	0.20	0.15	0.12	0.11	0.10	0.09	0.08	0.08
900	0.21	0.15	0.13	0.11	0.10	0.09	0.09	0.08
800	0.21	0.15	0.13	0.11	0.10	0.09	0.09	0.08
700	0.20	0.15	0.12	0.11	0.10	0.09	0.09	0.08
600	0.19	0.14	0.12	0.10	0.09	0.09	0.08	0.07
500	0.18	0.13	0.11	0.09	0.09	0.08	0.07	0.07
400	0.15	0.11	0.09	0.08	0.07	0.07	0.06	0.06
300	0.12	0.09	0.08	0.07	0.06	0.05	0.05	0.05
200	0.10	0.08	0.06	0.05	0.05	0.05	0.04	0.04
100	0.08	0.06	0.05	0.04	0.04	0.03	0.03	0.03

Table 3.4- Absorptivity table for CO_2

Calculation of values for qr can then be derived, using the figures in Tables 3.2 to 3.4. If A_G is the contact surface area of the gas with the internal furnace surfaces, for a gas temperature of 1300º C for example, at an initial surface temperature of 100º C , the radiation heat transfer is as follows(for CO_2):

$$q_r = 5.67 \times 10^{-8} \times A_G \times (0.055 \times 1573^4 - 0.17 \times 373^4) \quad \mathrm{W}$$

$$\frac{q_r}{A_G} = 18.9 \text{ KW/m}^2$$

at gas temperature of 1200º C and surface temperature of 700º C, the radiation heat transfer is :

$$q_r = 5.67 \times 10^{-8} \times A_G \times (0.063 \times 1573^4 - 0.07 \times 973^4) \quad \mathrm{W}$$

$$\frac{q_r}{A_G} = 18.3 \text{ KW/m}^2$$

For H_2O the results are as follows: (for T_G=1200º C and T_S= 100º C)

$$q_r = 5.67 \times 10^{-8} \times A_G \times (0.2 \times 1573^4 - 0.60 \times 373^4) \quad \mathrm{W/hr}$$

$$\frac{q_r}{A_G} = 68.7 \text{ KW/m}^2$$

and for T_G=1200º C and T_S= 700º C

$$q_r = 5.67 \times 10^{-8} \times A_G \times (0.2 \times 1573^4 - 0.25 \times 973^4) \quad \mathrm{W/hr}$$

$$\frac{q_r}{A_G} = 56.7 \text{ KW/m}^2$$

Note that these results make a lot of assumptions and do not allow for reflection from inside the gas volume . It would be expected that this internal interaction would represent a large amount of the transferred energy, however this area is one where the authors do not offer much opinion.

3.3.3 Theoretical Heat Load to Achieve Thermal Cycle

If it is assumed that there are a number of possible heat transfer coefficients possible in the furnace ranging from 10 to 100 W/m^{2}K , as a combination of convective and radiative heat transfer mechanisms, The Biot numbers for given thicknesses of steel can be derived from Tables such as in Kreith and Bohn (2001), or in the case of the lumped capacitance techniques, from the following formula:

$$Bi = \frac{\bar{h}L}{k_s} \qquad \text{where L = half the thickness of the slab}$$

A set of these values as calculated for thicknesses from 20 to 100mm is shown in Table 3.5 as well as the reciprocal number 1/Bi. These figures are applicable to an infinite plate of width 2L.

mm	$\bar{h}_c$	10	20	30	40	50	60	70	80	90	100
t=20	Biot Number Bi	0.002	0.005	0.007	0.009	0.012	0.014	0.016	0.019	0.021	0.023
t=40		0.005	0.009	0.014	0.019	0.023	0.028	0.033	0.037	0.042	0.047
t=60		0.007	0.014	0.021	0.028	0.035	0.042	0.049	0.056	0.063	0.070
t=80		0.009	0.019	0.028	0.037	0.047	0.056	0.065	0.074	0.084	0.093
t=100		0.012	0.023	0.035	0.047	0.058	0.070	0.081	0.093	0.105	0.116

mm	$\bar{h}_c$	10.0	20.0	30.0	40.0	50.0	60.0	70.0	80.0	90.0	100.0
t=20	1/Bi	430.0	215.0	143.3	107.5	86.0	71.7	61.4	53.8	47.8	43.0
t=40		215.0	107.5	71.7	53.8	43.0	35.8	30.7	26.9	23.9	21.5
t=60		143.3	71.7	47.8	35.8	28.7	23.9	20.5	17.9	15.9	14.3
t=80		107.5	53.8	35.8	26.9	21.5	17.9	15.4	13.4	11.9	10.8
t=100		86.0	43.0	28.7	21.5	17.2	14.3	12.3	10.8	9.6	8.6

Table 3.5- Biot numbers for various steel thicknesses

For various furnace and load temperature differentials, time periods can be calculated for the load temperature to raise to a certain figure by the use of the relevant Biot number from Table 3.5 and the Fourier number corresponding to the values arrived at once the following expressions has been evaluated. This enables the amount of heat required to achieve the furnace specified thermal cycle. Note that only a few of the Biot numbers are greater than 0.1 therefore the lumped capacitance method can be used for most of the values.

The temperature response for the lumped capacitance method is given by:

$$\frac{T - T_\infty}{T_o - T_\infty} = e^{-BiFo}$$

Given the above , times for the load to reach the prescribed cycle temperatures can be calculated for various thicknesses as shown in Table 3.6, assuming certain heat transfer coefficients and furnace temperatures, only for values of Bi above 0.1 (which is only applicable to two of the values shown in the table, no calculations will be done with this method as it is only applicable to a limited range.

The temperature response for the lumped capacitance method is given by:

$$\frac{T - T_\infty}{T_o - T_\infty} = e^{-(\bar{h}A_s / C_p V)t}$$

with calculated values of BiFo being used instead of the chart method for calculation of the time taken for certain temperature parameters of the material to change under defined temperature load conditions. Various time periods to reach required load center temperatures are then calculated. Further manipulation of the first equation gives the following:

$$\frac{T - T_\infty}{T_o - T_\infty} = e^{-BiFo}$$

$$-\ln\left(\frac{T_i - T_\infty}{T_o - T_\infty}\right) = -BiFo$$

$$\text{where } Fo = \frac{\alpha t}{L^2}$$

$$\Rightarrow \frac{\alpha t}{L^2} = \frac{-\ln\left(\frac{Ti - To}{To - T\infty}\right)}{Bi}$$

$$\Rightarrow t = \frac{-L^2 \ln\left(\frac{Ti - T\infty}{To - T\infty}\right)}{Bi\alpha}$$

Example calculation for table:

For time taken for core temperature to reach 300º C from 20º C with a effective heat transfer coefficient of 10 and a furnace temperature of 1200º C for a 20mm plate. (Biot number = 0.002 from Table 3.5)

$$t = \frac{-L^2 \ln\left(\frac{Ti - T\infty}{To - T\infty}\right)}{Bi\alpha}$$

$$t = \frac{-0.01^2 \times \ln\left(\frac{300 - 1200}{20 - 1200}\right)}{0.002 \times 1.17 \times 10^{-5}}$$

t = 1157.65 seconds

= 0.32 hours

2L = 20mm Ambient temperature Ti=20º C	Time to reach 300º C from ambient		
Furnace temperature	1200º C	900º C	600º C
$\overline{h}_c = 10$	0.32 hr	0.46 hr	0.8 hr
$\overline{h}_c = 50$	0.05 hr	0.076 hr	0.13 hr
$\overline{h}_c = 100$	0.028 hr	0.039 hr	0.07 hr
2L = 60mm Ambient temperature Ti=20º C	Time to reach 300º C from ambient		
Furnace temperature	1200º C	900º C	600º C
$\overline{h}_c = 10$	0.82 hr	1.17 hr	2.0 hr
$\overline{h}_c = 50$	0.16 hr	0.23 hr	0.4 hr
$\overline{h}_c = 100$	0.08 hr	0.12 hr	0.2 hr
2L = 100mm Ambient temperature Ti=20º C	Time to reach 300º C from ambient		
Furnace temperature	1200º C	900º C	600º C
$\overline{h}_c = 10$	5.3 hr	7.5 hr	13 hr
$\overline{h}_c = 50$	1.1 hr	1.6 hr	2.7 hr
$\overline{h}_c = 100$	0.55 hr	0.78 hr	1.35 hr

Table 3.6 Time for load temperature to reach 300º C from ambient

In a similar way, the ramp part of the cycle can be evaluated to ascertain how long the load will take to be raised from 300º C to the soak or hold temperature. This has not been explored any further as this approach is only applicable to a certain load geometry. It is expected that with real measurements, the effective heat transfer coefficient, which encompasses all of the transfer mechanisms, can be established.

3.3.4 Heat storage

In many furnace applications heat storage, or the amount of heat stored in the refractory is not considered as the furnaces are often being constantly reloaded with product and the refractory doesn't actually cool down. In the case of the furnace for this project, the furnace is a batch type and it is unlikely that the next load will be ready before the refractory cools to ambient. In terms of the overall energy utilization the heat storage is very important as there is considerable energy stored in the refractory which does not contribute to any effective thermal processing after the thermal cycle is complete, apart from assist in maintaining a certain temperature as the furnace cools. The storage can be evaluated as follows. A nominal specific heat value of 1070 KJ/kg.K is chosen as the refractory used for the furnace insulation does not have published specific heat information. The formula is given by:

$$Q = c\rho L(Ti - T\infty)$$

Based upon an average refractory density of 92.5 kg/m^3 this gives a value for the walls of :

$$Q = 1070 \times 92.5 \times 0.05(720 - 56) = 3.3 \text{ MJ/m}^2$$

For the sand, the figure is:

$$Q = 830 \times 1600 \times 0.075(720 - 38) = 67.9 \text{ MJ/m}^2$$

Chapter 4 Experimental Results, Measurement and Computer Assisted Modelling

4.1 Experimental results

Conduction experiments were set up in accordance with the following:

Furnace floor simulation:

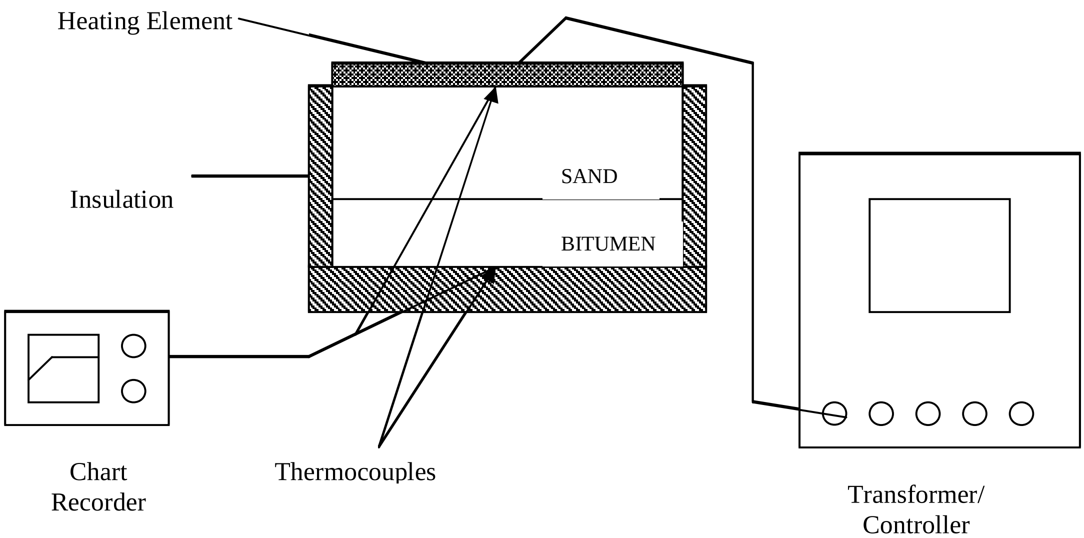


Figure 4.1- Furnace floor conductance experiment setup

The heating element is programmed to reproduce the cycle that would exist under normal furnace operation, the chart recorder records temperature levels from both the top of the sand and the bottom of the bitumen layers, given a graphic representation of the relative temperature states at any given time.

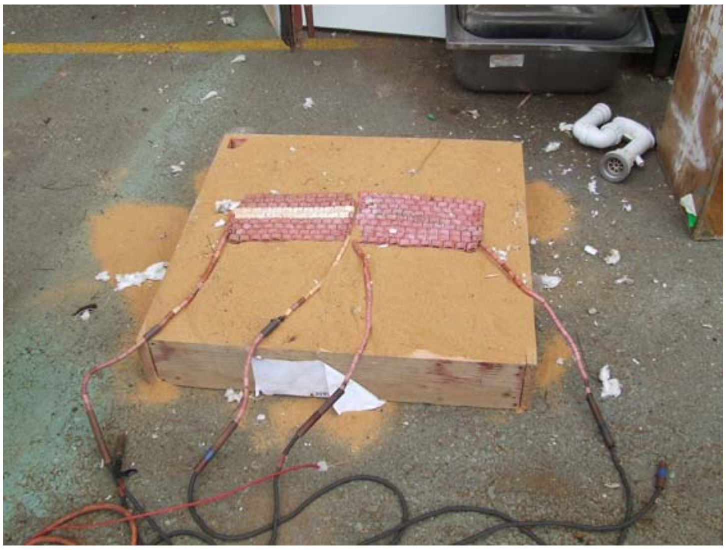


Figure 4.1a- Furnace floor experiment setup (without covering insulation)

Furnace wall simulation (Figure 4.2)

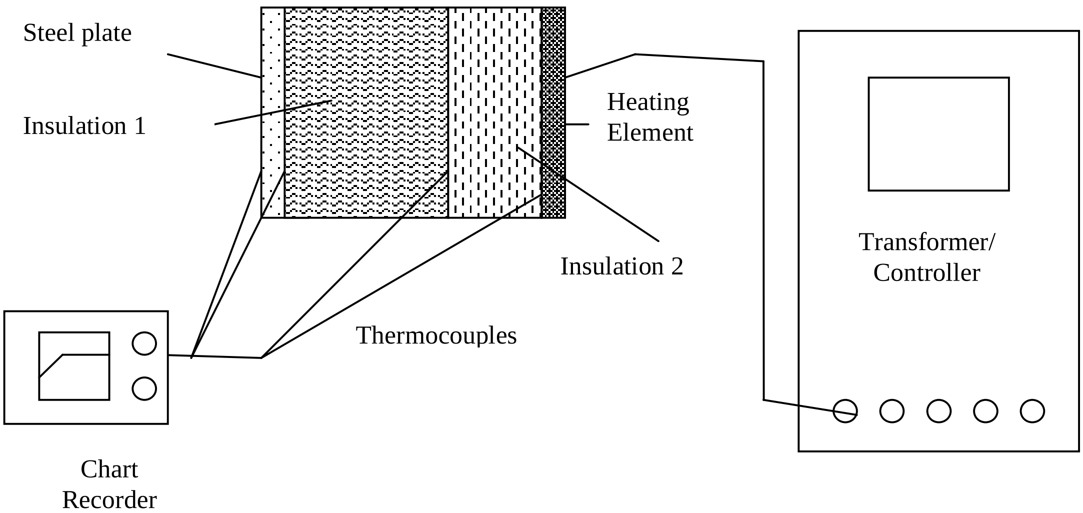


Figure 4.2- Furnace wall conductance experiment setup

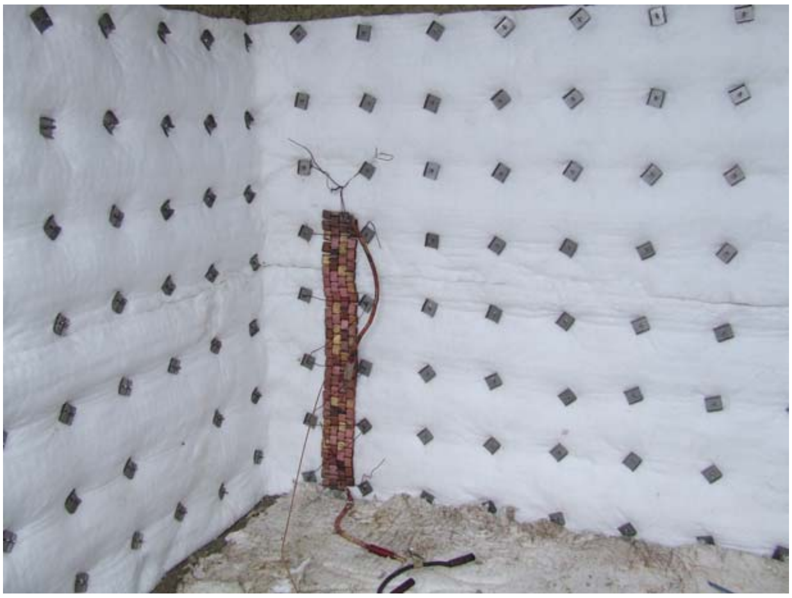


Figure 4.2 a- Furnace wall experiment setup (without covering insulation)

The furnace wall simulation is basically the same as the floor simulation although intermediate temperatures between all layers are measured. For the full furnace simulation , the following thermocouple locations are proposed, in addition to the fixed thermocouples which are permanent fixtures of the furnace wall. (refer Figure 4.3)

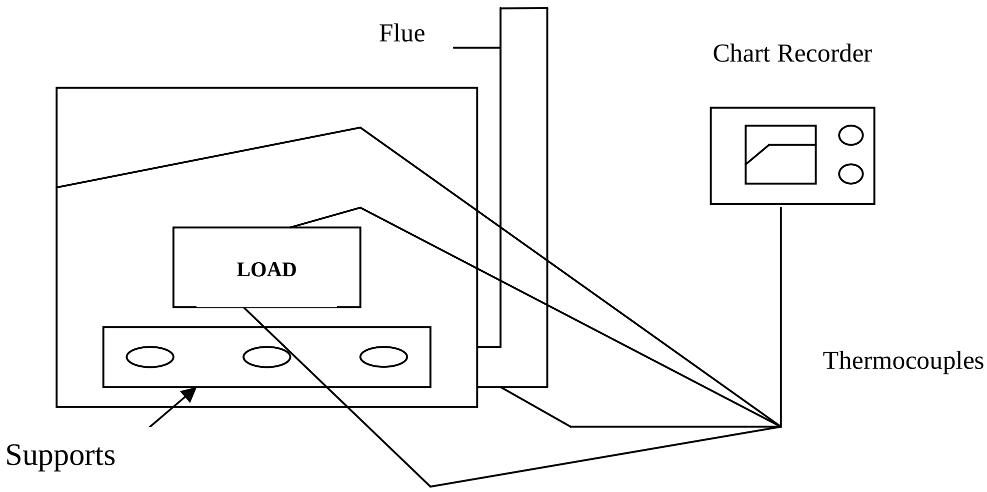


Figure 4.3- Furnace Temperature Measurement Setup

On the basis of the floor conductance chart (Figure 4.4) the steady state external temperature with the internal temperature at 720º C was 38º C. The theoretical thermal conductivity versus heat flux for the wall is derived from:

$$L = \frac{k\Delta T}{q/A}$$

The heat flux supplied is on the basis of the electrical element output, which is nominally 2.7KW and has an effective area of 0.05m^2, giving a heat flux of 135W/m^2. If it is assumed that at least half of this flux goes through the wall (allowing for half to go into the covering insulation) , the calculated k value for the floor would be:

$$0.17 = \frac{k \times 682}{67.5/0.5}$$

k= 0.33 W/m^{2}K

For the wall the external temperature was 56º C at an internal steady state temperature of 720º C. The k value for this setup was therefore:

$$0.103 = \frac{k \times 664}{67.5/0.05}$$

k=0.21 W/m^{2}K

This compares with the calculated figures of

$$q_k = \frac{\Delta T}{L_F / k_F} \quad = 1290.7 \text{ W/m}^2 \text{ (from 3.2.3)}$$

therefore k_F =0.32 W/m^{2}K for the floor

and $$q_k = \frac{\Delta T}{L_W / k_W} = 1195.2 \text{ W/m}^2$$

therefore k_w = 0.185 W/m^{2}K for the wall

Reasonable correlation between the figures has been achieved.

4.2 Computer assisted thermal analysis

The demonstration versions of WinTherm, Nastran and Thermal Desktop were used to attempt to simulate the furnace model. Apart from the most basic conduction models , the node limitations on the software prevented the modeling of the furnace overall and in addition the modeling at the combustion heat source was unachievable.

To fully enable this element of the project to be completed , a full version software package will have to be purchased. The restrictions of the software available at this time made this exercise of no practical significance to the project.

4.3 Measurement techniques to be used with the furnace

The main items used in the furnace to measure temperature are two off thermoelectric pyrometers. These are permanently attached to the furnace, unlike the thermocouples which are attached to various points in the load. The principle is as follows with reference to Figure 4.4 . The hot junction (formed by dissimilar metals joined together at the end produce an emf when subjected to a temperature. This is measured by the meter or recorder through compensating leads . The compensating leads are used in lieu of the two (considerably more expensive) leads so that the distance between the meter and the pyrometer does not affect the measured emf to any great extent.

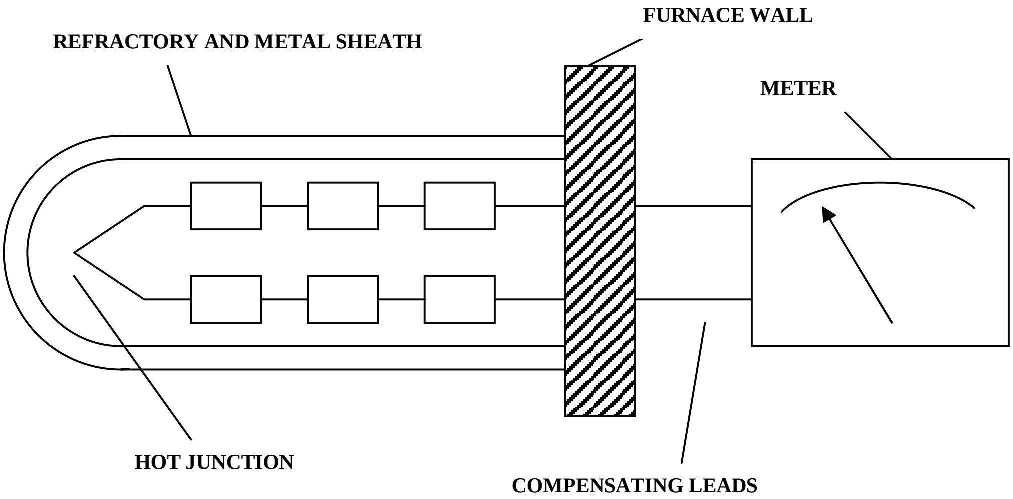


Figure 4.4 – Primary measurement device for furnace temperature

Measurement of the gas temperature in the furnace is considerably more difficult. The only way to achieve this is by the use of a suction pyrometer which extracts the gas into a radiation shielded chamber with a thermocouple inside. This allows the measurement of the gas without radiation effects giving false indications. As these are quite expensive, requiring both the measuring unit and pumping equipment, they were not considered for this stage of the project. Radiation pyrometers which are purely concerned with measuring radiation being emitted to the furnace walls are also available. Once again these are extremely expensive but may be considered in the future if there appears to be a need for more accurate measurements of this phenomenon.

Measurement of relative gas constituency is also possible and may give some benefits in understanding the internal mechanisms of the furnace. Childs, P (2001) describes a couple of techniques for measurement of both O_2 content in gas mixtures and measurement of exhaust gases from various processes. It is possible that these may be used in the future to measure the constituency of flue gases. Michalski, L et al. (2001) describe difficulties with measurement of the temperature of high velocity gas streams and the techniques to try and ascertain these. Given the context of this project, it is considered uneconomical to attempt to perform any of these proposed measurement techniques. The application of the temperature measuring devices described herein will therefore form the basis for measurement when the furnace is completed.

Chapter 5- Major System Components

Apart from the thermal circuit component, there are other elements of in the furnace that require design to enable the furnace to work effectively and safely. These are the:

- The structural component
- The Gas System/Combustion Component
- The electrical component

5.1 Structural component

From a structural perspective, the furnace frame and furnace frame support has to achieve the following:

- Furnace frame and walls- Suitable rigidity to withstand wind loads
- Lifting devices- Suitability to lift full mass of furnace, including frame, refractory , burners, piping, electrical and fan.
- Mounting points- Suitability to take loads imposed by the fan rotation and burner thrust and weight.
- Floor- Suitable to take the loading imposed by the furnace structure , furnace load and base ,material.

5.1.1 Furnace frame and walls

A review of the Australian Standards for residential/light commercial slabs and footings (AS2870) , the SAA Loading Code (AS1170) and also Steel Structures (AS4100) was conducted to establish the adequacy of the structure in terms of these standards.

As the structure only weights under 2000kg, and is not bolted to the ground, it is not considered necessary to perform a detailed analysis of these aspects. The load from the furnace to the ground is reasonably well distributed by the four walls. The only factor considered of relevance is the wind loading. As the structure is fairly light, and the walls represent a substantial wind resistance, it is considered necessary to tie each corner down to a concrete block of at least 500kg, through tensioned cables. There are also bolting points which connect to hold the structure together. These comprise a captive nut welded to the internal section of the frame and a bolt running through the bolting fixture, which is a corner section welded to one edge of the frame and bolted to the adjoining member (refer Appendix B)

5.1.2 Lifting devices

In the case of the furnace there are two types of lifting devices:

- Panel lifting devices
- Overall furnace lifting devices

The panel lifting devices are primarily for locating the panels for furnace assembly, whilst the overall furnace lifting devices lift the furnace on and off the load.

The overall devices therefore have to lift the full weight of the full furnace assembly, with fan, burners, electrical and plumbed fittings , furnace panels and refractory.

The full weights of these elements are shown in Appendix E . These figures include all attachments to the panel sections. Assessment of the lifting lug requirements is presented below (Refer Figure 5.1)

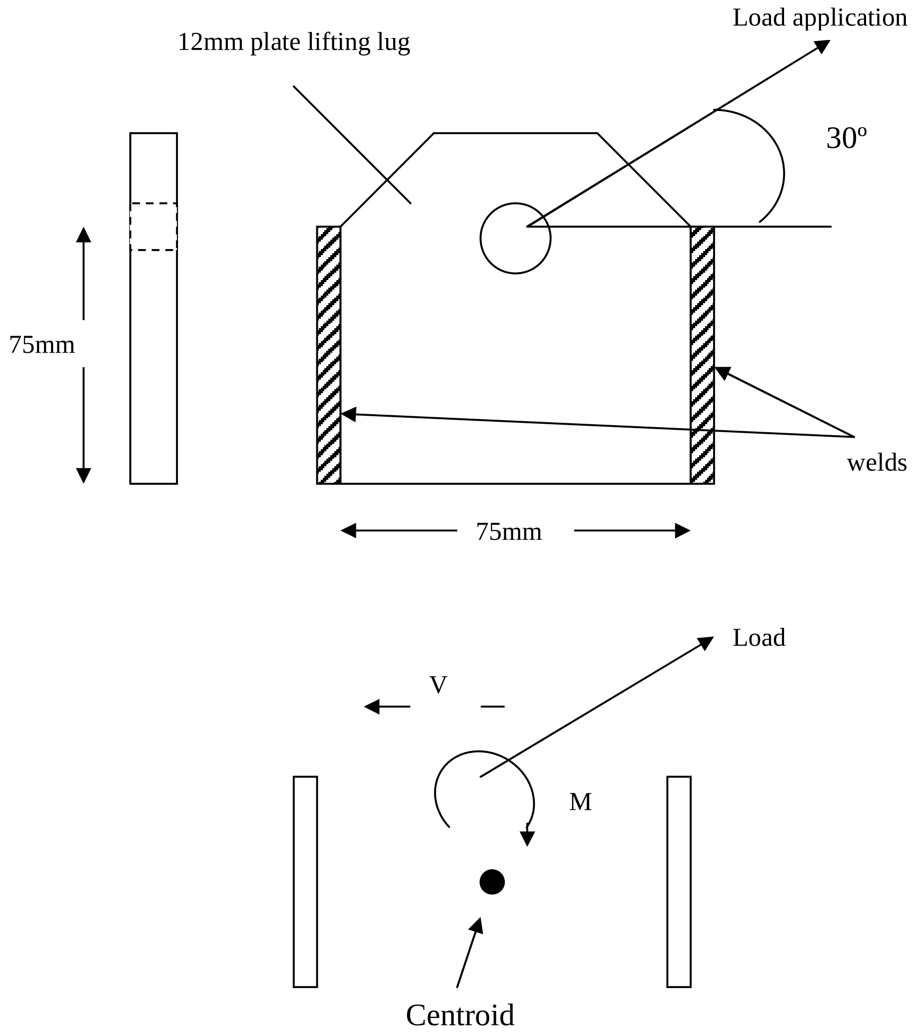


Figure 5.1- Lifting lug configuration and loading diagram

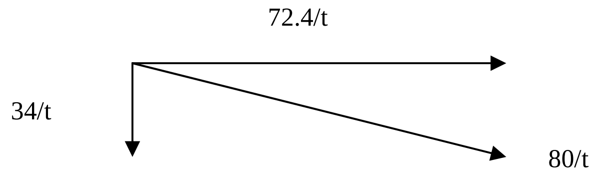
Bending stresses:

Load conditions are as follows for side panels: Total weight 425.4 x 9.8 = 4169 N

Load on one lug = 2084.5 N , $F_x = F_y/\tan 30º = 3610$ N.

$$\tau' = \frac{V}{A} = \frac{3610}{1.414t \times 0.075} = \frac{34}{t} \text{ MPa}$$

$$\sigma = \frac{Mc}{I_x} = \frac{Mc}{\frac{d^3 t}{6}} = \frac{72.4}{t} \text{ MPa}$$



Assuming E60 welding rods (Sy = 350 MPa) and applying a safety factor of 2 gives the following, using distortion energy theory (Juvinall, Marshek , 2000)

$$Ss_y = S_y \times 0.58 = 203MPa$$

$$\frac{80}{t} = \frac{203}{2} = (\frac{S_{sy}}{SF}) \quad \Rightarrow t = 0.79mm$$

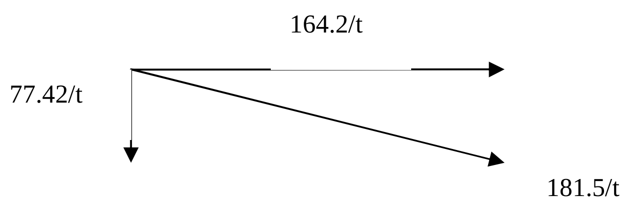
As the minimum fillet size for AS4100 is 3mm this will be the minimum weld size used. The axial load is given by:

$$\sigma = \frac{F_y}{tL} = \frac{2090}{150t} = \frac{13.9}{t} \quad \Rightarrow t = 0.13mm$$

For the full furnace lifting lugs , the load on each lug is F_y= 4740 N and F_x=8210 N. Weld areas, lengths and centroids are all the same as the first lug setup.

$$\sigma = \frac{Mc}{I} = \frac{307875 \times 37.5}{70312.5t} = \frac{164.2}{t}$$

$$\tau = \frac{V}{A} = \frac{8210}{1.414t \times 0.075} = \frac{77.42}{t}$$



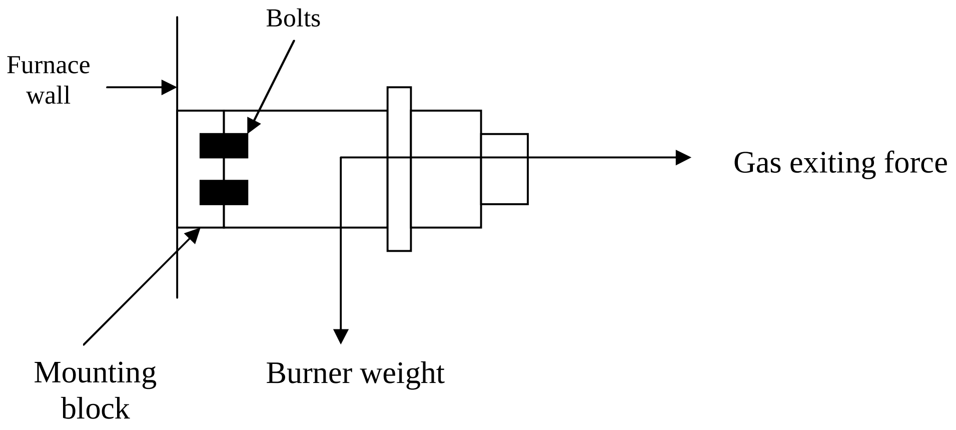
Therefore: $$\frac{203}{2} = \frac{181.5}{t} \Rightarrow t = 1.8mm$$

A 3mm fillet will also be adequate.

5.1.3 Mounting Points

These are two major mounting point types requiring consideration and these are the burner mounts and fan mounts. The forces imposed by these loads are shown below:

The burner and fan mounting configurations are as follows (Figure 5.2):



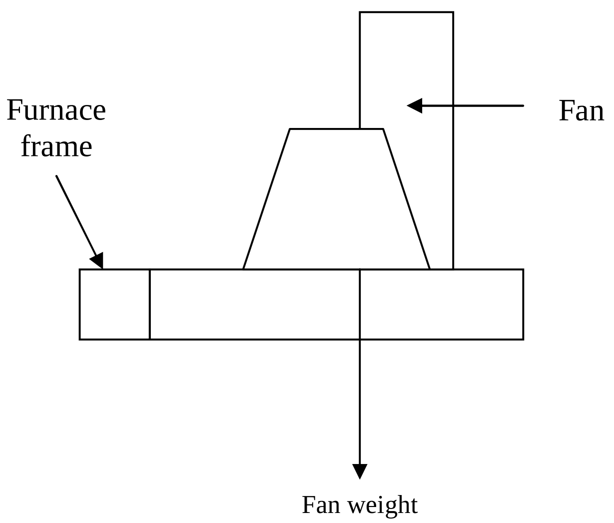


Figure 5.2- Burner and fan mounting diagrams

The fan also causes a moment to be imparted on the mounting due to impeller rotation. As the discharge is pointed upwards, this not affect the structure to any great extent so wont be considered in the mounting force analysis.

The fan mounting primarily has to take the weight of the fan when the furnace is lifted. There is a direct shear stress and a bending moment stress to be accounted for, and the moment of inertia also has to be accounted for. The fan load is only a fraction of the loads imposed by the lifting lugs so a 3mm or greater weld thickness will be satisfactory.

5.2 Gas System/Combustion Component

There is some overlap between the electrical and gas systems in the furnace but essentially they are separate entities.

The combustion/gas model in simple terms can be described as follows:

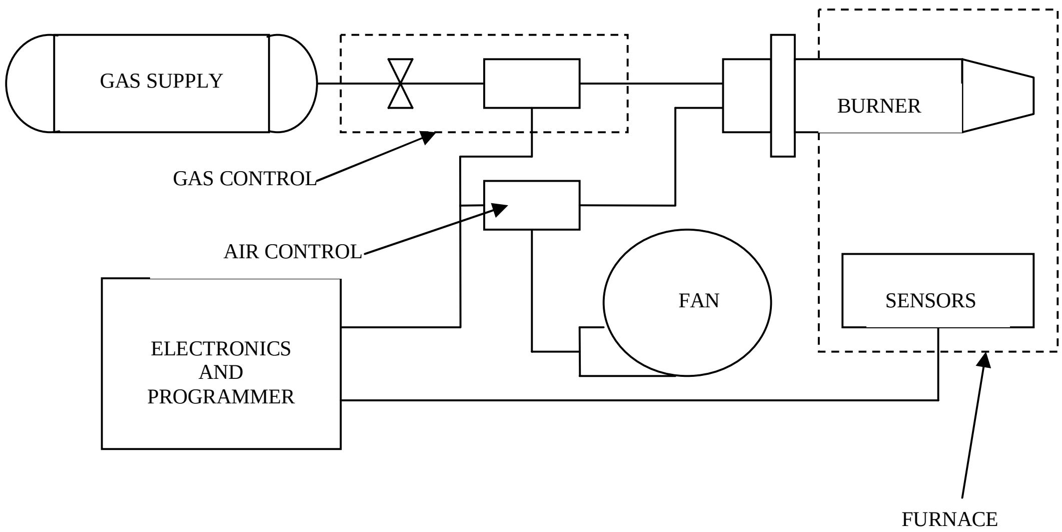


Figure 5.3 - Combustion system model

The designed system is as shown in Appendix C.

MAJOR SYSTEM COMPONENTS

A description of the way the control system regulates the required gas and air supply (as directed by the programmer) is as follows:-

The ratio regulator senses the air pressure to the burner through the impulse line. Once the burner is lit, the air control butterfly opens up and provides a large volume of air while the gas is at its "high fire" setting. This ensures the high combustion gas velocity required in the furnace to stir the furnace atmosphere (and also to improve the convective heat transfer coefficient).

To control the heat, the bleed off control valve bleeds off a part of the impulse line pressure signal that the ratio regulator receives, which tricks the ratio regulator into sensing a low air pressure. The gas is then turned down by the regulator to maintain the correct ratio , although the air is actually still at high pressure.

This allows variation of the flame temperature while still maintaining high velocity.
In addition , on the impulse line, there is a limiting valve which regulates the impulse air flow so that the bled off air is not made up by an increase in the impulse line pressure.

Using this system gives a thermal turndown of around 40 to 1. If a 1000MJ/Hr high fire setting exists it can be turned down to 25MJ/hr on low fire. The temperature of the combustion products can theoretically be altered from 100 to 1500º C.

Another consideration of the system is the production of unwanted combustion products. If low gas and high air is used in a cold furnace, a large amount of CO is produced. To prevent this, the air control butterfly is held at low fire until a furnace temperature of 500º C is reached. This has an effect on the rapidity with which the thermal cycle can be fulfilled but needs consideration to reduce unwanted emissions.

5.3 Electrical Component

The electrical schematics are as shown in Appendix C. A brief description of the system is as shown in Figure 5.4.

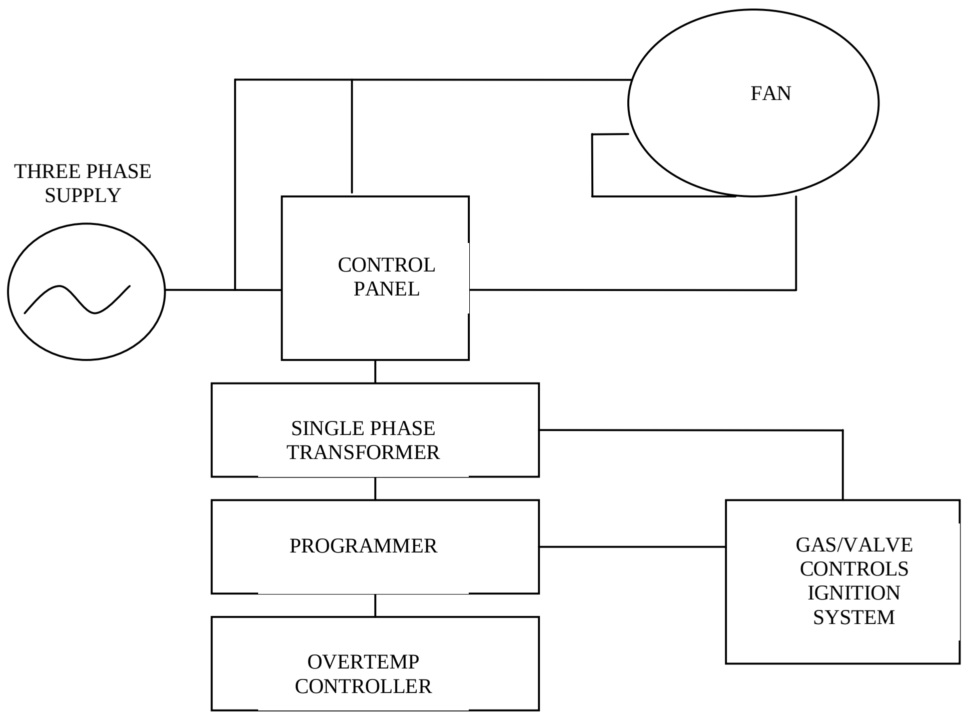


Figure 5.4- Basic Circuit Overview

Essentially the electrical system consists of a three phase power supply which supplies the control panel and the fan. This in turn drives a single phase transformer which drives the programmer and overtemp controller, and hence the gas/valve controls and the ignition system.

The programmer is the core of the system and is set in such a way as to reproduce the up and down ramps as shown in Figure 2.2.

The overtemp/limit controller cuts in if for some reason the load or furnace temperature has exceeded a preset point, to ensure no damage is done to the refractory surfaces and/or the load.

Chapter 6- Consequential Effects and Risk Analysis

6.1 Risk Analysis

Prior to commencement of this project activity a risk analysis was conducted to try and establish the expected hazards to be encountered in the project.

The most obvious issue is the explosive and fire risks associated with use of the gas. Lesser risks include electrical hazards, and accidents associated with the fabrication of the furnace itself.

The use of qualified tradesmen to perform the gas and electrical work as well as being a statutory requirement reduces these risks considerably, however, once the furnace becomes operational, a maintenance regime must be established to ensure continuity of furnace safety. The selection of a designer for the gas and electrical control system, rather than following the manufacturers fairly detailed instructions for the design and commissioning of the combustion system, is also considered a risk reduction exercise.

Refer to Appendix 6 for the full risk analysis.

6.2 Consequential Effects

Consequential effects can be broken down into the following areas:

- Sustainability
- Safety
- Ethics

6.2.1 Sustainability

The concept of sustainability brings forward the subject of Life Cycle Assessment (LCA). Life Cycle Assessments are a tool that can be used to determine the sustainability of a product from a conceptual to a disposal stage. AS/ISO 14040:1998 describes methodologies for performing a life cycle assessment as follows:

The assessment is defined as “the compilation and evaluation of the inputs , outputs and the potential environmental impacts of a product system through its life cycle”

Fabrycky and Blanchard (1991) break the product life cycle into two main phases and these are: Acquisition and Utilisation. Referring to Figure 6.1 these phases are further broken down into other sections. It is clear from these diagrams that major improvements to the life cycle process can be made at the earliest stages of a project. (i.e in the conceptual stages). As this project is not necessarily a new design , the conceptual decisions have already been made , however, future refinements to the system may be made with more of an improved life cycle approach built in.

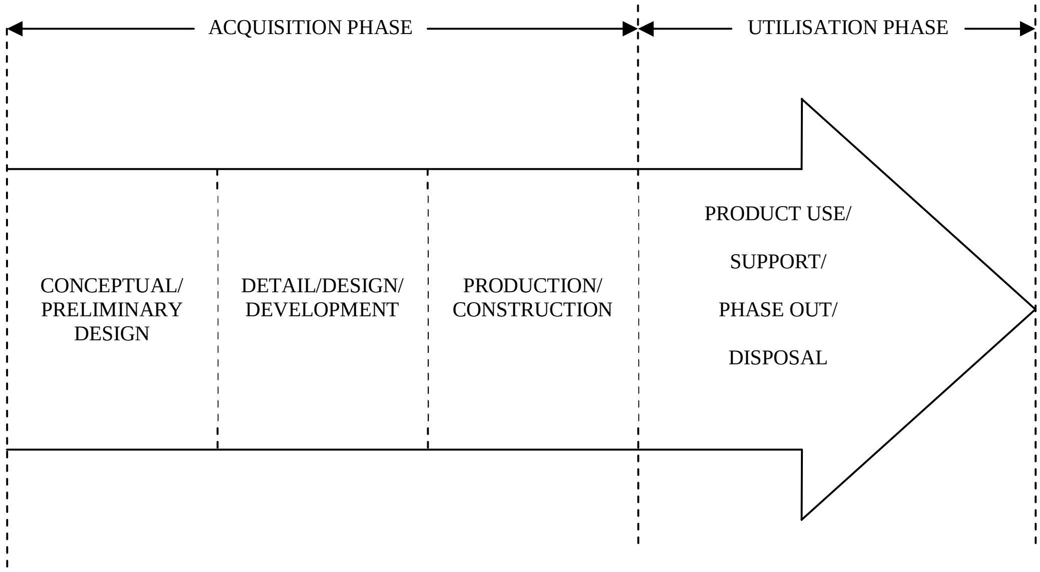


Figure 6.1- Acquisition/Utilisation Flow Diagram

As it stands there are a number of ways this project impacts on the environment and they are in the construction phase and the utilization/disposal phase. Figure 6.2 summarises the interrelations between the inputs and outputs to the system.

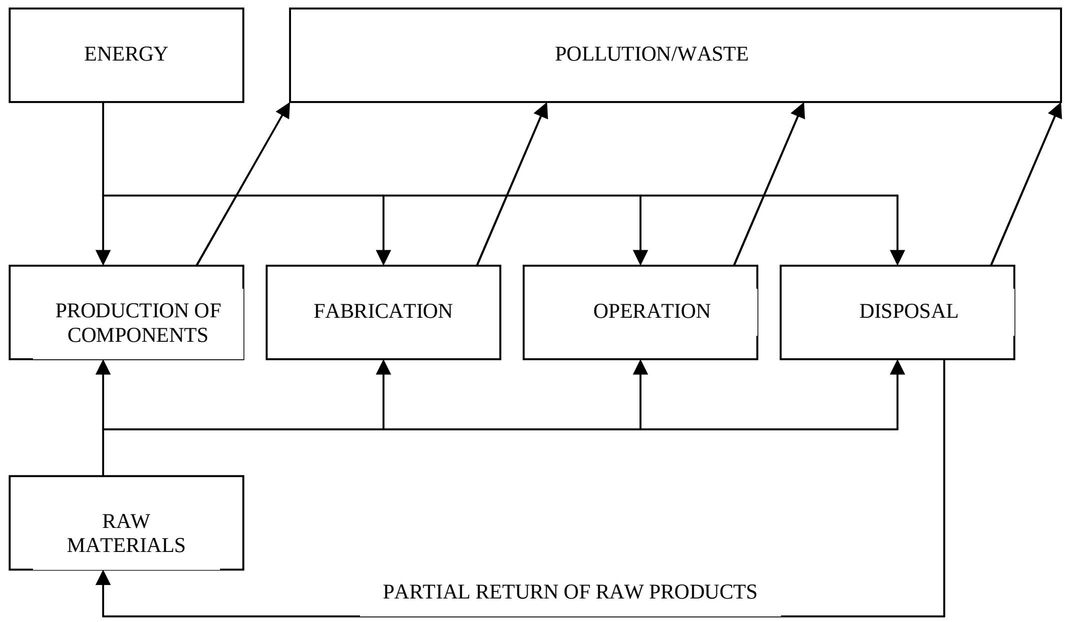


Figure 6.2- Flow diagram for furnace life inputs and outputs

The main issues regarding the life cycle of the furnace are:

- The disposal of the refractory materials
- The waste of large volumes of gas
- Polluting gases (including greenhouse gases)

The disposal of the refractory materials poses an ongoing environmentally negative output of the furnace. The materials are non-biodegradable, retain water and are irritants to skin and lungs. If the landfill soils are reused at some time in the future these pose considerable problems once the refractory has again dried out, as it can easily become airborne once dry. The refractory needs to be replaced regularly as it wears out due to thermal cycling.

The furnace produces large volumes of combustion gases, mainly CO_2, CO and NO, which are non-desirable. Over the life of the furnace, the volumes of these products released to the atmosphere will be substantial. Energy losses are another unwelcome outcome of the furnace operation . As the process is extremely inefficient, it is primarily environmentally unsound from all perspectives.The main reasons for the use of gas in furnaces such as these is to heat treat large masses of product quickly.

It poses a serious challenge for the future to invent new ways to perform these activities whilst restricting the waste of energy resources and generation of pollution , both airborne and to landfill.

6.2.2 Safety

As discussed in Section 6.1, the consequences of improperly maintaining the furnace could be deadly. If the gas system develops leaks or improper firing, explosion risks are very high which could lead to major injury, equipment damage, or death. Deterioration or damage to the system is also a potential life threatening risk.

Also related to maintenance is the replacement of worn out refractory materials. Over time, the glass wool becomes powdery and must be replaced. The wool is a skin and respiratory irritant and must be handled with care.

This can have a negative impact on the maintenance teams health. It is essential that appropriate Personal Protective Equipment is used whenever performing these activities.

6.2.3 Ethical Issues

The main ethical issues associated with the project relate to the safety and sustainability issues raised by the previous two sections.

Neglecting either of these and not trying to provide improvements in future developments of the furnace (unless outrageously expensive) would constitute a breach of the ethics of the Institution of Engineers.

Because the project is a real world entity, and not just a theoretical exercise, these issues have a long term implication.

Chapter 7- Results and Conclusions

7.1 Results Summary

Essentially the core of the project has attempted to derive reasonable approximations for the various phenomena that occur in and around the furnace environment. At the outset of the project it was clear that there would be some difficulty in assessing certain components but nevertheless an attempt has been made to assess the core system parameters. Table 7.1 summarises the particular elements assessed during the project and the relative perceived accuracy of these figures, with reference to the single gas zone model presented in Figure 2.3.

Element	Description	Accuracy of derivation	Importance of model accuracy
Q_G/Q_A	Gas/Air energy input	Reasonable	High
Qg	Flue gas losses	Estimated only	High
Q_{re}	External radiation losses	Estimated only	Low
Q_{ce}	External convective losses	Estimated only	Low
Q_{cw}	Convective transfer gas to wall/floor	Reasonable	Low
Q_{cl}	Convective transfer gas to load	Estimated only	Medium
Q_{rgl}	Radiative transfer gas to load	Estimated only	High
Q_{rqw}	Radiative transfer gas to wall/floor	Estimated only	High
q_{kw}/q_{kf}	Conductive heat transfer through wall and floor	Reasonable through experimentation and calculation	Medium
Q_{kl}	Conductive heat transfer in load	Estimated only	Low

Table 7.1 Summary of derived parameters and assigned accuracy

As can be seen from this table, a large number of the high importance figures are estimated only, and in many cases are based upon specific shape factors , as well as guesses about specific heat transfer coefficients are what the environmental factors are expected to be. It should be noted that many of the phenomena described in the literature relate to clearly defined geometries. In the case of the furnace , very simple shapes (such as flat plates) have been evaluated. Further explanation of the relative merits of the techniques used in the project are described briefly here.

Combustion- the energy conversion process and the effectiveness thereof is well understood and documented. The actual performance of the burner systems will only be properly evaluated when the furnace is operating and possibly with the analysis of the flue gases with one of the measurement techniques discussed in section 4.3. The total gas input for a given thermal cycle will be established by use of a gas metre on the piping to the furnace. The electricity utilized in the process to operate the control system and the fan will also be taken into account. The total energy used to perform a specific thermal cycle will then be evaluated against the theoretical energy input shown in Section 3.1. From this an overall system efficiency figure can be obtained. It is already clear (without these calculations) that the system is extremely inefficient when all the losses are considered. Exactly how much so will be judged by the energy usage versus the theoretical optimum figures.

Conductive heat losses- compared to the flue losses these represent a much smaller quantity of the total losses. Importantly, these are one element of the furnace that can be modified fairly easily by the designer, by the utilization of materials with smaller conductivity ratings. The down side of this is that the specific heat, which adds to the heat storage losses, also has to be accounted for. Nevertheless, it is likely that this area of the furnace is one that may be improved with technological advances in materials.

Radiative and convective heat losses- these are fixed and cannot be varied by the user to any great extent as they are linked to the available heat coming through the furnace. They are a reasonably small loss in terms of the overall. The only way to eliminate these losses is to reduce the external temperature to that of the ambient temperature and this is obviously not practical. Once again technological advancements may improve this area of the furnace performance. The conduction through the walls and floor should theoretically match the convective and radiative losses externally to achieve an overall heat flux balance, however due to the inaccuracies of the calculations these do not match. Referring to Table 7.2 it can be seen that conduction through the walls and roof equals 37.4 KW/m^2 whilst the convective and radiative transfers add up to 30.6 KW/m^2. Note also that the surface areas have been made equivalent. Edge effects at the corners and other factors have not been considered. Factoring these in may lead to convergence of the two figures.

Internal heat transfer mechanisms- the techniques used to evaluate these were the most theoretical of the project and based upon guess work and use of theory that may not necessarily have been applicable to the particular application being reviewed. The load conduction lends itself to much difficulty in deriving experimentally. Unless an accurate thermal measurement can be made from selected points within the object undergoing the thermal cycle, temperature versus time phenomena cannot be established. This would be the ideal way to understand the apparent heat flux on the external surfaces of the item, assuming that it meets the shape criteria that has documented conduction theory applicable to it.

Table 7.2 summarises the derived values from Chapter 3.

Parameter	**Derived Value**	**Quantity**	**Total Value**	**% of input power**
Conductive transfer wall	1269 W/m^2	29.5 m^2	37.4 KW	8.4 %
Conductive transfer floor	1291 W/m^2	5.9 m^2	7.6 KW	1.7 %
Convective transfer external	584 W/m^2	29.5 m^2	17.2 KW	3.9 %
Radiative transfer external roof	338 W/m^2	4.7 m^2	1.6 KW	0.4 %
Radiative transfer external walls	483 W/m^2	24.8 m^2	11.9 KW	2.7 %
Convective heat transfer internal	33 W/m^2	N/A*	N/A*	N/A*
Radiative heat transfer internal	87.6 W/m^2	N/A*	N/A*	N/A*
Heat Input	444 KW	1	444 KW	100%
Flue losses	133 KW	1	133 KW **	30%
Heat Storage floor (based upon total ten hour cycle)	67.9 MJ/m^2	5.9 m^2	400.6 MJ = 11.1 KW	2.5 %
Heat Storage walls (based upon total ten hour cycle)	3.3 MJ/m^2	29.5 m^2	97.3 MJ =2.7 KW	0.6 %

*- N/A – Not assessed- The transfer mechanism is a general term and may not necessarily apply to gas to wall transfer.

**- Energy losses to the flue could be considerably more (maybe up to 50%) as suggested in some texts

Table 7.2 Derived values for various furnace parameters

It should also be noted that many of the values derived are for a specific instance of the thermal cycle. There is constant variability in many of the parameters over time as the temperature differential is constantly changing and also because the furnace controllers are changing heat input as well. Given this high degree of variability in many of the parameters, to truly determine the actual performance of the furnace, it will be necessary to trial the furnace and then take extensive measurements from as many points as possible to develop a truly representative picture of the various parameter values that exist, if indeed they can be separated from one another.

As all aspects of the temperature at a specific moment in time change many of the parameters, it would also be considered necessary to establish a much more sophisticated mathematical model for all the parameters . How to actually establish all elements of this model would require an extensive research project in its own right.

7.2 Conclusions

7.2.1 Comparisons

A review of the furnace losses show that at least 50% of the energy input is lost in non-constructive processes such as convective, radiative, conduction losses to outside but primarily through the flue gases. Some of this energy can be fed back into the burners by recirculation, but requires a different kind of burner to be employed, with added expense. In addition, the use of excess air as a control mechanism is energy inefficient and a pulse fired burner system almost eliminates this problem. This again is a fairly expensive upgrade to this basic furnace design. It is unlikely in the near future that a quantum leap will be made in the efficiency of furnace designs of this nature. A review of the other types of stress relief is considered here to offer alternatives to this approach. These are as follows:

Induction heat treatment- An high frequency electromagnetic field is induced in the steel by a surrounding coil which sets up a current within the material. This in turn produced heat which be controlled to achieve set thermal parameters. It is generally restricted to simple geometries and small parts as it is difficult to generate a uniform electromagnetic field in complex shapes.

Vibratory stress relief- In this process the structure is subjected to a series of varying vibratory excitations while being isolated from fixed surface by rubber mountings. The concept is that the exciting energy allows redistribution of the internal stresses. The process takes a much shorter time that the furnace work but is limited to smaller structures also. It can also be used as the welding is carried out. Obviously this has some unwanted time delays on finishing the fabrication.

Electric Element- Ceramic heating elements are wrapped around the part and current passed through them, thus heating the product. These elements represent a much more direct means of applying heat than does the furnace but is restricted by the size of the elements, the power availability to run the elements and is very time consuming in the setup of the equipment, as it is very labour intensive. Nevertheless, in terms of energy efficiency, it have massive advantages over the furnace system, as do all the alternatives. Practicality dictates that the furnace technique will continue to be used until a major advance in stress relieving processes is found.

7.2.2 Achievement of objectives

The research aspect of the project was achieved satisfactorily, although differing opinions were obtained from different sources. Parameters required for the furnace design were derived satisfactorily, although further work can be done on environmental data in the furnace vicinity. A suitable housing was designed for the furnace and the thermal circuit model for the furnace was established. The use of computer software to analyse the system was of no benefit except to highlight the difficulties with modeling the complex environment within the furnace. This area was not pursued as it was obvious that the basic software available was incapable of the complicated modeling required of this area. The prototype furnace was almost completed in time for performance evaluation, however monetary constraints from the company and lack of subcontract labour availability especially in the wiring of the control panel made this impossible. This in turn affected the evaluation and extrapolation exercises detailed in the Project Specification- Appendix A (Points 5 and 6)

The two remaining objectives (Appendix A – Points 9 and 10) were partially completed with the life cycle assessment being addressed in Chapter 6.

In summary, the project has been successful in achieving the majority of objectives and lays a foundation for re-examining the various methodologies used for determining specific furnace parameters. When the prototype furnace is eventually operational, the remaining elements of the project specification will be finalized and comparisons/adjustments made to the model.

Chapter 8- Further Work

As this is a real world project and the furnace will be utilized as an operating device, as mentioned in the conclusion, the assessment of the prototype will be carried out in due course and various parameters will be varied to assess furnace characteristics.

In addition to this, the purchase of a full version heat transfer software package will enable spatial distribution of temperature with varying shaped loads and varying input and refractory conditions to be assessed without the extremely tedious mathematical modeling . This can only be done by using the software and developing skills over a period of time, so that less complicated thermal models must be attempted prior to this extremely complicated model.

The company who sponsored this project will be using parts of this paper to introduce new employees and trainees to the theory behind the furnace concept and the shortfalls of the technology with respect to the energy efficiency and environmental problems.

The search for much better ways to perform this kind of operation without the excessive usage of gas must be a priority for any personnel involved in this industry, but especially for engineers and scientists who can possibly make these changes happen.

References

1. AS2870-1996, *Residential Slabs and Footings- Construction,* Standards Australia, Homebush, NSW

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Appendix A

Project Specification

APPENDIX A PROJECT SPECFICATION

University of Southern Queensland
FACULTY OF ENGINEERING AND SURVEYING

ENG 4111/4112 RESEARCH PROJECT

PROJECT SPECIFICATION

FOR: **BRETT GRIBBLE**

TOPIC: STRESS RELIEVING FURNACE MODELLING AND ANALYSIS

SUPERVISORS: H KU and S GOH (University of Southern Queensland)
TECH ADVISOR: Pat Colgan, Puni Holdings Pty Ltd

ENROLMENT: ENG 4111-S1, EX, 2006
ENG 4112-S2, EX, 2006

PROJECT AIM: The project aims to design and analyse the operation of a gas fired stress relieving furnace using traditional thermal design and computer assisted techniques and the effects of system variables on the performance of the furnace.

SPONSORSHIP: Aztec Heat Treatment

PROGRAMME: **Issue B, 23rd March, 2006**

PHASE 1- PROJECT APPRECIATION

Prepare project appreciation assignment based upon points 1 through 10 as detailed in PHASE 2

Begin: 27th March 2006
Completion: 22nd May 2006
Hours: 50 hours

PHASE 2- PROJECT EXECUTION

1. Research furnace technology related to the project with respect to the structural, combustion and control systems, and the thermal circuit model.

   Begin: 22nd May 2006

   Completion: 16th June

   Approx. Hours: 40 hours

2. Establish and specify parameters which will require consideration in the design process. (e.g. heating and cooling rates, hold time at temperature, maximum temperature, wind loading on furnace)

   Begin: 16th June 2006
   Completion: 22nd June 2006
   Approx. Hours: 10 hours

3. Design a suitable insulated housing and heating system using traditional thermodynamic theory to achieve the design parameters. Establish a thermal circuit model for the furnace.

   Begin: 22nd June 2006
   Completion: 8th July 2006
   Approx Hours: 25 hours

4. Analyse the housing and system using one or more thermal analysis software packages.

   Begin: 8th July 2006
   Completion: 24th July 2006
   Approx.Hours: 25 hours

5. Construct prototype furnace(s) and heating system based upon points 1-4 and assess the performance of the prototype with differing mass, furnace volume and heat input.

   Begin: 24th July 2006
   Completion: 28th August 2006
   Approx. Hours: 60 hours

6. Extrapolate results from point 5 to predict the performance of a full sized furnace.

   Begin: 28th August 2006
   Completion: 3rd September 2006
   Approx. Hours: 10 hours

7. Prepare draft dissertation for submission to supervisor

   Begin: 16th August 2006
   Completion: 22nd Sept 2006
   Approx. Hours: 50 hours

8. Prepare final form of dissertation and submit

   Begin: 7th October 2006
   Completion: 2nd Nov 2006
   Approx. Hours: 30 hours

As time permits:

9. Perform a basic life cycle assessment of the furnace and system construction and assess more environmentally friendly alternatives

   Begin: 1st September 2006
   Completion: 8th September 2006
   Approx. Hours: 8 hours

10. Assess fully sized furnace for practical performance and compare to theoretical.

    Begin: 8th September 2006
    Completion: 15th September 2006
    Approx. Hours 8 hours

AGREED:

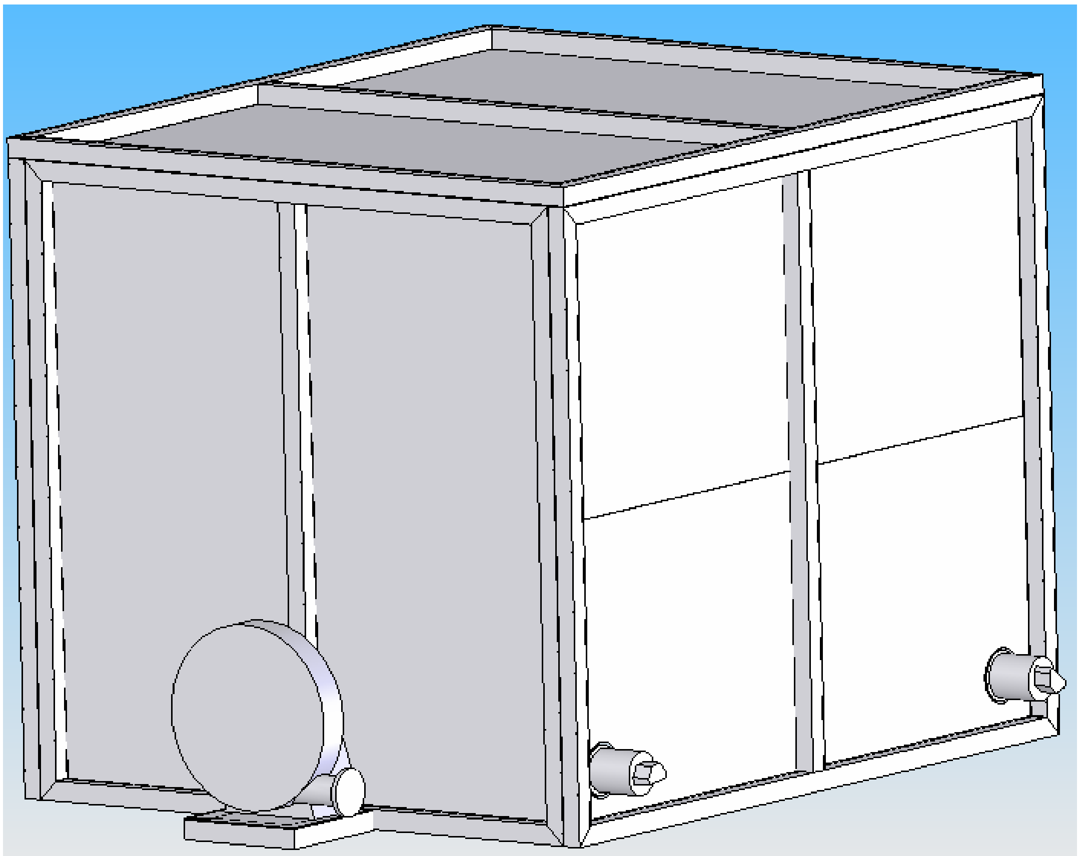
__________________________ (Student) ____________________, __________________ (Supervisors)

____/ ____/____ ____/ ____/____ ____/ ____/_

Appendix B

Furnace Overview

APPENDIX B Furnace Overview



Furnace Overview

Fabricated Furnace Panel

Partial Assembly showing refractory pins

Partial Assembly showing Insulation Layers

APPENDIX B Furnace Overview

Overall view of partial assembly

Gas Burner

Air Blower

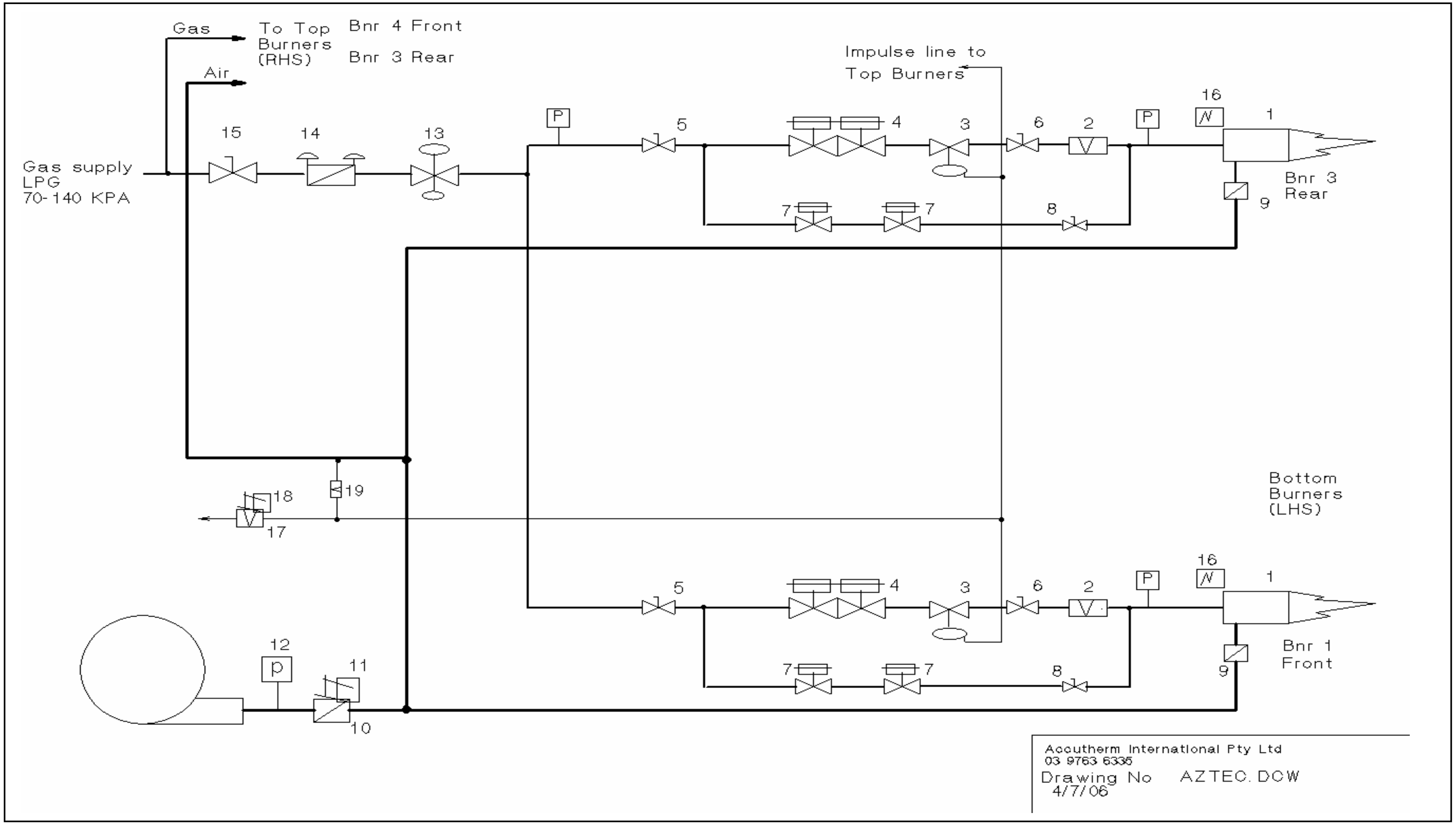
Gas Train Assembly

Outside of Control Panel

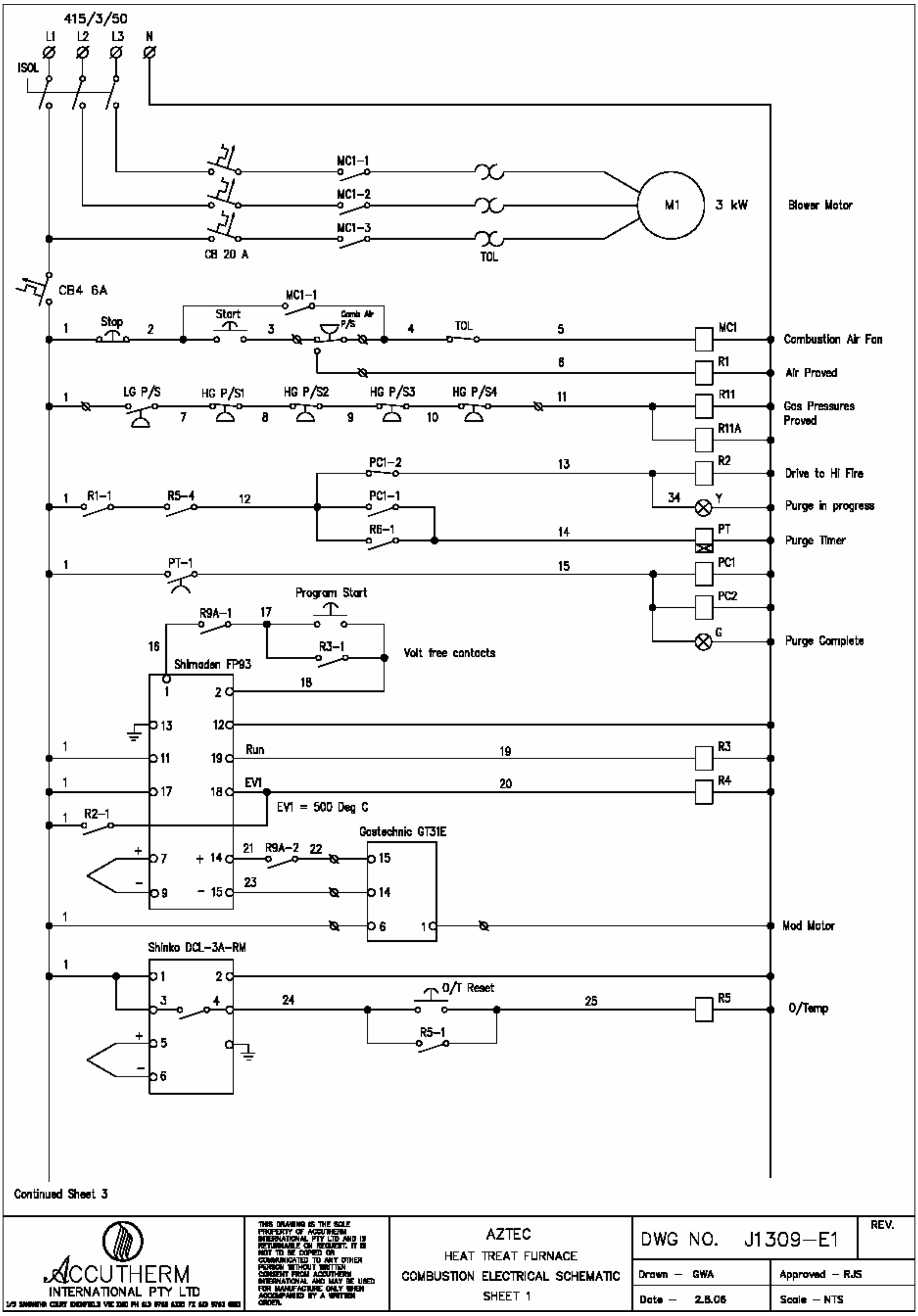
Inside of Control Panel

Appendix C

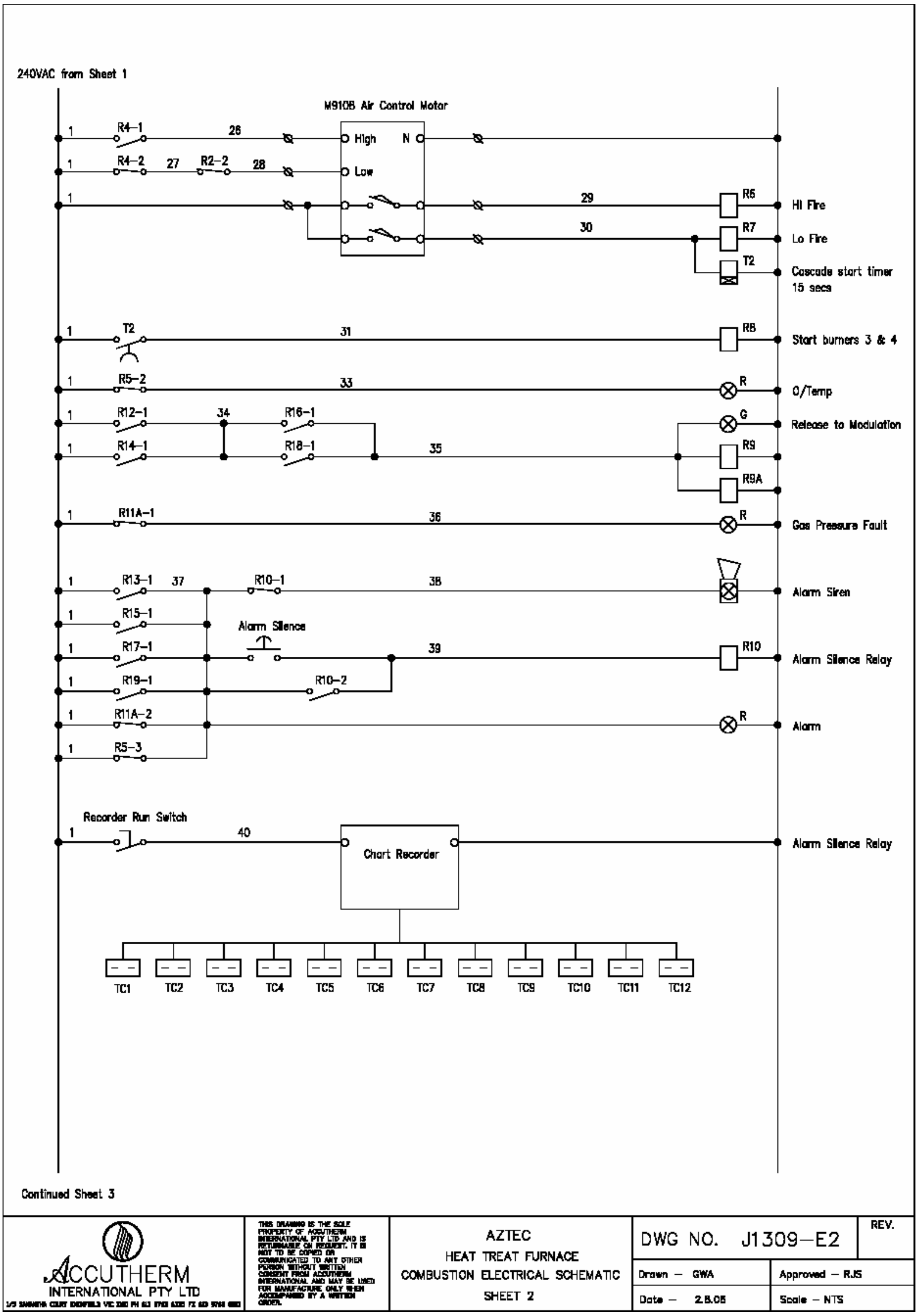
Control Schematics and Equipment Data



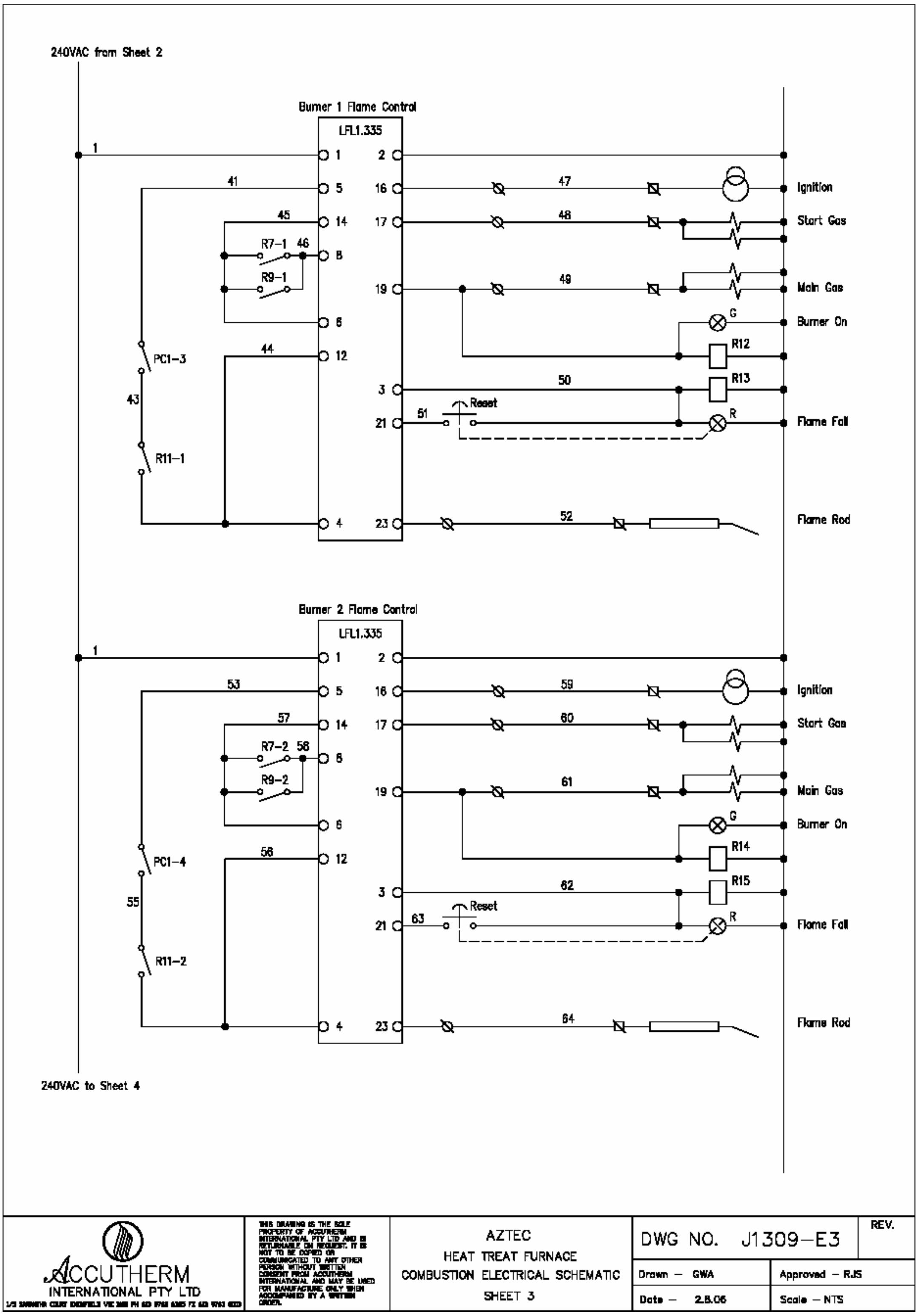
Gas burner Control Schematic



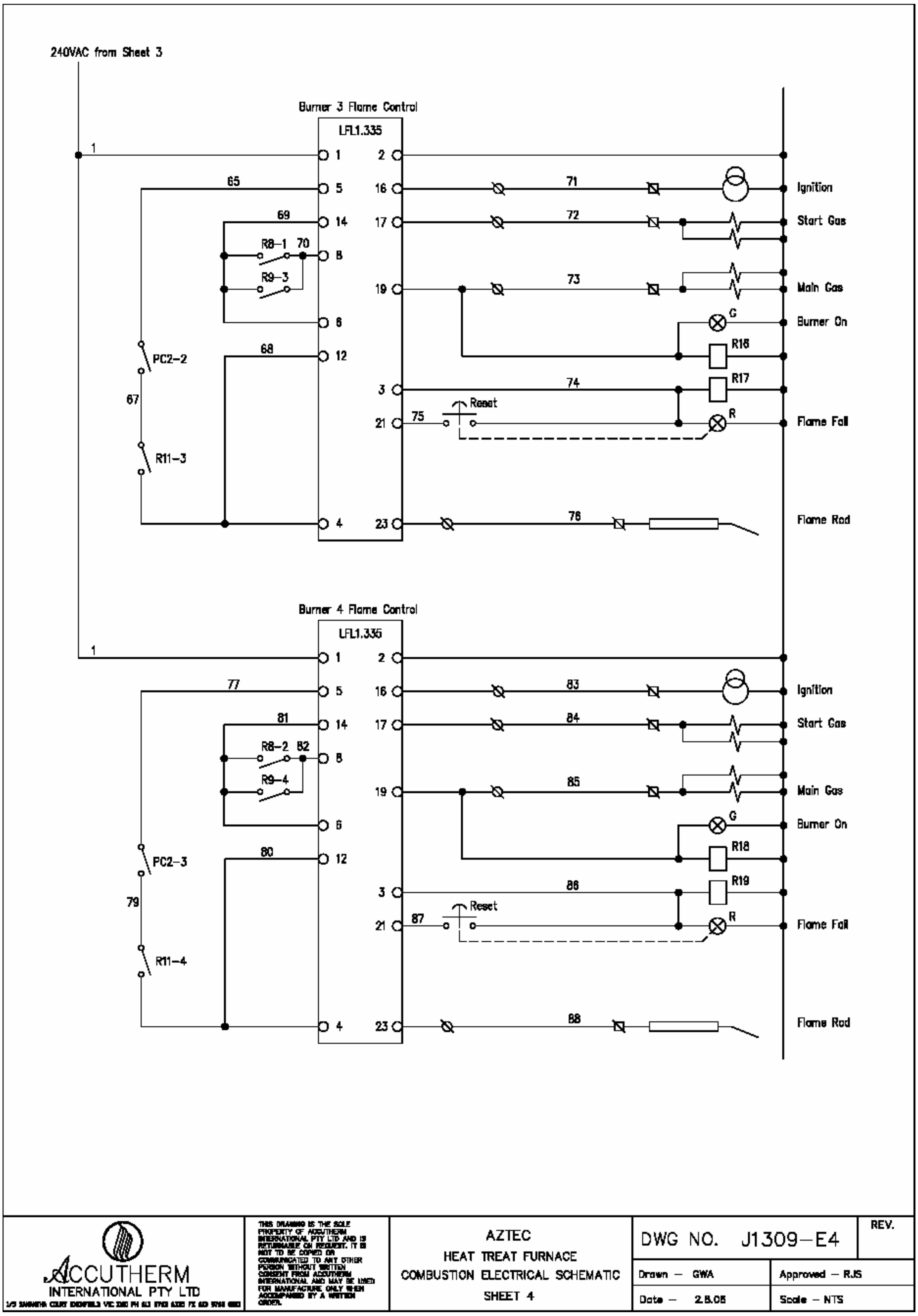
Electrical Control Schematic Page 1



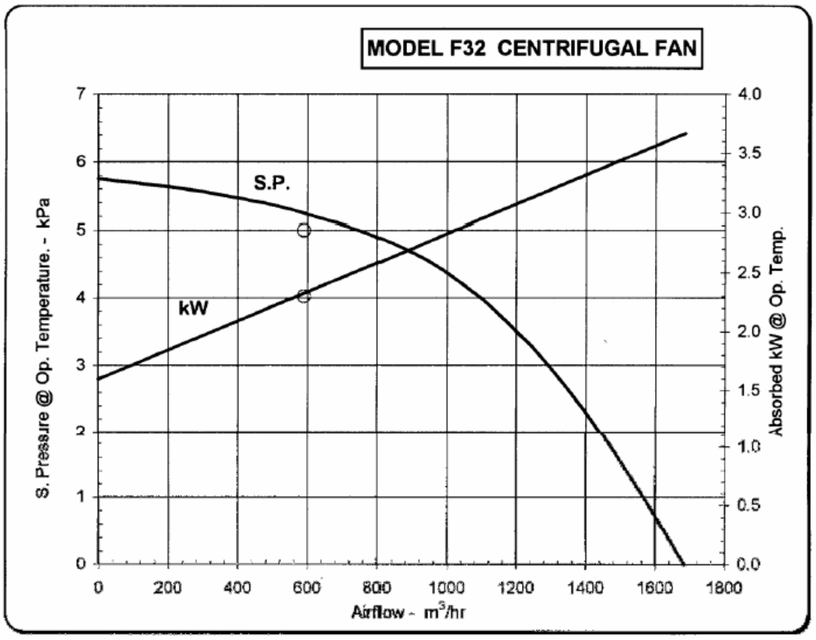
Electrical Control Schematic Page 2



Electrical Control Schematic Page 3



Electrical Control Schematic Page 4

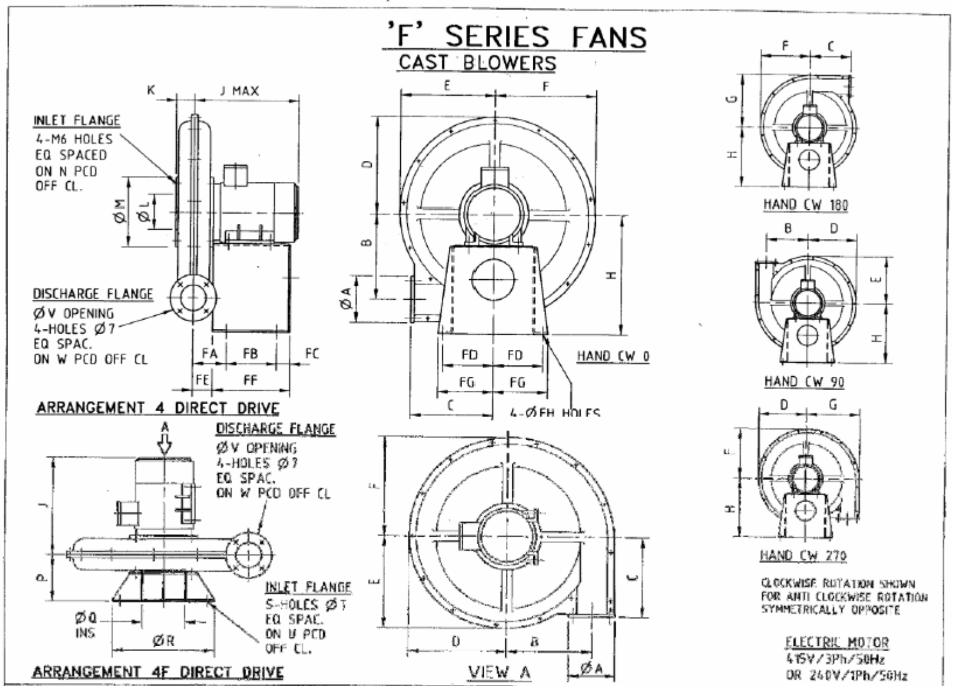


Fan impeller (wheel) diameter	mm	550
Fan speed	rpm	2900
Airflow @ fan inlet	m3/hr	590
Inlet static pressure @ oper. temp.	kPa	0.0
Outlet static pressure @ oper. temp.	kPa	5.0
Allowance made for silencer loss	kPa	0.000
Total static pressure @ oper. temp.	kPa	5.0
Operating temperature	^{0}C	20.0
Absorbed power @ operating temp.	kW	2.3
Absorbed power @ 20^0 C	kW	2.3

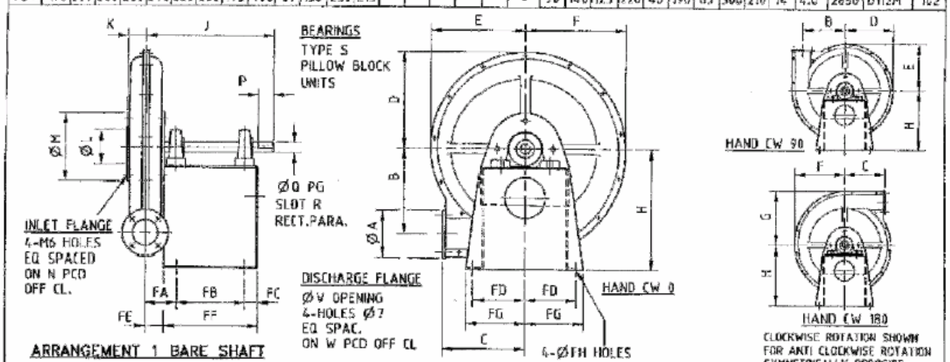
Altitude	m	0
Inlet density	kg/m^3	1.200
Tip speed	m/s	83.6
Outlet velocity	m/s	25.8
Inlet velocity	m/s	12.3
Inlet diameter, I.D.	mm	130
Outlet diameter, I.D.	mm	90
Fan casing thickness	mm	5.0

Octave band centre frequency, Hz	63	125	250	500	1000	2000	4000	8000	TOTAL	
Fan internal Sound Power Level, dB	106	103	104	100	101	97	94	90.7	111	dB
Fan inlet or outlet S. Pressure L., dB	95	90	91	87	88	84	81	77.7	92	dBA
Casing breakout S. Pressure L., dB	87	81	78	70	68	63	59	54.2	74	dBA
Inlet or outlet S.Pressure L. with silencer, dB										dBA
Casing breakout with lagging dB										dBA
Motor noise - free field , dBA @ 1metre										dBA

The above noise data is based on free field (no reflecting surface) conditions at 1metre. If the fan is installed in an enclosed area, the final noise level would increase by approximately 3 dBA for single reflecting wall (floor), 5 dBA for two reflecting walls (floor and 1 wall), 8 dBA for three reflecting walls (floor and 2 walls ie. corner). Noise levels do not include noises generated by bearings, vee belts, cooling fins etc. Casing breakout noise is through fan casing only and does not include breakout through ducting or flexible connections.



	GENERAL										INLET FLANGE									DISCH.		FOUNDATION FOOTINGS								MOTOR			WT
SIZE	A	B	C	D	E	F	G	H	J	K	L	M	N	P	Q	R	S	T	U	V	W	FA	FB	FC	FD	FE	FF	FG	FH	KW	RPM	FRAME	KG
F10	112	185	205	205	200	225	235	275	235	36	90	156	130	105	96	250	6	10	210	65	97	70	110	35	120	45	170	135	10	0.55	2850	D71	23
F20	137	250	240	285	275	295	315	350	290	52	76	200	165	140	125	300	8	12	256	76	110	98	150	40	145	58	230	160	12	1.5	2850	D90S	44
F22	137	250	240	285	275	295	315	340	290	52	100	200	165	140	125	300	8	12	256	76	110	98	150	40	145	58	230	160	12	1.1	2850	D80	38
F24	137	250	240	285	275	295	315	350	290	52	100	200	165	140	125	300	8	12	256	76	110	98	150	40	145	58	230	160	12	1.5	2850	D90S	44
F26	137	250	240	285	275	295	315	350	290	52	100	200	165	140	125	300	8	12	256	76	110	98	150	40	145	58	230	160	12	2.2	2850	D90L	46
F28	137	250	240	285	275	295	315	350	290	52	125	200	165	140	125	300	8	12	256	76	110	98	150	40	145	58	230	160	12	2.2	2850	D90L	46
F32	170	309	300	360	346	366	380	463	383	61	130	250	215	-	-	-	-	-	-	90	140	123	220	40	190	83	300	210	14	3.0	2850	D100L	85
F34	170	309	300	360	346	366	380	475	406	61	130	250	215	-	-	-	-	-	-	90	140	123	220	40	190	83	300	210	14	4.0	2850	D112M	102



	GENERAL										INLET FLANGE							DISCH.		FOUNDATION FOOTINGS								WT
SIZE	A	B	C	D	E	F	G	H	J	K	L	M	N	P	Q	R	S BRGS	V	W	FA	FB	FC	FD	FE	FF	FG	FH	KG
F10	112	185	205	205	200	225	235	275	260	36	90	156	130	40	25	8	UKP 206	65	97	75	110	35	120	36	104	135	10	18
F24	137	250	240	285	275	295	315	350	380	52	100	200	165	50	30	8	UKP X07	76	110	93	280	40	150	53	280	170	12	30

Fan Data Page 2

Data 205-2.5
4/20/04

ThermJet Burners

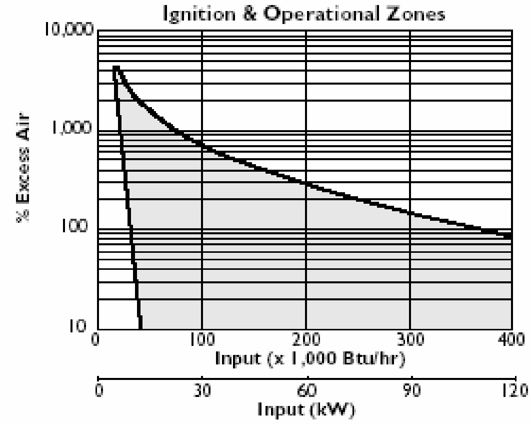
Model TJ0040
Version 2

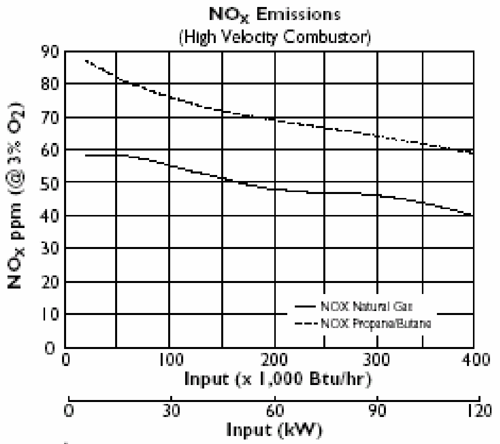
PARAMETER	BURNER VELOCITY		MODEL TJ 0040
Maximum input Btu/hr (kW)	Medium & High Velocity		400,000 (117)
Minimum Input, on-ratio Btu/hr (kW)	Medium & High Velocity		40,000 (11.7)
Minimum Input, fixed air (Btu/hr)	Medium & High Velocity		8,000 (2.3)
Gas inlet pressure required "w.c. (mbar) • Fuel pressure at gas inlet (Tap "B" – see page 3)	High Velocity	Nat. Gas Propane Butane	12.0 (29.9) 13.0 (32.4) 12.0 (29.9)
	Medium Velocity	Nat. Gas Propane Butane	5.5 (13.7) 5.5 (13.7) 5.0 (12.5)
Air inlet pressure required "w.c (mbar) • 15% excess air at maximum input (Tap "A" – see page 3)	High Velocity	Nat. Gas Propane Butane	15.5 (38.6) 17.0 (42.3) 17.0 (42.3)
	Medium Velocity	Nat. Gas Propane Butane	9.0 (22.4) 9.5 (23.7) 9.5 (23.7)
High Fire Flame Length Inches (mm) (measured from end of combustor)	High Velocity	Nat. Gas Propane Butane	14.0 (356) 17.0 (432) 17.0 (432)
	Medium Velocity	Nat. Gas Propane Butane	18.0 (457) 19.0 (483) 19.0 (483)
Maximum flame velocity ft/s (m/s) • 15% excess air, at maximum input	High Velocity		500 (152)
	Medium Velocity		250 (76)
Flame detection	U.V. scanner available for all combustors Flame Rod available for use with alloy or silicon carbide combustors only		
Fuel	Natural Gas, Propane, Butane *For any other mixed gas, contact Eclipse for orifice sizing.*		

- All information is based on laboratory testing in neutral (0.0"w.c.) pressure chamber. Different chamber size and conditions may affect the data.
- All information is based on standard combustor design. Changes in the combustor will alter performance and pressures.
- All inputs based upon gross caloric values.
- Eclipse reserves the right to change the construction and/or configuration of our products at any time without being obliged to adjust earlier supplies accordingly.
- Plumbing of air and gas will affect accuracy of orifice readings. All information is based on generally acceptable air and gas piping practices.
- Do not install the burner with the gas inlet rotated 90° clockwise with respect to the air inlet if operating on Natural Gas and using a flamerod.

Burner Data Page 1

Performance Graphs



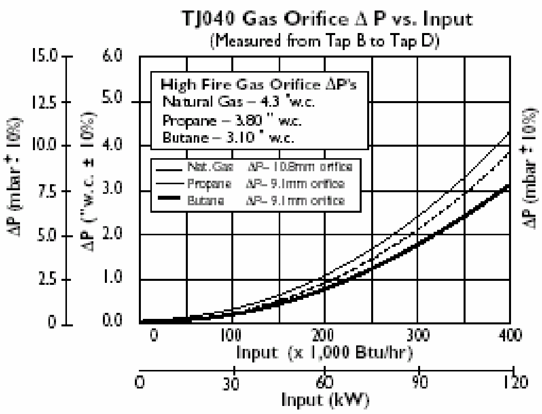


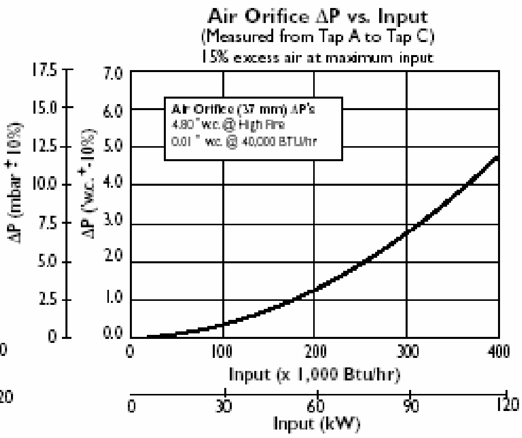
Correction factor for medium velocity combustor is 1.20
On Ratio: Combustion air is adjusted to 3% O_2 at any given input.

Emissions from the burner are influenced by:

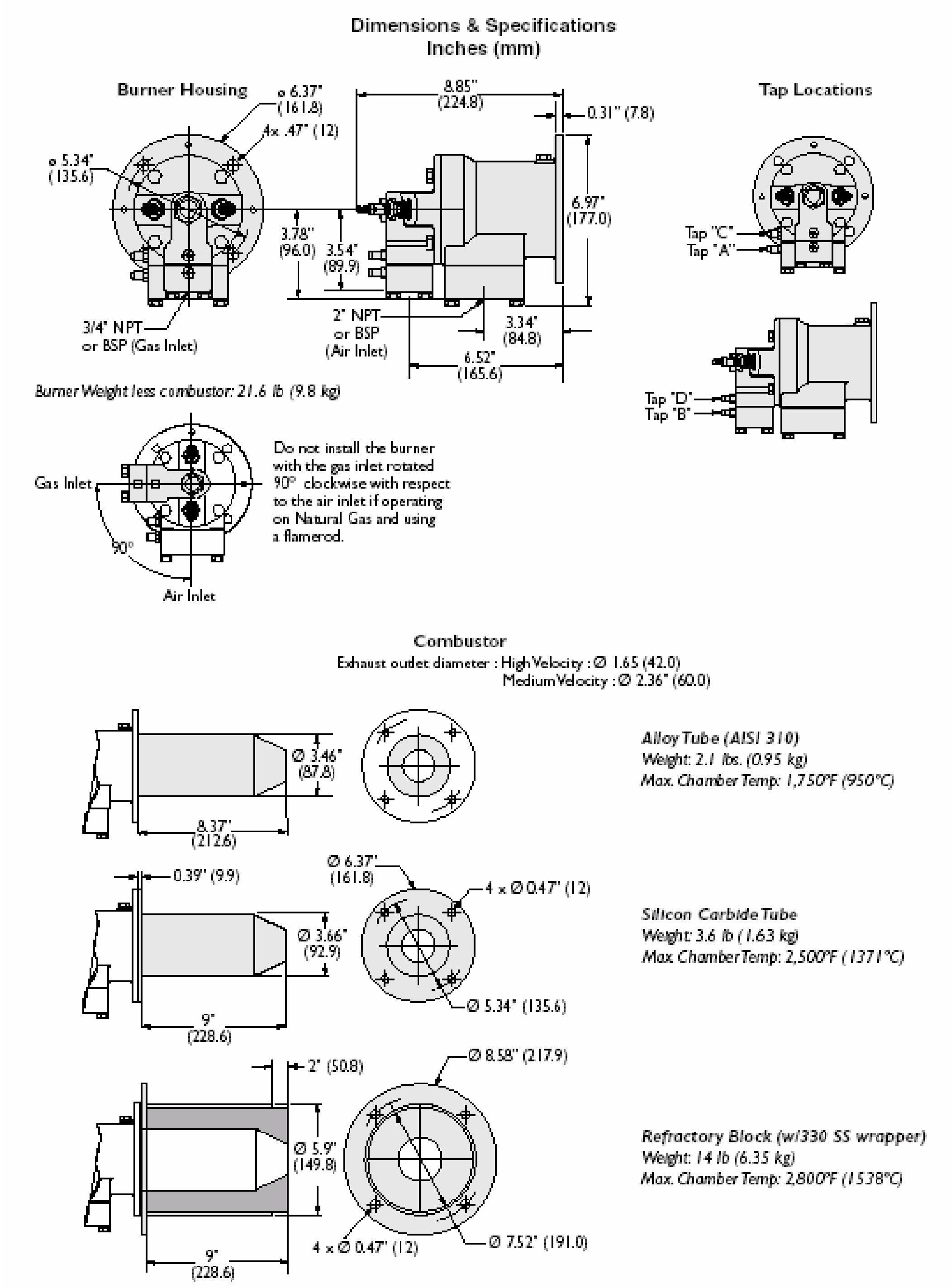
- fuel type
- combustion air temperature
- firing rate
- chamber conditions
- percent of excess air

For estimates of other emissions, contact Eclipse Combustion.





Burner Data Page 2



Burner Data Page 3

Appendix D

Referred Values

APPENDIX E Weights of Furnace Elements

MATERIAL	DESCRIPTION	SPECIFIC HEAT J/KG.K	DENSITY KG/M^3	THERMAL CONDUCTIVITY W/M.K
MAT_A	STEEL	452	7880	43
MAT_B	INSULATION TYPE 1	1070	85	0.18
MAT_C	INSULATION TYPE 2	1070	100	0.18
MAT_D	SAND	830	1600	0.35
MAT_E	BITUMEN	N/A	1000	0.17
MAT_F	SOIL*	N/A	1400	2

- NOTE THAT SOIL PROPERTIES ARE HIGHLY VARIABLE, THESE ARE A GUIDE ONLY.

Appendix E

Weights of Furnace Elements

APPENDIX E WEIGHTS OF FURNACE ELEMENTS

Side Panels			
Element	Quantity (qty)	Mass per qty	Total mass (kg)
Steel framing 75 x 75 x 5mm SHS	12.9 m	10.3 kg/m	132
Steel Plate 3mm	7.17 m^2	25.55 kg/m^2	183
Weld material (estimated)	0.0005m^3	7750 kg/m^3	3.9
Insulation Type 1	0.16 m^3	64 kg/m^3	10.2
Insulation Type 2	0.16 m^3	100 kg/m^3	16
Pipework Type 1 (Gas)	4 m	2.5 kg/m	10
Pipework Type 2 (Air)	4 m	5.4 kg/m	21.6
Wiring	13 m	0.25 kg/m	3.2
Burners and Mountings	2 off	12 kg	24
Gas trains and Associated Parts	2 off	8.5 kg	17
Insulating Pins	0.000405	7750 kg/m^3	3
Pipe Brackets	3 off	0.5 kg	1.5
Side Panel Weight Summary			425.4 kg
End Panel with Fan			
Element	Quantity (qty)	Mass per qty	Total mass (kg)
Steel framing 75 x 75 x 5mm SHS	10.8 m	10.3 kg/m	111
Steel Plate 3mm	5.37 m^2	25.55 kg/m^2	137
Weld material (estimated)	0.0004m^3	7750 kg/m^3	3.1
Insulation Type 1	0.081 m^3	64 kg/m^3	5.2
Insulation Type 2	0.081 m^3	100 kg/m^3	8.1
Pipework Type 1 (Gas)	3 m	2.5 kg/m	7.5
Pipework Type 2 (Air)	4 m	5.4 kg/m	21.6
Wiring	4.5 m	0.25 kg/m	1.125
Burners and Mountings	0 off	0 kg	0
Gas trains and Associated Parts	0 off	0 kg	0
Insulating Pins	0.0003	7750 kg/m^3	2.3
Pipe Brackets	4 off	0.5 kg	2
Fan	1 off	85 kg	85
Fan Mounting	1 off	23 kg	23
End Panel with Fan Weight Summary			406.9 kg

APPENDIX E WEIGHTS OF FURNACE ELEMENTS

End Panel without fan			
Element	Quantity (qty)	Mass per qty	Total mass (kg)
Steel framing 75 x 75 x 5mm SHS	10.78 m	10.3 kg/m	111
Steel Plate 3mm	5.37 m^2	25.55 kg/m^2	137
Weld material (estimated)	0.0004m^3	7750 kg/m^3	3.1
Insulation Type 1	0.081 m^3	64 kg/m^3	5.2
Insulation Type 2	0.081 m^3	100 kg/m^3	8.1
Insulating Pins	0.0003	7750 kg/m^3	3
End Panel without Fan Weight Summary			266.7 kg
Roof			
Element	Quantity (qty)	Mass per qty	Total mass (kg)
Steel framing 75 x 75 x 5mm SHS	12.9 m	10.3 kg/m	133
Steel Plate 3mm	7.17 m^2	25.55 kg/m^2	183
Weld material (estimated)	0.0005m^3	7750 kg/m^3	3.9
Insulation Type 1	0.15 m^3	64 kg/m^3	9.6
Insulation Type 2	0.15 m^3	100 kg/m^3	15
Insulating Pins	0.00035	7750 kg/m^3	2.7
Purlins	12	1.78 kg	21.4
Sheeting and Fixtures	12	3.26 kg	39
Roof Summary			407.6

Total mass of Furnace

Side Panels	**852.8**
End Panel with Fan	**406.9**
End Panel without Fan	**266.7**
Roof	**407.6**
TOTAL	**1934 KG**

Appendix F

Risk Assessment / Job Safety Analysis

RISK ASSESSMENT		
PROJECT SPECIFICATION SECTION	BRIEF DESCRIPTION	**RISK TYPE**
PHASE 2 SECTION 1	**RESEARCH FURNACE TECHNOLOGY**	**MINIMAL NO DISCUSSION REQUIRED**
PHASE 2 SECTION 2	**ESTABLISH FURNACE DESIGN PARAMETERS**	**MINIMAL NO DISCUSSION REQUIRED**
PHASE 2 SECTION 3	**DESIGN FURNACE HOUSING AND HEATING SYSTEM**	**MINIMAL NO DISCUSSION REQUIRED**
PHASE 2 SECTION 4	**ANALYSIS OF HOUSING AND SYSTEM WITH THERMAL ANALYSIS SOFTWARE**	**MINIMAL NO DISCUSSION REQUIRED**
PHASE 2 SECTION 5	**CONSTRUCT PROTOTYPE AND TEST**	**SEPARATE JOB SAFETY ANALYSIS REQUIRED (SEE NEXT PAGE)**
PHASE 2 SECTION 6	**EXTRAPOLATE RESULTS**	**MINIMAL NO DISCUSSION REQUIRED**
PHASE 2 SECTION 7	**PREPARE DRAFT DISSERTATION**	**MINIMAL NO DISCUSSION REQUIRED**
PHASE 2 SECTION 8	**PREPARE FINAL FORM**	**MINIMAL NO DISCUSSION REQUIRED**
PHASE 2 SECTION 9	**PREPARE LIFE CYCLE ASSESSMENT**	**MINIMAL NO DISCUSSION REQUIRED**
PHASE 2 SECTION 10	**ASSESS FULLY SIZED FURNACE FOR PRACTICAL PERFORMANCE**	**AS PER SECTION 5**

JOB SAFETY ANALYSIS- FURNACE MANUFACTURE AND OPERATION					
ACTIVITY	**RISK TYPE**	**APPLIES TO (PERSONNEL NUMBERS)**	**ELIMINATION**	**PPE**	**OTHER**
RECEIVE MATERIALS • **STEEL PLATE** • **STEEL SHS** • **REFRACTORY BLANKET** • **REFRACTORY PINS** • **BOLTS**	• MANUAL HANDLING INJURIES • **CRUSHING INJURIES** • **CUTTING INJURIES**	**INDIVIDUAL** **GROUP (IF OTHERS WORKING NEARBY)**	• **HANDLE WITH ASSISTING LIFTING WHEREVER POSSIBLE**	• **GLOVES** • **SAFETY BOOTS**	**N/A**
PROCESS MATERIALS • **CUT STEEL SECTIONS** • **CUT REFRACTORY** • **WELD SECTIONS** • **FIX REFRACTORY TO FRAME**	• MANUAL HANDLING INJURIES • **CRUSHING INJURIES** • **CUTTING INJURIES** • **IMPACT INJURIES** • **BURN INJURIES** • **INHALATION OF FIBRES** • **ARC FLASH INJURIES** • **EXPLOSION RISKS** • **ELECTROCUTION**	**INDIVIDUAL AND GROUP**	• **HANDLE WITH ASSISTED LIFTING WHEREVER POSSIBLE** • **REMOVE ALL FLAMMABLE MATERIALS FROM WORK AREA** • **ENSURE CUTTING DISCS ARE IN GOOD CONDITION** • **DON'T WELD IN WET CONDITIONS** • **ENSURE WELDING LEADS IN GOOD CONDITION**	• **SAFETY GOGGLES** • **BOOTS** • **HEARING PROTECTION** • **RESPIRATORY PROTECTION** • **WELDING SHEILDS AND GLOVES**	**N/A**

JOB SAFETY ANALYSIS- FURNACE MANUFACTURE AND OPERATION

ACTIVITY	RISK TYPE	APPLIES TO (PERSONNEL NUMBERS)	ELIMINATION	PPE	OTHER
CONSTRUCT FURNACE PANELS AND FIT BURNERS TO ORIFICE (S)	• MANUAL HANDLING INJURIES • CRUSHING INJURIES • INHALATION INJURIES	**INDIVIDUAL**	• **USE ASSISTED LIFTING DEVICES**	• **BOOTS** • **GLOVES** • **MASKS**	**N/A**
GAS BOTTLE HANDLING	• MANUAL HANDLING INJURIES • EXPLOSION RISKS	**INDIVIDUAL GROUP**	• **USE ASSISTED LIFTING DEVICES** • **ENSURE BOTTLES ARE KEPT IN CRADLES WITH VALVE PROTECTION**	• **GLOVES**	**N/A**
GAS FITTING	• EXPLOSION RISKS • MANUAL HANDLING INJURIES	**INDIVIDUAL GROUP**	• **ALL FITTING BY LICENSED GAS FITTER** • **USE ASSISTED LIFTING DEVICES**	• **GLOVES** • **SAFETY GLASSES** • **BOOTS**	**N/A**
ELECTRICAL FITTING	• ELECTROCUTION	**INDIVIDUAL GROUP**	• **ALL ELECTRICAL WORK**	• **GLOVES** • **SAFETY GLASSES** • **BOOTS**	**N/A**
LOADING/UNLOADING OF FURNACE	• MANUAL HANDLING INJURIES • CRUSHING INJURIES • BURNS	**INDIVIDUAL**	• **USE ASSISTED LIFTING DEVICES** • **MEASURE SURFACE TEMPERATURES WITH LONG TEMP PROBE PRIOR TO TOUCHING OR HANDLING**	• **GLOVES** • **BOOTS** • **SAFETY GLASSES**	**N/A**