DESIGN AND EVALUATION THROUGH SIMULATION AND EXPERIMENTAL APPARATUS OF A SMALL SCALE WASTE HEAT RECOVERY SYSTEM

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DECLARATION

I, the undersigned, hereby declare that the work contained in this assignment is my own original work and that I have not previously, in its entirety or in part, submitted it at any university for a degree.

ABSTRACT

Realisation of the depletable nature of fossil fuel has increased the need for its optimal use. Increasing global pressure to reduce the emission of greenhouse gases and other harmful gases that affect the chemical cycles or destroy the greenhouse gases in the tropospheric ozone, has attracted a increased worldwide concern. Waste heat recovery devices have been around for more than 50 years and researches and scientists have been very much involved in identifying the correct type of systems to meet the requirements of industries and mankind more efficiently. Waste heat can be identified in the form of unburned but combustible fuel, sensible heat discharges in drain water, and latent and sensible heat discharge in exhaust gases.

In this project the feasibility of a small scale waste heat recovery system has been investigated. Sets of preliminary investigations were performed to evaluate the amount of waste heat that can be extracted from the exhaust gases of typical diesel powered truck engines. A waste heat recovery unit was designed, implemented and evaluated through simulation and experimental investigations.

Preliminary calculations were performed using the readings presented by Koorts (1998) for a typical 6-litre diesel engine. The calculations showed that it is possible to extract about 77kW of waste heat from the exhaust gases from such an engine. A simple Rankine cycle was then investigated to be operated on the waste heat recovered. The optimal parameters for such a Rankine cycle was determined using a spreadsheet program and was found to be an optimal pressure of 800kPa with a temperature of 227.2°C and a water mass flow rate of 0.0015kg/s as the working fluid. For such a Rankine cycle, based on the efficiencies of commercially available pumps, turbines and heat exchangers it was found that it is possible to extract 2782kW of power per unit mass flow rate of water.

The next stage of the project was designing and implementing an exhaust gas pipe network from the engine test cells at the Centre for Automotive Engineering (CAE) located on the ground floor to the Energy Systems Laboratory (ESL) at the first floor. This pipe network was equipped with a valve system that can be operated from the ESL and allows the selection of the route of the exhaust gases and two bellows to

compensate for thermal expansion. A continuous combustion unit was also linked to the exhaust gas supply pipes as an alternative source of exhaust gases. The waste heat exchanger designed and selected was purchased and linked into the exhaust gas stream after calibration tests were carried out on the same in the wind tunnel. The water supply and a steam separator were then connected to the waste heat exchanger.

In the final experimental stage of the project, two sets of tests were carried out. The first set of tests was performed using exhaust gases from the continuous combustion unit and the second using exhaust gases from the internal combustion engines in CAE.

Superheated steam was obtained in both cases indicating the possibility of operating a turbine with the dry steam generated. With exhaust gases originating from the continuous combustion unit, an air fuel ratio of 9.14:1 was used and exhaust gases at a temperature of 540°C were obtained with an air inflow of 1400kg/h and a fuel consumption rate of 7.11 kg/h. The exhaust gases degraded to 360°C at the waste heat recovery inlet due to losses through the bare pipes. 11.12kW of energy was extracted from the exhaust gases to the water stream with an efficiency of 98%. With the exhaust gases from the 10-litre diesel internal combustion engine, an exhaust gas flow rate of 0.22kg/s was used and with a heat transfer efficiency of 89%, 18.5kW of power was extracted at the waste heat recovery unit. This represents a 4.9% of the thermal content of the fuel used. A rate of energy production balance on the internal combustion engine showed that 34% is lost in exhaust gases and 29% in coolant and other losses while only 37% is used produced as shaft power.

The results obtained therefore show that there is ample room for further investigation for the use of waste heat in exhaust gases of typical diesel engines.

It can therefore be concluded that the aims of the project that were to set up a testing facility and an exhaust gas pipe network and evaluation of a small scale waste heat recovery apparatus were achieved.

The tests performed can still be optimised with more waste heat removal from the exhaust gases of typical diesel truck engines and hence better recovery of waste heat and a reduction of fuel consumption.

OPSOMMING

Met die besef van die kwynende beskikbaarheid van fosielbrandstof het die behoefte vir die optimale benutting van die brandstof toegeneem. Toenemende globale druk om die emissies van groenhuis gasse en ander gevaarlike gasse wat chemiese siklusse beïnvloed in die troposfeer te verminer, geniet wêreldwye aandag. Oorskotenergie-toestelle is alreeds beskikbaar die afgelope 50 jaar en navorsers en wetenskaplikes was tot op hede betrokke met die identifisering van die korrekte tipe sisteme om meer effektief aan die industrie en samelewing se behoeftes te voldoen. Oorskotenergie bestaan uit onder andere onverbrande maar brandbare brandstof, voelbare warmte in dreinwater, en latente en voelbare warmte in uitlaatgasse.

In hierdie projek word die lewensvatbaarheid van 'n kleinskaal oorskotenergie herwinningsisteem ondersoek. Voorlopige ondersoeke was gedoen om die hoeveelheid oorskotenergie te bepaal wat herwin kan word uit die uitlaatgasse van 'n tipiese 6 liter vragmotor dieselenjin. 'n oorskotenergie herwinningseenheid was ontwerp, geïmplimenteer en ge-evalueer deur similasies en eksperimentele ondersoeke.

Voorlopige berekeninge was uitgevoer op data wat deur Koorts (1998) saamgestel is vir 'n tipiese vragmotor dieselenjin. Die berekeninge toon dat dit moontlik is om ongeveer 77kW oorskotenergie van die uitlaatgasse van so enjin te onttrek. Die moontlikheid was toe ondersoek om die herwinne energie te gebruik om 'n eenvoudige Rankine siklus aan te dryf. Die optimale parameters vir die Rankine siklus was bereken deur van 'n sigblad program gebruik te maak en dit was gevind dat die optimale druk is 800kPa, die optimale temperatuur is 227.2°C teen 'n water massa vloeitempo van 0.0015kg/s. Vir so 'n Rankine siklus, gebaseer op die effektiwiteit van kommersiële beskikbare pompe, turbines en warmteruilers, was dit gevind dat dit moontlik is om 2782kW drywing per eenheidsmassa vloeitempo van water, te onttrek.

Die volgende stadium van die projek was die ontwerp en implimentering van 'n uitlaatgas pypnetwerk vanaf die toetsselle van die Centre for Automotive Engineering (CAE) op die grondvloer na die Energy Systems Laboratory (ESL) op die eerste vloer. Die pypnetwerk was toegerus gewees met 'n kleptstelsel wat vanaf ESL bedryf kan word en wat dit moontlik maak om die roete van die uitlaatgasse te beheer. Twee samedrukbare koppelstukke was ook ingesluit in die lang reguit pypseksie om vir termiese uitsetting te kompenseer. 'n Aaneenlopende verbrandingseenheid was ook gekoppel met die uitlaatgasse toevoerpype as 'n alternatiewe bron van uitlaatgasse. Die oorskotenergie warmteruiler wat ontwerp en geselekteer was, was aangekoop en opgekoppel met die uitlaatgas-stroom nadat kalibrasie toetse op die warmteruiler gedoen was in 'n windtonnel. Die watertoevoer en 'n stoomskeier was gekoppel aan die oorskotenergie warmteruiler.

Twee toetse was uitgevoer in die finale eksperimentele stadium van die projek. Die eerste stel toetse was uitgevoer deur gebruik te maak van die uitlaatgasse van die aaneenlopende verbrandingseenheid en met die tweede toets is van die uitlaatgasse van die interne verbrandingsenjins van CAE gebruik gemaak.

Oorverhitte stoom was verkry in beide gevalle en wys dus dat daar 'n moontlikheid is om 'n turbine met droë stoom aan te dryf. 'n Lug tot brandstof verhouding van 9.14 : 1 was gebruik gewees in die aaneenlopende verbrandingseenheid om uitlaatgasse te verskaf teen 540°C. Die massavloeitempo van die lug was 1400kg/h en die brandstof 7.11kg/h. Die uitlaatgasse se temperatuur het afgeneem tot 360°C tot voor die oorskotenergie herwinningseenheid as gevolg van hitteverliese vanaf die ongeïsoleerde pypnetwerk. 11.12kW energy was onttrek vanaf die uitlaatgasse en oorgedra aan die waterstroom met 'n effektiwiteit van 98%. Die 10 liter diesel interne verbrandingsenjin het uitlaatgas gelewer met 'n massa vloeitempo van 0.22kg/s. 18.5kW energie was herwin gewees met 'n effektiwiteit van 89%. Dit verteenwoording 4.9% van die termiese inhoud van die brandstof gebruik. 'n Energie balans op die interne verbrandingsenjin het getoon dat 34% energie gaan verlore in die uitlaatgasse, 29% word aan die verkoelingsmiddel oorgedra en 37% is bruikbare meganiese drywing.

Die resultate wat verkry is, wys daarop dat daar nog groot ruimte is vir verdere ondersoeke in die gebruik van oorskotenergie in uitlaatgasse van tipiese vragmotor dieselenjins.

Die gevolgtrekking kan dus gemaak word dat die doelwitte van die projek naamlik die opstel van 'n toetsfasiliteit, installering van 'n uitlaatgasse pypnetwerk en die toets van a kleinskaalse oorskotenergie herwinningseenheid, bereik was.

Die toetse wat uitgevoer was kan nog ge-optimeer word om meer energie te herwin vanaf die uitlaatgasse van 'n tipiese vragmotor dieselenjin om sodoende beter brandstofverbruik te bewerkstellig.

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TABLE OF CONTENTS

Content	***Page***
DECLARATION	i
ABSTRACT	ii
OPSOMMING	iv
ACKNOWLEDGEMENT	vii
TABLE OF CONTENTS	viii
LIST OF FIGURES	xi
LIST OF TABLES	xii
LIST OF SYMBOLS	xiii
1. INTRODUCTION	1
1.1 Problem Statement	3
1.1.1 Objectives of the project	3
1.1.2 Limitations and constraints	4
2. MOTIVATION	5
3. LITERATURE REVIEW	6
3.1 Cogeneration	6
3.2 Review of Published Papers	7
3.3Discussion	21
4. ENERGY AVAILABILITY	23
4.1 The Delivery Temperature of Exhaust Gases	23
4.2 The Lower Temperature Limit	23
4.2.1 Typical calculation of fuel-air ratio	24
4.2.2 Calculation of minimum temperature	25
4.3 Calculation of the Amount of Useful Energy Available	26
4.4 Conversion of Recovered Waste Heat	26
5. THE WASTE HEAT RECOVERY APPARATUS	29
5.1 Pipe Network	29
5.1.1 The initial proposal	30
5.1.2 Drawbacks and modifications of the initially proposed pipe network	32

5.2 The Waste Heat Exchanger 34
5.2.1 The three stages waste heat recovery steam generator 35
5.2.2 Identification of the drawbacks of the three stages waste heat recovery steam generator 35
5.3 The Continuous Combustion Unit 36
5.4 Water Connection to Waste Heat Exchanger 40
5.5 The Steam Separator 41

6. THE WASTE HEAT EXCHANGER 43
6.1 Specification and Preliminary Calculations 43
6.1.1 Specifications of the finned heat exchanger 43
6.1.2 Preliminary calculations 44
6.2 Preliminary Tests on Waste Heat Exchanger 53
6.2.1 Test description 53
6.2.2 Results from preliminary tests on waste heat exchanger 54

7. CONTINUOUS COMBUSTION UNIT 57
7.1 Procedure for Obtaining the Required Exhaust Gas Temperature 57
7.2 Settings of the Continuous Combustion Unit to Produce Exhaust Gases for Desired Operating Conditions 58
7.3 Operating Conditions at Varying Air-Fuel Ratios 59

8. THE INTERNAL COMBUSTION ENGINE 61
8.1 The Internal Combustion Engine 61
8.2 Readings for the C-10 Engine 62
8.3 Determining the Power Output and the Mass Flow Rate of Air 63
8.4 Power Balance on Engine 63

9. PRESENTATION AND DISCUSSION OF RESULTS 65
9.1 Presentation of Results 65
9.1.1 Exhaust gases from the continuous combustion unit 65
9.1.2 Exhaust gases from the internal combustion engine 67
9.2 Analysis and Discussion of Results 67
9.2.1 Exhaust gases from continuous combustion unit 67

9.2.2. Exhaust gases from internal combustion engine 68

9.3 Comments 70

10. CONCLUSIONS 72

11. RECOMMENDATIONS 74

12. LIST OF REFERENCES 76

APPENDICES

APPENDIX A ENGINE PARAMETERS FOR THE 6 LITRE AND 10 LITRE DIESEL ENGINES

APPENDIX B GRAPHS AND COMMON EQUATIONS

APPENDIX C CALCULATIONS OF PARAMETERS FOR RANKINE CYLE AT VARYING PRESSURES

APPENDIX D WASTE HEAT EXCHANGER CALCULATIONS

APPENDIX E EXHAUST GAS PROPERTIES

APPENDIX F THERMAL EXPANSION OF EXHAUST PIPE NETWORK

APPENDIX G PRESSURE DROP APPROXIMATION IN THE PIPE NETWORK

APPENDIX H PIPE INSULATION THICKNESS

APPENDIX I QUICK REFERENCE GUIDE FOR CONTINUOUS COMBUSTION UNIT OPERATION

APPENDIX J DRAWINGS

LIST OF FIGURES

Figure 2.1 Distribution of fuel energy for a typical diesel engine

Figure 4.1 Net power output from the Rankine cycle at different pressures

Figure 4.2 Temperature difference at pinch point for different pressures

Figure 4.3 Cycle efficiency at different pressures

Figure 5.1 Pipe network below floor level in ESL

Figure 5.2 Pipe network above floor level in ESL

Figure 5.3 Modification to the pipe arrangement below floor in ESL

Figure 5.4 Piping system below floor level (for two exhaust gas lines)

Figure 5.5 Schematic diagram of the continuous combustion chamber and all its components

Figure 5.6 Exhaust gas pipe link at the outlet of continuous combustion unit to the waste heat recovery system

Figure 5.7 Water stream connection to waste heat exchanger

Figure 5.8 Steam separator

Figure 7.1 Temperature of the exhaust gases for different air-fuel ratios

Figure 8.1 Power distribution for the Caterpillar C-10 diesel truck engine

Figure 9.1 Power distribution for the Caterpillar truck engine after waste heat recovery

LIST OF TABLES

Table 6.1 Heat exchanger specifications and details

Table 6.2 Preheater fluid conditions

Table 6.3 Properties of air at average temperature in preheater section

Table 6.4 Properties of water at average temperature in preheater section

Table 7.1 A typical setting for required exit conditions

Table 8.1 Caterpillar C-10 diesel truck engine specifications

Table 8.2 Operating conditions on Caterpillar C-10 diesel truck engine

Table 9.1 Settings and readings on the continuous combustion unit

Table 9.2 Readings at the waste heat recovery apparatus (exhaust gases from continuous combustion unit)

Table 9.3 Readings at the waste heat recovery apparatus (Exhaust gases from the IC engine)

LIST OF SYMBOLS

Nomenclature

A	Area
a	Constant
b	Constant
c	Specific heat capacity
C	Heat capacity
D	Nominal diameter
e	Efficiency
eff	Effectiveness
F	Variable
f, f_D	Friction factor
G	Mass flux density
h	Heat transfer coefficient, enthalpy
ID	Inner diameter
k	Thermal conductivity
K	Constant
$\dot{m}$	Mass flow rate
N	Rotational speed
OD	Outer diameter
P	Pitch
Pr	Prandlt number
Q	Power, Heat energy
Re	Reynolds number
St	Stranton number
t	Tube, thickness
T	Torque
x	Quality
w	Humidity ratio

Greek symbols

α	Constant
β	Constant
γ	Constant
δ	Constant
ε	Constant
μ	Viscosity
ν	Number of moles, specific volume, constant
ρ	Density
σ	Area ratio
ϕ	Relative humidity
χ	Variable

Subscripts

a, A	Air, airside
av	Average
B	Boiling
c	Critical
cv	Control volume
ex, exh	Exhaust gas
f	Fin
FC	Forced convection
fr	Frontal
g	Gas
hyd	Hydrodynamic
i	Inner
l	Longitudinal
o	Outer
out	Exit
p	Constant pressure
s	Stoichiometric
sat	Saturated
t	Transverse
trans	Transferred
w	Water

1. INTRODUCTION

Non-renewable energy sources, which include natural gas, petroleum and coal, are the primary resources for all the technological activities of the present day mankind. The economic growth of every society is related to the per capita consumption of non-renewable energy sources. Three quarters of mankind's carbon dioxide emissions is due to non-renewable energy use. At the same time other gases are produced which are not greenhouse gases but influence the chemical cycles in the atmosphere that produce or destroy greenhouse gases such as the tropospheric ozone (INSREC, 1998). The use of oil expanded after the post World War II economic takeoff (INSREC, 1998) but now the realisation of the depletable nature of the fossil fuel which is still the backbone of nearly all the economy in the world and the effect of global warming due to increasing air pollution has attracted worldwide concern.

While it is not easy to find an immediate substitute for the non renewable energy sources, it is crucial that these fuels be utilised optimally and with the least possible harm to the environment.

Waste heat recovery devices have been around for more than fifty years and scientists and researchers have been very much involved in identifying the correct type of systems to meet the requirements of industries and mankind more efficiently. Recovery of waste heat has been more intensively practiced in power generation and energy intensive industries.

Waste energy can be identified in the form of unburned but combustible fuel, sensible heat discharged in drain water, and sensible and latent heat discharge from exhaust gases (Najjar et al., 1993).

Recovery of waste heat offers several benefits and the following can be quoted as the main ones:

1. The energy that would otherwise have been thrown away is now used to produce electricity, mechanical work, refrigeration or converted into some other form of useful energy.

2. Reduction of waste heat energy also reduces the cost involved in the disposal equipments required; for example a smaller cooling tower will be required, if any. Less feed water will be required and maintenance costs involved is reduced.
3. Environmental pollution is reduced particularly in terms of the amount of heat released into the atmosphere.
4. The overall amount of energy extracted from the primary fuel source is higher and hence a greater overall efficiency for the engine is achieved.

In this project, the implementation of a small scale waste heat recovery unit is investigated. More specifically the generation of superheated steam that can be used to operate a Rankine cycle and hence the production of electrical power from the exhaust gases of a diesel truck engine is investigated.

In the next two chapters the motivation behind this project and a literature review of published papers are presented.

The estimation of the amount of thermal energy available for extraction in a typical diesel powered truck engine is performed in chapter 4. This includes preliminary calculations performed using the results and readings obtained from the work of Koorts (1998). The sample calculations carried out give an indication of the amount of energy available and the amount of useful energy that can be extracted. The constraints imposed on the implementation of a waste heat recovery system are identified and the optimum values of the concerned parameters are calculated.

In chapter 5, the waste heat recovery system that was design and implemented is described. This includes a description of the pipe network, the waste heat exchanger, the continuous combustion unit, the water connection to the waste heat exchanger, and the steam separator. Schematic diagrams are also presented to give clearer indication of the waste heat recovery unit that was implemented during the course of this project.

In chapter 6, the heat exchanger used for waste heat extraction from the exhaust gases is presented. The preliminary calculations and tests performed on the selected waste heat exchanger are also presented.

Results obtained from the waste heat exchanger and the calibration of the continuous combustion unit which is also used as an alternative source for exhaust gases are presented in chapter 7.

The continuous combustion unit used as the auxiliary source of exhaust gases for the waste heat recovery unit is presented in chapter 7.

The internal combustion unit for experimentation on the waste heat recovery unit is presented in chapter 8. An energy balance on the internal combustion engine is also presented.

In chapter 9, the results obtained during experimental evaluation of the waste heat recovery unit are presented and discussed.

In chapter 10, the conclusions are made based on the results and the findings of the project.

Finally, based on the conclusion and the findings of the project recommendations are made in chapter 11.

1.1 Problem Statement

1.1.1 Objective of the project

The objective of this project is to design and evaluate through simulation and experimental apparatus a small scale waste heat recovery system. This includes carrying out a study of the utilisation of waste heat available in the exhaust gases emitted from typical diesel truck engines, implementation of a waste heat recovery unit, and generation of superheated steam that can be used to operate the turbine of a simple Rankine cycle. Ways of improving the waste heat recovery and utilisation will also be identified and any possible modification or shall be made.

1.1.2. Limitations and constraints

During the duration of the project most time was consumed in performing preliminary calculations, analysing the heat available and designing and setting up the waste heat recovery system apparatus. Not much time was available for intensive modification after the apparatus has been set up.

Constraints on the project implementation are as briefly enumerated below.

1. The system should be cost effective.
2. The availability of components for such a small scale level waste heat recovery system.
3. Back pressure imposed in the engine exhaust will affect the efficiency of the main internal combustion engine.
4. Time available for completion of the project.

2. MOTIVATION

Twenty five percent of the energy consumption in the western world is centered in the transport sector. Automobiles and trucks account for approximately 80 % of all transportation energy expenditures. These thermal engines operate with a thermal efficiency of about 40%, Koehler et al. (1997). According to Corrado (Joubert, 1996), nearly a third of the power generated is thrown away in the exhaust gases and about 25% in cooling oil and water. Figure 2.1 below gives an indication of the distribution of fuel energy in such diesel engines.

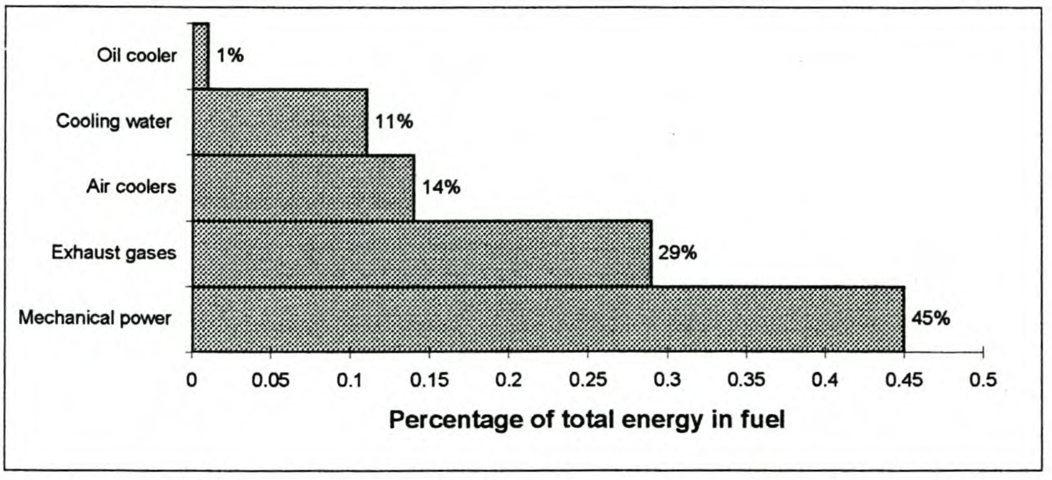

Figure 2.1 : Distribution of fuel energy for a typical diesel engine

It can be seen that considerable amount of energy is lost in the exhaust gases and it was found in literature, to be presented next, that much work has been done to recapture waste heat in the exhaust gases. The results are very promising and give a very good incentive to carry out further research to utilize the waste heat in the exhaust gases and hence improve the efficiency of utilization of the fuel. One effective way of utilizing otherwise wasted heat is to produce electricity. From the statistics in figure 2.1, it can be seen that there is indeed enough practical and economical evidence to investigate and set up a waste heat recovery plant.

3. LITERATURE REVIEW

As mentioned earlier a lot of effort is being made by scientists and researchers to extract the maximum amount of energy possible from the limited sources of fossil fuels.

In this section an overview of articles found to date in literatures is presented. Numerous publications were found in literature on implementations and investigations carried out to date on a variety of waste heat recovery systems. Before proceeding further, the concept of cogeneration, which is used in most of the literature shall be explained.

3.1 Cogeneration

Cogeneration is the simultaneous production of electrical energy and heat often in the form of process steam. This description will apply in applications where waste heat is extracted using water as the working fluid. This technology is not new and it can be traced back to World War II years (Rabghi et al., 1993).

The main component of a cogeneration system is an engine, steam turbine or combustion turbine that drives an electrical generator. A waste heat exchanger recovers waste heat from the exhaust gases and transfers it to the process fluid stream. The process fluid in most cases is water. Two main types of cogeneration systems can be distinguished: 'The Topping Cycle' and ' The Bottoming cycle'.
In a 'Topping cycle', electrical energy is generated first. Fuel is burnt to produce power and the heat in the exhaust gases are used to generate steam which is drives a secondary steam turbine. A 'topping cycle' always requires additional fuel beyond what is required for manufacturing, so there is an additional operating cost associated with the power generation (Hesse ,1999).

'Bottoming cycle' plants are much less common than 'Topping cycle' plants. In the former plants a waste heat recovery system recaptures waste heat from a manufacturing heating process. This waste heat is then used to produce steam that drives a steam turbine to produce electricity. Since fuel is burnt first in the production process, no extra fuel is required to produce electricity. Applications of 'Bottoming cycle' plants are more common in heavy industries such as glass or metals manufacturing where very high temperature furnaces are used (Hesse, 1999).

3.2 Review of Published Papers

Pasquinelli (1982) presents a gas fired open cycle vapour compression industrial heat pump being developed by Thermo Electron Corporation. The heat pump consists of a rotary screw compressor, 37.3kW Caterpillar gas fuelled internal combustion engine, gearbox coupler and waste heat boiler. The waste heat boiler recovers heat from the gas engine's cooling system and produces additional steam, which is added to that, produced by the compressor. The prototype was designed with a nominal flow rate of 4535kg/h. Steam was upgraded from 2.07bar to 6.21bar. The author claims that possible fuel savings of 50 to 70 percent over conventional fossil fuelled boilers can be achieved with a payback period of one to two years.

Hall et al. (1982) examine the principles of design, operating case studies, economics and potential for heat recovery boilers especially those for dusty and difficult gases. The authors identify four types of applications for which waste heat boilers can be used, namely,

1. Chemical and oil refinery process heat recovery boilers, where often products of intermediate reactions have to be cooled down before further process.
2. Non-ferrous and glass heat recovery boilers, where the reactions employed result in heated end products.
3. Heat recovery with incinerators where disposal of industrial or domestic, liquid or solid combustible waste results in hot gases reaching up to 1100°C.
4. Diesel engine and gas turbine exhaust recovery boilers where the exhaust gas temperature is in the range of 250°C to 550°C. The authors claim that the overall thermal efficiency of gas turbines or diesel engines can be raised from 27 to over 40 percent in situations where waste heat is used to raise steam which is then supplied to turbo-generators for electrical power generation.

As one of the case studies, the authors cite the *Greater London Council Edmonton* incinerators where modifications were made on existing boilers. These modifications allowed Edmonton to settle down to reliable operation over the next five years achieving throughputs of refuse of 400 000 tonnes per year and an electricity sales valued at $5.25 million.

Shlyk et al. (1984) investigated the utilisation of heat in exhaust gases from heat treated and pre-heat furnaces at Energomasshpetssal Works in Kramtorsk (USSR). Their investigation consisted of utilising heat in the 336,000m^3/h of exhaust gases at a temperature of 290°C that was released into the atmosphere. The waste heat process consisted of two stages. The first one comprises of cast iron economisers where the exhaust temperature drops from 290°C to 130°C. The second stage comprises of bimetallic heaters and spiral fins. The exhaust gases leave this stage at 80°C. The authors claim that utilisation of the waste heat would result in a saving of 238,260GJ of thermal energy per year.

Devotta et al. (1985) compares the theoretical Rankine power cycle performance of 24 working fluids has been.. The authors explain that the boiling temperature T_{bo} is largely dependent on the temperature of the available heat supply and the gross temperature drop (T_{bo}-T_{co}) is determined by the temperature of the coolant in the condenser. Therefore the only way of varying the theoretical Rankine power cycle efficiency and the pressure ratio is to choose another working fluid. Of the 24 working cycles chosen only 18 of them were found to have a critical temperature high enough to be considered for boiler temperature above 120°C and only 7 for boiler temperature above 200°C. The authors list values of the theoretical Rankine power cycle efficiency for gross temperature drop varying from 20 to 70 (steps of 10) for boiler temperature of 80,100,120,160 and 200°C.

In his paper, Cemenska (1988) describes the design and initial test results of a project to demonstrate a diesel engine- Rankine Bottoming Cycle power plant. The engine used was a 16 cylinders, heavy duty Caterpillar 3500 series engine. The author claims 10.4 percent fuel savings and a thermal efficiency of 44.5 percent.

Rosenblatt (1992) presents the Novel Combine-Cycle Low-Temperature Engine System (LTES). The LTES cycle involves combining a refrigeration cycle and a power turbine cycle. The author reviews the thermodynamic relationships between the cycles and presents an analytical method that permits an increase in the magnitude of potential-power to be expected. The author claims net-power increases of 6% for coal fired plants, 12% for nuclear plants and 22% for geothermal plants.

Rabghi et al. (1993) present a review of waste heat recovery and utilisation. The potential for reusing the otherwise wasted heat in different branches of industries is discussed by the authors. They also propose the recovery of waste energy by installation of combustion equipment in the case of wasted unburned fuel in exhaust gases and hence leading to the recovery of sensible and latent heat. The economical features involved in deciding whether to apply a waste heat recovery system or not are also mentioned. The authors point out that research and development efforts seem to be focused especially on heat exchangers that utilise heat pipes, Rankine cycles and heat pumps.

Najjar et al (1993) reviewed waste heat recovery systems involving cogeneration with gas turbine engines. The authors also present an energy map of process plants which indicate the temperature ranges associated with the various equipments used in industries. The authors also point out the use of cogeneration in areas other than district heating and air-conditioning.

Bringmann (1993) considers the use of exhaust heat in gas engine driven heat pumps. The author outlines the heat gain that can be achieved with the aid of heat pumps. He gives the example of the *Jacobs Sports Centre* in Constance (Germany) where a coefficient of performance of 2.5 was measured i.e. the useful thermal energy was 2.5 times the quantity of the primary energy expended. Another example given is the water/water heater pump operating at a heating estate where 505.5kW of the 800.4kW heating capacity is obtained from the heat pump condenser (heat source).

Tessier (1993) presents the idea of combined cycles associating gas turbines, boilers to recover waste heat at the exhaust, and steam turbines using steam produced. The 900MW electricity power plant ordered by the Malaysian Electricity Board is used to illustrate the evolution of the above-mentioned idea. The characteristics of this plant are as follows: The 900MW of electricity is generated in three combined cycles of 300MW each. Each cycle comprises of two 100MW gas turbines and one 100MW steam turbine fed by two waste heat boilers which each produce 154221kg/h of high pressure steam and 36287kg/h of low pressure steam. All the high-pressure steam and some of the low-pressure steam are fed to the turbine while the rest of the low-pressure steam is used to heat the feedwater tank.

Oomori and Ogino (1993) investigated a waste heat recovery system for a passenger car using a combination of Rankine bottoming cycle and evaporative engine cooling system. The waste heat energy source was the heat from the engine cooling. The working fluid used was HFC123. The authors point out that at a running condition of 100km/h the exhaust gases is the highest among the waste heat sources but in the case of passenger cars the temperature and calorific value of exhaust gases tend to fluctuate markedly because of frequent repetition of starting and stopping in urban areas. The calculations performed by the authors show a fuel economy rate of 4.5 to 7 percent for low running loads with a pressure ratio of 2 and 3 respectively. The experimental results they obtained however showed a fuel economy rate of 3 percent at 40km/h constant speed running. The authors attribute the lower fuel economy rate obtained in the experimental results to the lower efficiency of the expander and the greater pressure loss due to the complex path in the cylinder head as the conventional cooling water jacket was used without modification. The ambient temperature was 25°C and the authors claim that the energy recovered varies in line with ambient temperature.

Bolland et al. (1995) present a thermodynamical analysis of the air bottoming cycle and the result of a feasibility study for using the air bottoming cycle for gas turbine waste heat recovery and the power generation on oil/gas platform in the North Sea. The feasibility study was made for the exhaust gases from an LM2500Pe turbine. The optimal pressure ratio was found to be 8:1 for a recommended 2-shaft engine with 2 compressor intercoolers for the air bottoming cycle. They conclude that the Air

Bottoming cycle is an economical alternative to for power generation on both new and old oil platforms with demand for more power.

Koehler et al. (1997) designed, built and tested a prototype of an absorption system for truck refrigeration using heat from the exhaust gases. The recoverable energy was analysed for representative truck driving conditions for city traffic, mountain roads and flat roads. Results obtained showed a coefficient of performance (COP) ranging from 23 to 30%. The authors claim that the COP can be increased well over 30%. The adsorption system was run on a vapour-compression cycle using ammonia as the working fluid. The authors also concluded that the system is very promising for long distance driving on flat roads.

Darkwa et al. (1997) carried out an analytical evaluation of using inorganic oxides as the storage material in a thermochemical store in automobile engines. The implementation of such systems would according to the authors reduce the consumption of energy and also pollution . The heat energy present in the exhaust gases was identified as the source of potential energy input for regenerating the store.

Zhang et al. (1997) presented a numerical study of the dynamic performance of an adsorption cooling system for automobile waste heat recovery system. They developed a new lumped parameter for non-equilibrium model and used to investigate and optimise the waste heat cooling system. The effects of operating temperature and the overall heat transfer coefficient on the system performance were also investigated. They concluded that the specific cooling power (SCP) is more sensitive to parameter changes than the coefficient of performance of the cycle (COP) and recommend further research to be done on the SCP of the system.

Horuz (1999) presents an experimental investigation into the use of vapour absorption refrigeration (VAR) systems in road transport vehicles using the heat in the exhaust gases of the main propulsion unit as the energy source. The author also made a comparison of the performance of a VAR system fired by a natural gas and the same system driven by engine exhaust gases. A Ford 150 (Dover) 6 litre diesel internal combustion engine was used to provide the necessary waste gases. The author claims that sufficient waste heat can be recovered to obtain the rated cooling effect of approximately 10kW.

One large application area of cogeneration is district heating. Many colleges and cities, particularly in the United States, which have extensive district heating and cooling systems have cogeneration facilities. The University of Florida has a 42MW gas turbine cogeneration plant (Gator Power) built in partnership with the local utility (http//www.gatorpower.ufl.edu, 1999).

Salan El-din (1999) investigated the use of a heat pump to utilize the waste heat from a heat engine. In his first model he considered the irreversibilities in the heat exchanger only and in his second model he includes the irreversibilities in the heat pump and the heat engine. He concludes that the heat delivered is limited by the highest temperature and as the internal irreversibility of the heat engine increases the highest temperature becomes less. He also states that the temperature at which heat is delivered from the pump is always less then 1.125 of the absolute temperature of the heat pump. He finally concludes that as the internal reversibility of the heat engine increases, the work ratio increases significantly and the power used in driving the heat pump is more than 60 percent of the output power from the heat engines heat engine for all cases.

Heyen et al. (1999) investigated parallel proposals for upgrading existing power plant by comparing suitable advanced thermal cycles. The alternatives investigated are repowering of existing steam cycle by topping with a gas turbine or by using a partially oxidising reactor. The authors claim that the partial oxidation reactor offers a more moderate cost of implementation compared to the topping with a conventional gas turbine. They also claim that by combining the turbomachines specifically with built such operations, with optimal steam cycles, the efficiency can be raised beyond 60 percent.

Najjar (1999) compares the performance of the integrated gas and steam cycle with the combined cycle. The author used a program that takes evaluated the performances over a wide range of operating conditions. The main variables used were the compression ratio and the turbine inlet temperature. The author concludes that the performance results showed that, when the cooling air is disregarded the combined cycle produces 7 percent more power than the IGSC but the combined cycle has a 6 and 12 percent, better performance for the overall fuel consumption and the overall efficiency respectively.

Scott *et al.* (1999a, b) performed an experimental and theoretical study of an open multi-compartment adsorption heat transformer (MAD-transformer) for different steam temperatures. They developed and designed a MAD transformer in the part I (Scott et al. 1999a) of their work. The authors then present the investigation of the hydrodynamics and the heat transfer characteristics of the MAD transformer and they concluded that the mathematical model derived from the experimental results are capable of predicting steady state, transient and dynamic behaviour. In Part II of their work Scott et al. (1999b) modelled and simulated the process using the mathematical models derived from part I of their work and they claim that the computer model developed is capable of calculating the different profiles for temperature, concentration, flow rates and energy streams in all the lamellas and compartments of the MAD-transformer. They conclude that good agreement was obtained with the experimental results and that the transient and dynamic simulations showed that the steady state is reached in 40-60 minutes. They also mention that the part III of their work will a technical and economic feasibility of the MAD-transformer with an evaporation-continuous crystallization plant in a major Swedish sugar mill.

Najjar (2000) reviews ten of his research investigations, which he carried out with his associates during the past ten years. His review consisted of four main sections:

Fundamental analysis

This sections includes a quantitative analysis in three cases and these include the use of waste heat in gas turbines, the relative effect of pressure losses and the inefficiencies of turbomachines on the performance of the heat exchange gas turbine cycle and a comparison of performance of cogenerative systems using single or twin shaft gas turbine engines.

Regenerative cycles using steam

In this section Najjar (2000) reviews two cases. The first one is about enhancing gas turbine engine performance by means of evaporative regenerative cycle. The author analysed the performance of the evaporative regenerative cycle parametrically taking as main variables, the compressor ratio, the turbine inlet temperature, and the humidity ratio. He concludes that the evaporative regenerative cycle outperforms an equivalent regenerative cycle by about 57 and 13 percent in power and efficiency respectively. The second case analysed is an intercooled low pressure turbo steam-injection gas turbine with cogeneration. In this research the author compares a system of intercooled low pressure steam injected turbine with cogeneration to an intercooled cogenerated one. He claims that the former outperforms the latter by 21 and 16 percent in power input and overall efficiency respectively.

Gas turbines with hydrogen

The three papers reviewed by the author are "Hydrogen fuelled and cooled gas turbine engine", "Cryogenic gas turbine using hydrogen for waste heat recovery and regasification of LNG" and "The over-expansion gas turbine cycle using hydrogen".

Gas turbines and the refinery

In this section the author describes his investigations for saving energy in refineries by means of expanders in fluidised-bed catalytic cracking in the petroleum industry and the energy conservation in the refinery by utilising reformed fuel gas and furnace flue gases. In this section the researches were mainly for energy intensive industries. In the second analysis proposed in this subsection the author investigated the utilisation of a refineries reformer gas in the gas turbine, and furnaces flues gases together with the engines exhaust gases in a heat recovery steam generator. The author claims that his results show that the proposed system offers a 100 percent overall efficiency and $5.25 million annual saving for a 12MW gas turbine.

Najjar (2000) based on his research, points out that the worldwide concern about cost, environment and quick availability to meet continuous load growth will encourage the use of gas turbines in power systems. He also points out that the cogeneration systems he analysed have superior performance and are economically more feasible and the application of cogeneration will increase due to an escalating interest and efficient use of power. The author also found that the engines relative merits are utilized by implementation of cogeneration and the increase in thermal efficiency thus achieved even at part load makes it attractive to power and industry.

Bhatt (2000a,b) presents two energy audit cases. The first audit case, Bhatt (2000a) presents is related to steam systems and the author presents an analytical diagnostic tool for energy audit of steam systems. He concludes that while focus in most energy audit is on improving the boiler efficiency, most important losses occurs in the steam lines and the product of these segment efficiencies give the circuit efficiency. The author points out that, dividing the boiler efficiency into combustion efficiency, heat transfer efficiency, and material efficiency, better improvement can be made.

The second energy audit case presented by Bhatt (2000b) is related to air conditioning. The author discusses the methodology for the determination of each segment in a centralised air conditioning plant. He identifies major losses to occur in the chilled air ducts and cycling losses in the refrigeration circuit. The author also claims that the effective control and instrumentation of the air conditioning plant is the best and most cost effective method of energy conservation.

An automobile waste heat adsorption cooling system was designed and tested by Zhang (2000). The adsorption cooling system was driven by the waste heat of a diesel engine and the working fluid is Zeolite13x/water pair and a finned double tube heat exchanger was used as the adsorber. The author points out that the adsorption cooling is an excellent alternative because the supply of waste heat and the need for air conditioning both reach maximum level at the same time. Based on his experimental work, the author concludes that the coefficient of performance of the system is 0.38 and the specific cooling power during a cycle on the basis of the unit weight of adsorbent is 25.7W/kg. He also concludes that for a practical automobile waste heat adsorption cooling system, the demand for he coefficient of waste heat cooling (ratio of cooling producton to potential waste heat energy that can be recovered before due point corrosion) was satisfied, but the specific cooling power as far as the bulk and cost is concerned, further research is needed.

Borouis et al. (2000) studied the thermodynamic performance of a single stage adsorption/compression heat pump using a ternary working fluid for upgrading waste heat. The ternary fluid used was Trifluoroethanol-Water-Tetreethylenglycol dimethyl ether (TFE-H_2O-TEGDME). The authors developed a simulation program based on mass and energy balance on each component of the cycle. Their results show that use of the ternary fluid is more advantageous than the TFE-TEGDME binary working pair. Bourouis et al. (2000) claim that it is possible to upgrade thermal waste heat from 80°C to 120°C with a Coefficient of performance of about 6.4, a compression ratio of 4 and at a low pressure of 100kPa and a mole fraction of approximately 42 percent in the vapour.

Vasiliev et al. (2000) investigate the application of a latent heat storage module for preheating internal combustion engines to a bus petrol engine. The authors mathematically modelled a heat storage system for preheating the bus petrol engine before its ignition and then these models were experimentally investigated. The experimental values obtained correlate very well with those obtained from the mathematical model. The authors finally conclude that the mathematical models developed allow one to determine data and operational characteristics of heat storage. They finally conclude that the operational test have corroborated the validity of the developed engineering technique to calculate heat storage parameters.

Lijun et al. (2000) present a thermal load deviation model for superheater and reheater of a utility boiler. The thermal load deviation was determined and used for the prediction and prevention of boiler tube failures. The authors also point out that the temperature deviation is one of the root causes of boiler tube failures and are responsible for 40 percent of the forced power station outages. The authors applied the model to the reheater of a 300MW utility boiler and they claim that the temperature value taken in situ agree well with those obtained from the model. The authors also claim that the model can be applied to utility boilers of different manufacturers.

Pilavachi (2000) gives an overview of power generations with gas turbine and combined heat and power (CHP) systems. The author outlines the requirement of reducing the pollution level as per the Kyoto objectives for the European Member states, to decrease the greenhouse emissions by 8 percent in 2010 compared to the 1990 level. He also points out the promotion of the use of combined heat and power systems as expressed by the European Union combined heat and power strategy with the aim of increasing the participation of combined heat and power systems from the 9 percent of 1994 to 18 percent in year 2010. The author mentions several ways of improving the overall efficiency of gas turbines. The author concludes that the use of gas turbines for power generation has increased in the recent years and is likely to increase further due to the efficiency improvements and environmental benefits.

Ratts et al. (2000) present n experimental analysis of cycling in an automotive conditioning system. The authors used the second law of thermodynamics, in particular the entropy generation to quantify the thermodynamic losses of the refrigeration system's individual components under steady driving conditions at idle, 48.3 km/h and 96.6 km/h. The authors conclude that the performance of the system degrades with increasing speed. They claim that the thermodynamic losses increases to 18 percent as the vehicle changes speed from idle to 48.3 km/h and a further 5 percent as the speed if changed to a further 96.6 km/h. They attribute the deterioration of performance of the refrigeration system to the increasing cycling rate, with increasing speed, on the compressor, which is directly linked to the engine. The authors recommend that the solution might be a reduction in the range of operation of the compressor by using two cascaded compressors instead of one, or by not coupling the compressor to the engine.

Eames et al. (2000) evaluated the results of an experimental study of an innovative vapour recompression-adsorption refrigerator cycle. The cycle uses a steam jet-pump cycle, which acts as an internal heat pump, and upgrades the otherwise wasted heat from the solution concentrator. The upgraded heat is then used for part of the absorption process. The authors claim that the preliminary results indicate that the coefficient of performance is 14 percent less than anticipated from the calculations and they attribute it to the operation of the steam ejector that was used. They conclude that the vapour recompression-absorption refrigerator cycle was experimentally demonstrated to be practical and the cooling capacity was 5.5kW, which is 10 percent greater than it was designed for. The authors also point out that even though the coefficient of performance was 14 percent less than anticipated (1.03) it is still significantly greater than the usually quoted value of 0.7 for conventional single effect machines. The authors finally mention that the coefficient of performance is largely dependent on the performance of the steam ejector and they claim that it can be shown that the coefficient of performance is directly proportional to the entrainment ratio of the ejector.

Rousse et al. (2000) intended to find a way for fruit and vegetable producers in Québec to lower operating cost by recovering some of the heat from greenhouse ventilation systems, which are used to control humidity. The authors performed a feasibility study and built a counter-flow heat exchanger made of corrugated and flexible thermoplastic drainage tubing as the main core. The authors claim that the cost per unit met the requirements set by the *Union des Producteurs Agricole du Québec*. They also conclude that the unit performs well even in cold freezing climates and the average efficiency for volume exchange rates of 0.5 and 0.9 change/h were 84 and 78 percent respectively. Latent heat contributes about 40 percent of the total energy exchanged.

Shah et al. (2000) discuss in their paper various types of pollution imposed on the environment and the types of heat exchangers available. They discuss the role of heat exchangers for the reduction of air, water, land and thermal pollution. They also point out the requirement of heat exchangers that can be operated at high temperatures, beyond the capabilities of metals in highly corrosive environment. They also mention the dual function of catalytic reduction and heat transfer.

McDonald (2000) presents a low cost compact primary surface recuperator concept for microturbines. The author points out that microturbines in the 25-75kW range will find acceptance in larger quantities in the distributed power generation field and such small types of turbogenerators require an exhaust gas recovery recuperator in order to realise a thermal efficiency of 30 percent or higher. The author proposes and discusses a recuperator concept that meets the requirements for microturbines. In his paper, the author discusses the role that the recuperator has on the turbogenerator performance with the major requirements being the features and cost goals for a compact primary surface recuperator for microturbine service. The author mentions that the prototype matrix module was fabricated in Germany over two decades ago but was never commercialised because of lack of market in those days. The author claims that for a first generation state-of-art microturbine in the 50kW class, the basic recuperator matrix cost should not exceed $500 and he mentions that it is up to the heat exchanger manufacturers to realise the goal of about $10/kW for a new recuperator matrix in the near future.

Nguyen et al. (2001) developed a prototype low-temperature Rankine cycle electricity generation system. In their paper, the authors describe the development of the small scale system that uses a Rankine cycle with *n*-Pentane as the working fluid for the production of electricity from a low temperature heat source. The authors mention that the electricity generation system developed is able to operate using heat at 81°C and the estimated cost of the prototype unit is £ 21,560 and assuming a discount rate of 10 percent and a lifetime of 5 years, the authors claim the initial cost of electricity generated will be 64p/kWh for the 1.5kW prototype unit. The authors also claim that the prototype unit is capable of delivering 1.5kW of electricity with a thermal efficiency of 4.3 percent.

Maiza et al. (2001) investigated the thermodynamic and physical properties of some unconventional working fluids for use in a Rankine cycle supplied by waste energy sources. The authors present a series of results in graphical form for the various organic working fluids. The finally conclude that out of the fluids analysed in the study, R-123 and R-124 couple good system performance with high operative elasticity. They also conclude that the small recover system performance of R-401C is due to the strict thermal operative conditions imposed and because of the present interest of industries in blended fluids, R-401c or similar blended fluids may be the mist suitable fluid for an organic Rankine cycle.

Qu et al (2001) present a study on heat and mass recovery in adsorption refrigeration cycles. The authors developed an adsorption air conditioner and they performed experiments on mass and heat recovery. The authors claim that the mass recovery process will enhance the cooling capacity of per kg of air conditioning to about 20 percent and the promotion of the coefficient of cycle of the cycle depends on the operating conditions. They also conclude that the synthetical cycle with mass and heat recovery has the highest coefficient of performance among the calculated cycles and that it is 30 and 10 percent higher than the basic cycle and heat recovery cycle respectively.

Najjar (2001) presents a study of 12 research investigations performed to reflect the efficient use of energy by utilization of gas turbines combined cycles. The author points out that the gas turbine is characterized by its relatively low capital cost compared with steam power plants. The author also presents 10 of his research works carried out with his colleagues and some of which were also mentioned in his previous paper (2000). Some additional points mentioned are the repowering with combined cycles where the pinch point is not localised but rather delocalised, which means that there is no sudden phase change from the liquid to the gas phase. The advantage of the gas turbines outlined by the author are that of low cost, high flexibility, better reliability, early commissioning and commercial operation, and fast starting and loading.

Kolev et al. (2001) propose a new type of gas-steam turbine with increased efficiency, particularly effective for district heating. In the proposed type the main idea is a combination of a gas turbine, in which water steam is passed to the combustion chamber, with a system of contact economizers. The water vapour in the flue gases are condensed. This heat given off is then utilized thereby increasing the thermal efficiency. The authors claim that the thermodynamic efficiency is higher than the existing combined installations, including a gas turbine, a waste utilisation, and a steam turbine. The authors also claim that the overall thermodynamic efficiency calculated on the basis of lower calorific value reaches 108.7 percent.

Kato et al. (2001) present a thermal analysis of a magnesium oxide/water chemical heat pump for cogeneration. The authors outline the importance of heat storage in the field of energy utilization and the reduction of global carbon dioxide emissions. They investigated a chemical heat pump which achieves heat transformation via a chemical reaction. The authors proposed a combined system that consists of a heat pump and a diesel used for cogeneration. The diesel engine produces electricity and thermal energy simultaneously. The combined system used the thermal energy produced by the diesel engine. The authors claim that, by storing the surplus exhaust gas heat of the engine in the heat pump during low peak demand periods and by supplying the heat pump output during high peak demand periods they expect the heat output from the combined cycle during peak output to be several times that of a common cogeneration heat output using the existing exhaust gas boiler. The authors demonstrated the possibility of applying a chemical heat pump as a heat storage system by building and operating a laboratory scale model of the chemical heat pump.

3.3 Discussion

The literature reviewed shows that there is an enormous effort to improve the thermal efficiency of existing engines. This increased effort is boosted in many cases because of restriction on the limitation of the pollution level in all its form. However, there is increased pressure on thermal engines utilising fossil fuel, because of the global warming caused by excessive carbon dioxide emission and the Kyoto Protocol that limits the carbon dioxide emission level.

It was noticed while most of the effort are actually geared towards the larger engines like marine diesel engines and gas turbines, some work has already been done to investigate the waste heat recovery on smaller scales.

Enough information was obtained in the literature to motivate the investigation of a waste heat recovery system for smaller engines i.e. truck diesel engines.

4. ENERGY AVAILABLE FOR EXTRACTION

Before proceeding with the idea of setting up a waste heat recovery system it is necessary to establish whether implementation of such a system will indeed be profitable and feasible. Therefore, the amount of waste heat energy available in the exhaust gases should first be established and then, the conversion of the recovered energy should be investigated.

4.1 The Delivery Temperature of the Exhaust Gases

The location chosen for the waste heat recovery plant in the laboratory involves a travel distance of at least 3.9 metres for the exhaust gases (Koorts, 1998). The average delivery temperature of the exhaust gases from the engines was measured by Koorts (1998) to be nearly 600°C. This is a very suitable temperature for setting up the waste heat recovery system, however, the 3.9 metres pipes causes a considerable temperature drop of the exhaust gases. Calculations performed show that the exhaust gases can be supplied to the waste heat exchanger at a temperature of around 550 °C with 20mm fibre glass insulation around the pipes as opposed to supply temperature of 460°C with non-insulated pipes. The temperature profile of the exhaust gases in the pipe with various insulation thicknesses is presented in appendix H.

A delivery temperature of 500 °C is used in the calculations because the exhaust pipe section in the tests cells cannot be readily insulated and the length of travel of the exhaust gases from the engine to the exhaust chamber is quite significant. The silencers would also contribute to an increased temperature loss because of the larger surface area involved.

4.2 The Lower Temperature Limit

There obviously exists a lower limit to which the exhaust gases can be dropped. As the temperature of the exhaust gases is reduced, water will start condensing when the dew point temperature is reached and then there is risk a of sulphurous gases and other impurities dissolving and hence formation of acidic liquid which will corrode the pipes. In order to calculate the minimum exhaust gas temperature the fuel-air ratio has to be investigated. Sample calculations are presented in the next sections.

4.2.1 Typical calculation of fuel-air ratio

The worst case will occur when there is a maximum amount of moisture in the exhaust gases, as this will limit the lowest temperature the exhaust gases can be cooled down to and this will occur for a stoichiometric mixture ($\phi = 1$).

The general combustion equation for a hydrocarbon fuel in air is

$$\phi.\varepsilon.C_{\alpha}H_{\beta}O_{\gamma}N_{\delta} + (0.21\ O_2 + 0.79\ N_2) =$$
$$v1\ CO_2 + v2\ H_2O + v3\ N_2 + v4\ O_2 + v5\ CO + v6\ H_2. \quad \text{....(4.1)}$$

For diesel fuel,

$$\alpha = 14.4\ ;\ \beta = 29.9\ ;\ \gamma = 0\ ;\ \delta = 0,$$

and for a stoichiometric combustion

$$v4 = 0\ ;\ v5 = 0\ ;\ v6 = 0\ .$$

Therefore for a stoichiometric mixture combustion of diesel can be written as.

$$\varepsilon.C_{14.4}H_{29.9} + (0.21\ O_2 + 0.79\ N_2) = v1\ CO_2 + v2\ H_2O + v3\ N_2. \quad \text{....(4.2)}$$

Balancing the equation we obtain

$$v1 = 0.137\ ;\ v2 = 0.142\ ;\ v3 = 0.79\ ;\ \varepsilon = 9.5 \times 10^{-3}$$

Therefore the fuel to air ratio will be given by

$$F_s = \text{Mass of Fuel / Mass of Air}$$

After substituting the relative molecular masses of the constituents into the above equation and solving we obtain $F_s = 14.6$.

4.2.2 Calculation of minimum temperature

Molar mass of exhaust gases =

$$\Sigma \text{ Relative Molar Mass (RMM) of each gas } \times \text{ Mole fraction of gas}$$

$$= (\text{RMM of } CO_2 \,.\, \text{MF of } CO_2) + (\text{RMM of } H_20 \,.\, \text{MF of } O_2) + (\text{RMM of } N_2 \,.\, \text{MF of } N_2)$$

$$= (44.01 \times 0.137) + (18.015 \times 0.142) + (28.013 \times 0.79)$$

$$= \underline{30.717 g/mol}$$

The specific, w, and relative humidity, ϕ, of the exhaust gases can be calculated from equations given by Çengel and Boles (1989).

$$w = M_{vapour}/M_{gas} \qquad \text{....(4.2.2.1)}$$

$$= \text{Mass of water} / (\text{Mass of } N_2 + \text{Mass of } CO_2)$$

$$= \underline{0.09084}$$

$$\phi = wP/((0.622 + w)P_g) \qquad \text{....(4.2.2.2)}$$

At saturation, $\phi = 100\%$,

The exhaust pressure was measured to be 8kPa gauge by Koorts (1998).

Therefore,

$$P = 101.3 + 8 = 109.3 kPa. \qquad \text{....(4.2.2.3)}$$

Solving the above equation we obtain $P_g = 13.93$kPa.

From the tables presented by Çengel and Boles (1989), the corresponding condensing temperature is

$$T_{dew\ point} = T_{sat@Pg} = 52.1^{o}C.$$

Therefore, in order to avoid condensation in the exhaust gas stream the temperature should not drop below 52.1°C.

4.3 Calculation of the Amount of Useful Energy Available.

The specific heat capacity is calculated as by adding the product of each gas present in the exhaust gases with their respective specific heat capacity and this is presented in appendix D. Having obtained the temperature limits, the specific heat capacity and the measured mass flow rate, the amount of energy that can be extracted from the exhaust gases before condensation occurs can now be calculated using the following relation,

$$\Delta H = m_{ex}\, c_{p,exh}\, \Delta T \qquad \text{....(4.3.1)}$$

A typical exhaust gas mass flow rate is calculated in appendix A to be 0.0272 kg/s per litre engine capacity. Therefore for the 6l engine investigated in this case the mass flow rate is 0.1632 kg/s. The amount of potential heat energy that can be extracted is,

$$\Delta H = 0.1632 \times 1.07 \times (493 - 51.6) = \underline{76.99\text{kW}} \qquad \text{....(4.3.2)}$$

In view of the overall engine capacity, this amount of energy represents a significant portion of energy that can instead be used to produce superheated steam and hence improve the thermal efficiency of the fuel.

4.4 Conversion of Recovered Waste Heat

To utilize the waste heat recovered it was decided that the heat recovered will be used to produce steam which can be used in a Rankine cycle to produce work, that is for the generation of electrical power.

For maximum efficiency the Rankine cycle should be operated at the highest possible pressure. The optimum pressure at which the Rankine cycle can be operated is limited at the pinch point. At this point the temperature difference between the water and the exhaust gas stream is a minimum. The water is in the saturated liquid state. A detailed analysis of the Rankine cycle and the temperature profiles is presented in appendix C and the results are shown in graphical form in figures 4.1 to 4.3 below. It can be seen that the optimal pressure is 800kPa on the water side with a temperature difference at the pinch point of about 10°C, which is appropriate for proper heat transfer to take place between the two streams.

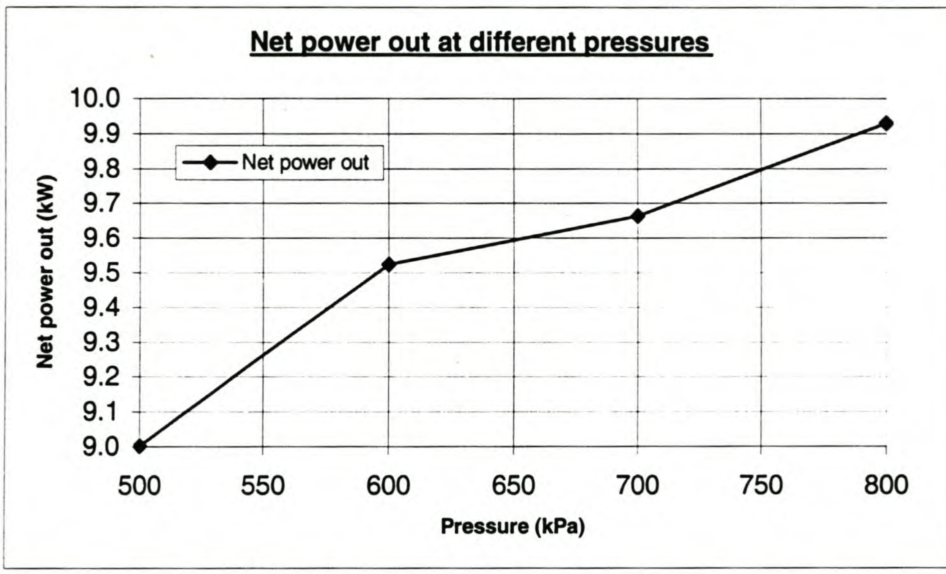

Figure 4.1: Net power output (m_w = 0.015kg/s) from the Rankine cycle at different pressures

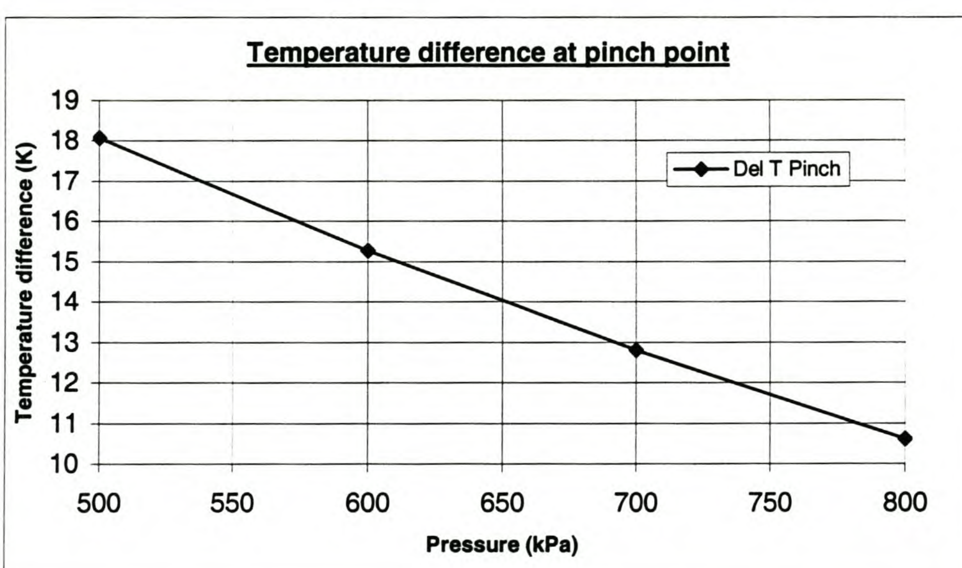

Figure 4.2: Temperature difference at pinch point for different pressures

As mentioned above the cycle efficiency of the Rankine cycle increase with increasing pressure and this is shown in figure 4.3 below.

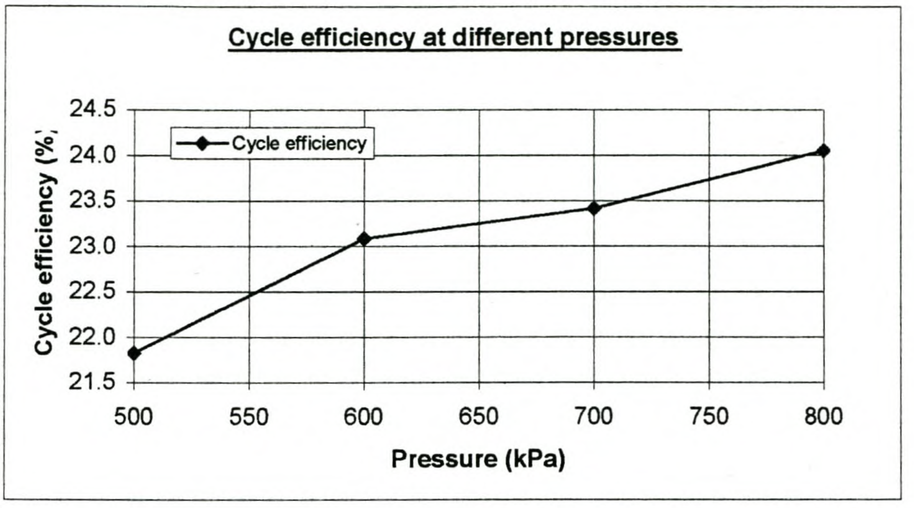

Figure 4.3: Cycle efficiency at different pressures

At this stage it can be stated that there is enough energy to operate a Rankine cycle. If operated at an optimal pressure of 800kPa, it can be seen that the Rankine cycle can be operated at a cycle efficiency of 24%. In the calculations it was assumed that the heat exchanger has a combined efficiency of 60%, the pump has an efficiency of 70% and the turbine an efficiency of 85% based on commercially available units.

5. THE WASTE HEAT RECOVERY EXPERIMENTAL APPARATUS

In this chapter the experimental apparatus set up for the purpose of this project is described.

Two main sources of exhaust gases were used in this project: a continuous combustion unit in the Energy Systems Laboratory (ESL) and three test cells from the Centre for Automotive Engineering (test cells number 5,6 and 7).

The three test cells are located at the ground floor and the experimental apparatus had to be set up in the ESL, which is situated on the first floor above these test cells.

5.1 Pipe Network

In order to direct exhaust gases from the test cells to the Heat Recovery Steam Generator (HRSG), a system of pipe network with the necessary accessories had to be designed.

The following had to considered while designing the pipe network:

1. The existing exhaust pipes initially purged all exhaust gases directly into a concrete exhaust gas chamber (below the floor of ESL) from where it was extracted and rejected into the atmosphere by a fan and this should still be possible whenever required.
2. The exhaust gases should not be allowed to leak into the ESL because of its hazardous nature and hence they should be contained in a closed system.
3. Pressure drop within the piping network should be kept as low as possible so as not to exert too much back pressure which in turn will affect the performance of the engine under test.
4. Three holes each of 200mm diameter were already drilled in the floor leading to the concrete exhaust channel from which the exhaust gases should enter the ESL. In addition to the pipes, a valve system had to be designed such that the exhaust gases are not blocked at any time. Therefore, at least two valves

should be present enabling the exhaust gases either to circulate into the pipe network or being purged directly into the concrete exhaust gas chamber.

5. The used exhaust gases from the HRSG should be directed into the concrete exhaust room from where it will be extracted.
6. The auxiliary unit, which is a continuous combustion unit, is coupled to the pipe network leading to the HRSG.
7. A special valve had to be designed to throttle the exhaust gases coming from the continuous combustion unit.
8. Because of the consequent length of the pipes involved the thermal expansion of the pipes has to be considered and as shown below they were found to be significant hence requiring the use of linear expansion bellows.

To gain a better understanding of the locations of the CAE test cells, the ESL and the concrete exhaust gas room, the reader is referred to appendix J.

5.1.1 The initial proposal

This project is the extension/implementation of the project prepared by Koorts (1998) as partial requirement towards his undergraduate degree. His proposal is briefly described first, then the drawbacks and modifications are explained in the following sections.

Pipe network in the concrete exhaust room

Each of the three sets of pipes connections proposed by Koorts (1998) comprised of a horizontal pipe from the existing exhaust gas discharge and a 90 degrees T-piece. One vertical pipe from the T-piece would lead the exhaust gases into the ESL through each porthole in the floor. A second horizontal pipe was to be welded on the remaining T-piece outlet on which, a butterfly valve was to be welded. Each butterfly valve was to be operated by an extended handle from the ESL through the same 200mm porthole into which the exhaust gas pipe runs.

Figure 5.1 below should give the user a better understanding of the location of the exhaust gas chamber and the piping structure initially proposed within, for diverting the exhaust gases into the ESL.

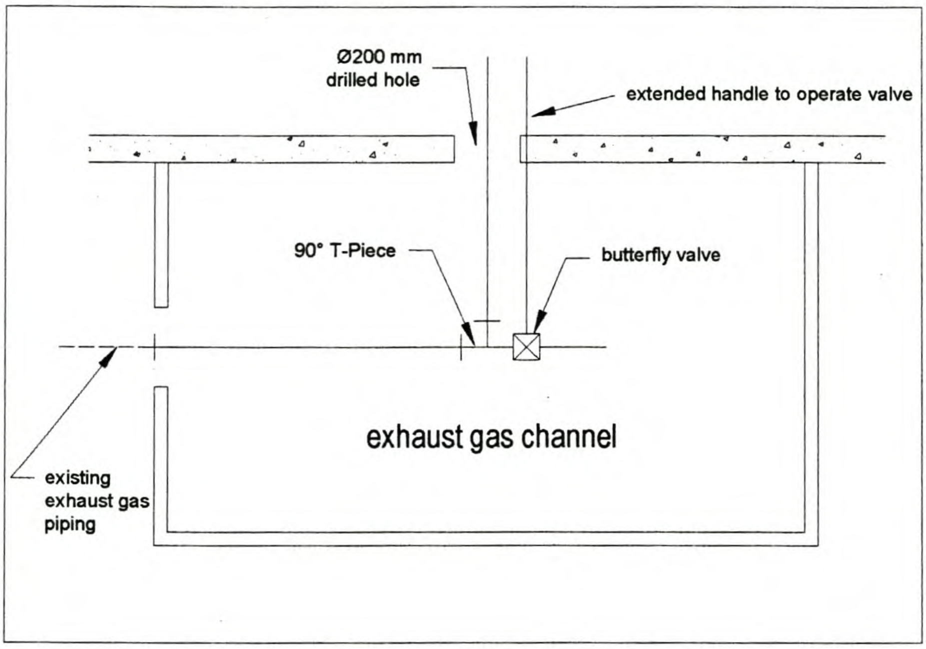

Figure 5.1: Pipe network below floor level in the ESL

Pipe network above the floor in the ESL

The pipe network inside the ESL as proposed by Koorts (1998) is shown in figure 5.2. The three vertical pipes from the portholes are to be connected in a straight line and then the pipeline runs back parallel to itself and finally discharge the exhaust gases into the concrete exhaust gas room after going through the waste heat exchanger. A continuous combustion unit, also referred to as the auxiliary unit, is to be connected to the pipe network just before the waste heat exchanger.

Two valves were also proposed just before the waste heat exchanger (not shown in figure 5.2). These valves would allow the user to select the source of exhaust gases i.e. from the pipe network running from the test cells or from the continuous combustion unit.

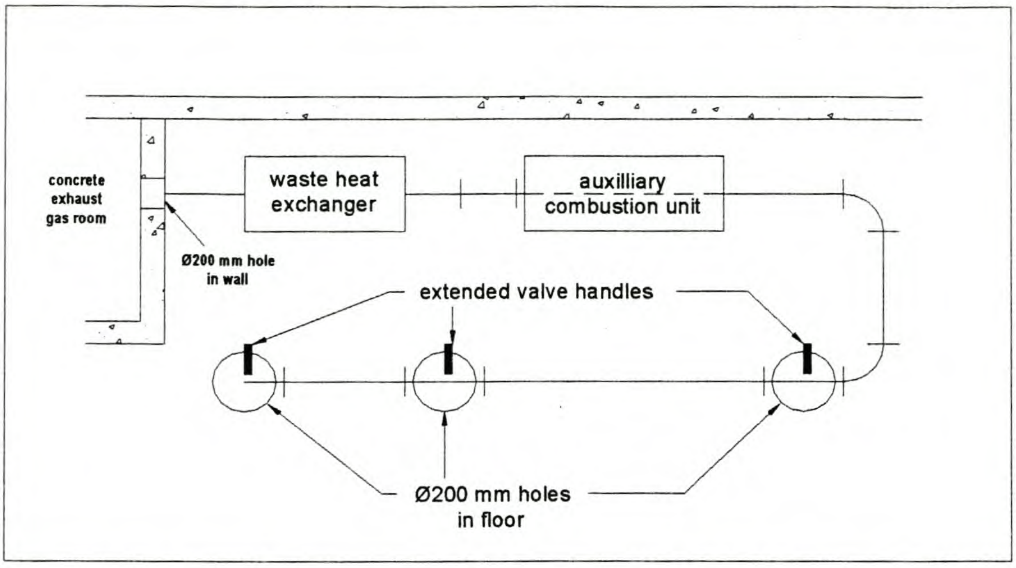

Figure 5.2: Pipe network above floor level in ESL

5.1.2 Drawbacks and modifications of the initially proposed pipe network

The initial proposed network was analyzed and the shortcomings were identified. The required modifications were then implemented to overcome the problems foreseen in the initial proposal. The first problem was that a hole of 200mm in the floor would not provide sufficient space, for both an extended handle of the butterfly valve and a 4 inches exhaust pipe if a T-piece and a butterfly valve were used in the concrete exhaust gas duct. The solution proposed and implemented was that of using a 45 degree T-piece and a 45 degree elbow used instead of a single 90 degree T-piece.

Also, instead of a commercial butterfly valve, a simple disk-in-tube valve was designed and manufactured out of mild steel in the workshop itself and this required less space to be fitted. The space limitation in the 200mm porthole is shown schematically in figure 5.3a. The main reason for this was the high cost of commercially available butterfly valves and the possibility of clogging up after long use. The new valves were made in the departmental workshop (SMD) out of mild steel only. In addition to the low cost and lower risk of being clogged up, these valves could be fitted inside the concrete chamber more compactly.

It was then possible to fit the handle and the pipe both through the 200mm holes. The 45 degrees take-off in addition to allowing more space for fitting the valve reduces the pressure drop as compared to the 90 degrees takeoff in the exhaust gas stream. The modified design, allowing both the exhaust pipe and the extended handle to be fitted in one porthole, is shown in figure 5.3b.

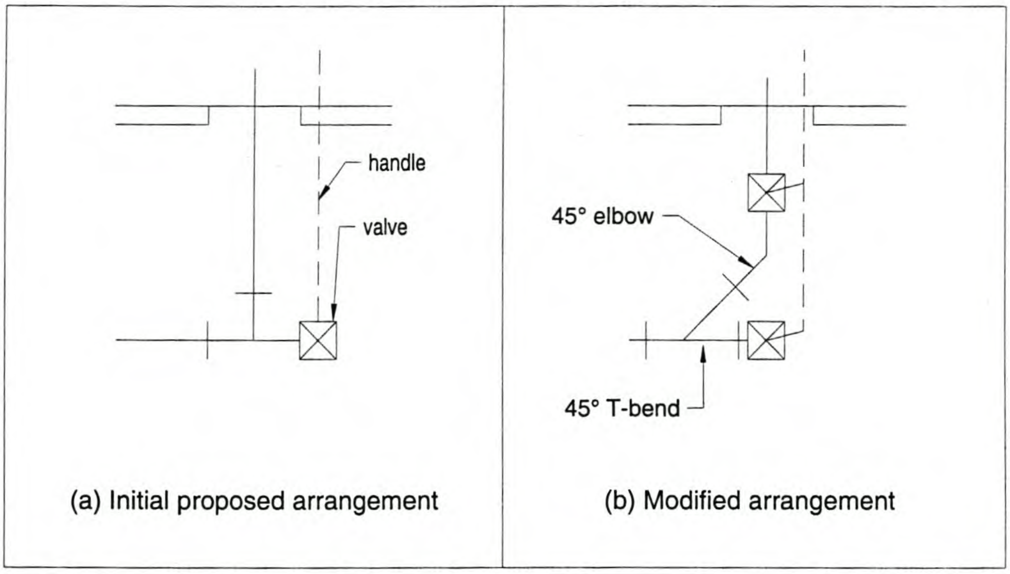

Figure 5.3: Modification to the pipe arrangement below floor level in ESL

It can also be seen in figure 5.3b that two interlinked valves were used instead of one as proposed by Koorts (1998). The use of two interlinked valves ensures that at any given moment at least one of the two valves is in the open position hence protecting the engine operating in the test cells. No exhaust gas is blocked inside the pipe network hence no excessive backpressure is exerted on the engines.

The other problem encountered was that the hole drilled in the floor for the test cell number 7 was in fact in the middle of two exhaust gas exit points both coming from the same test cell. A system different to the other to the take off from test cells 5 and 6 had thus to be designed taking into account the two exhaust gas intake points instead of one from the test cells. This led to the design shown schematically in figure 5.4. It required the use of three valves interlinked together such that when valves 2 and 3 are open valve 1 is closed or vice versa.

Again, the commercial butterfly valves proposed by Koorts (1998) were replaced by locally manufactured simple disc in pipe valves.

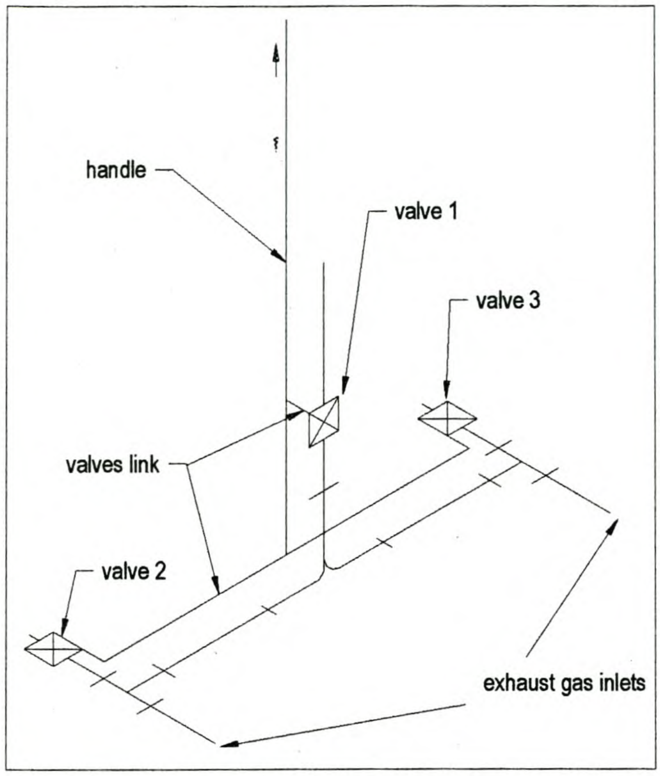

Figure 5.4: Piping system below floor level (for two exhaust gas lines).

A detailed drawing of the butterfly valve is given in appendix J.

For detailed drawings of the above mentioned networks systems and other components, the reader is referred to appendix J.

5.2 The Waste Heat Exchanger.

The waste heat recovery system proposed by Koorts (1998) comprised of three distinct units namely a pre-heater, a boiler and a super-heater. These three heat exchangers were to be arranged vertically such that the pre-heater is at the bottom and the super-heater on top with the boiler section in between. This arrangement was chosen such that gravity is used to ensure that only steam leaves the boiler to enter the super-heater. In the following section a brief description of the three units waste heat steam generator is presented and then the drawbacks identified.

5.2.1 The three stage waste heat recovery steam generator.

The pre-heater proposed was a counter flow arrangement in an annular construction of a 100mm pipe within a 150mm pipe. The inside fluid is exhaust gases and the outside fluid sub-cooled water. Water at a pressure of 600kPa is to enter at 35°C and leave the pre-heater as saturated liquid at 158.9°C.

The boiler unit is a shell and tube heat exchanger consisting of 19 tubes each of nominal diameter of 12mm and a shell diameter of 100mm. Again, the water and exhaust gases are in counterflow. Saturated water leaving the pre-heater are to enter the boiler at a pressure of 600kPa and leave as saturated vapor at a saturated temperature of 158.9°C.

The super-heater section has a similar arrangement to the boiler. Saturated vapor leaving the boiler will enter the super-heater and get superheated as much as possible depending on the maximum temperature at which the exhaust gas is available.

5.2.2 Identification of the drawbacks of the three stages waste heat recovery steam generator

Preliminary calculations performed on the pre-heater section showed that the length of such a heat exchanger meeting the saturated water exit temperature of 158.9°C would require a total length of 7m. Such a length was judged not feasible because of space limitation inside the ESL and also, there will be significant heat loss to the surrounding with the greater surface area involved. A detailed calculation performed on the pre-heater section is presented in appendix D.

The use of a shell and tube arrangement was then investigated for the pre-heater section. The arrangement considered was eight 10mm pipes inside a 106mm shell. It was found that, with such a heat exchanger of 2.5m length an exit temperature of approximately only 98°C as compared to the required 158.9°C would be achieved. Again increasing the length of such a heat exchanger was not considered, as it will again result in a large area exposed for heat loss to the surrounding and because of the space limitation. Detailed calculations for the shell and tube heat pre-heater are presented in appendix D.

From the calculations performed for the two pre-heater sections it was found as expected that the heat transfer coefficient on the air-side was the greater determining factor in the size of the heat exchanger. It was then decided that a finned heat exchanger be investigated. A heat exchanger manufacturing company was contacted to enquire about any standard types of finned heat exchangers produced by them. The company was provided with the water and exhaust data inlet conditions and outlet requirements. Preliminary calculations were performed and are presented in the next chapter. It was also decided that the pressure be reduced to atmospheric pressure. The types of heat exchangers considered are also discussed in the next chapter.

As optimization of the heat exchanger was not the aim of this project, the proposed heat exchanger was purchased, sponsored by the CAE. Tests were carried out to obtain the characteristics of the heat exchanger before being integrated to the waste heat recovery apparatus. It was intended rather to give insight into optimizing the waste heat recovery system. The tests performed are explained in chapter 6 and the results obtained during testing and after implementation of the heat exchanger into the waste heat recovery system are given in chapter 9.

5.3 The Continuous Combustion Unit

As mentioned earlier, a continuous combustion unit was used as an alternative source of exhaust gases for the purpose of this project.

During his student exchange program, Schwack (2000) was initially responsible to overhaul the continuous combustion unit on which further practical sessions were to be carried out by undergraduate students.

Schwack (2000) overhauled the apparatus to run on propane gas. However, for the purpose of this project the exhaust gas outlet from the combustion unit had to be linked to the HRSG via waste heat recovery pipe network. Also, the combustion unit required further modifications to allow it to run on diesel.

The following have to be considered when connecting the exhaust gas outlet to the pipe network in the laboratory:

1. How would the performance of the combustion unit be affected if backpressure is exerted in the combustion chamber?
2. Due to the harmful nature of the exhaust gases they have to be within a closed system until purged into the atmosphere.
3. It should be possible to relieve any backpressure in case of severe backpressure building up.
4. With the extraction fan running in the concrete exhaust chamber, a negative pressure will be exerted on the exhaust gas outlet, which in fact would be advantageous.

In the following sub sections the continuous combustion unit and the modifications made to link the exhaust gas supply of the continuous combustion unit to the waste heat recovery exhaust gas supply line are described.

The continuous combustion unit components[1]

The University of Stellenbosch purchased the continuous combustion unit in the 1970's. The manufacturer is P.A.Hilton Ltd. The continuous combustion unit was designed for research into combustion and fuel technology and for demonstration of the handling and operation of typical furnace equipment to students and technicians. It comprises of a water cooled stainless steel combustion chamber. Sight glasses are also provided to see the actual flame inside the combustion chamber. Air for combustion

[1] Information obtained from the Continuous Combustion Unit description pamphlet (P.A.Hilton Ltd.1997)

is supplied by a B.V.C type Y3/100 three stage centrifugal blower. The unit is also equipped with a Schiedeldrop No.3 Combustion burner.

Instrumentations present are:

- Orifice plate and differential manometer with direct reading scales for the air flow.
- Glass tubes Rotameters with direct reading scales for the fuel and water flow
- Manometer for the gas pressure measurement of outlet from gas pressure reducing valve.
- Water thermometers for inlet and exit water temperature measurement.
- Gas outlet thermocouple in chimney.
- Air inlet thermometer in lower outlet pipe.

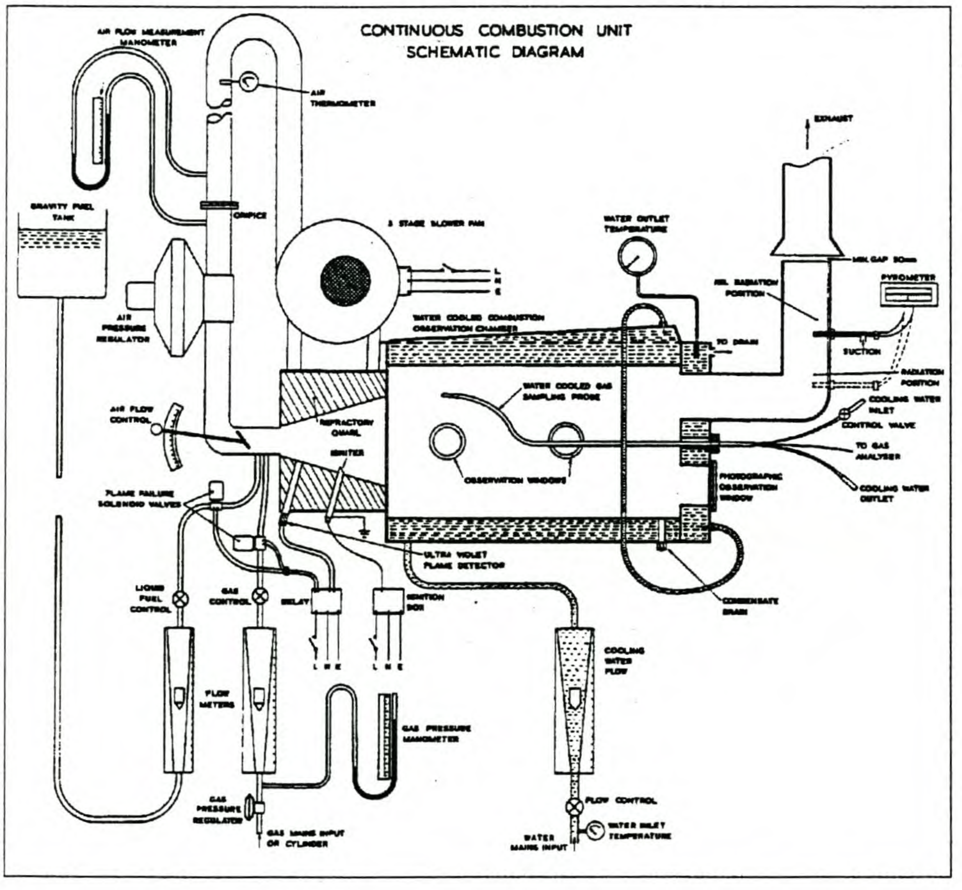

Figure 5.5 : Schematic diagram of the continuous combustion chamber and all its components (P.A. Hilton Ltd.)

Linking the exhaust gas outlet from the test cells to the waste heat recovery exhaust gas pipe network

To connect the exhaust gas outlet from the combustion unit to the waste heat recovery apparatus a 4 inch stainless steel pipe was welded as shown in the schematic below. The existing exhaust outlet was covered with a lid to prevent exhaust gases from being thrown into the ESL and at the same time to divert it into the exhaust gas supply line of the waste heat recovery system. Removing the lid will relieve excessive backpressure inside the combustion unit. It is expected that in case of high pressure build up this lid will be pushed open. The exhaust pipe from the combustion unit was not welded to the waste heat recovery pipe in order to allow for thermal expansion. A small gap was allowed, which was closed by means of a stainless steel sheet wrapped around the pipe and help by a clamp as shown in figure 5.6.

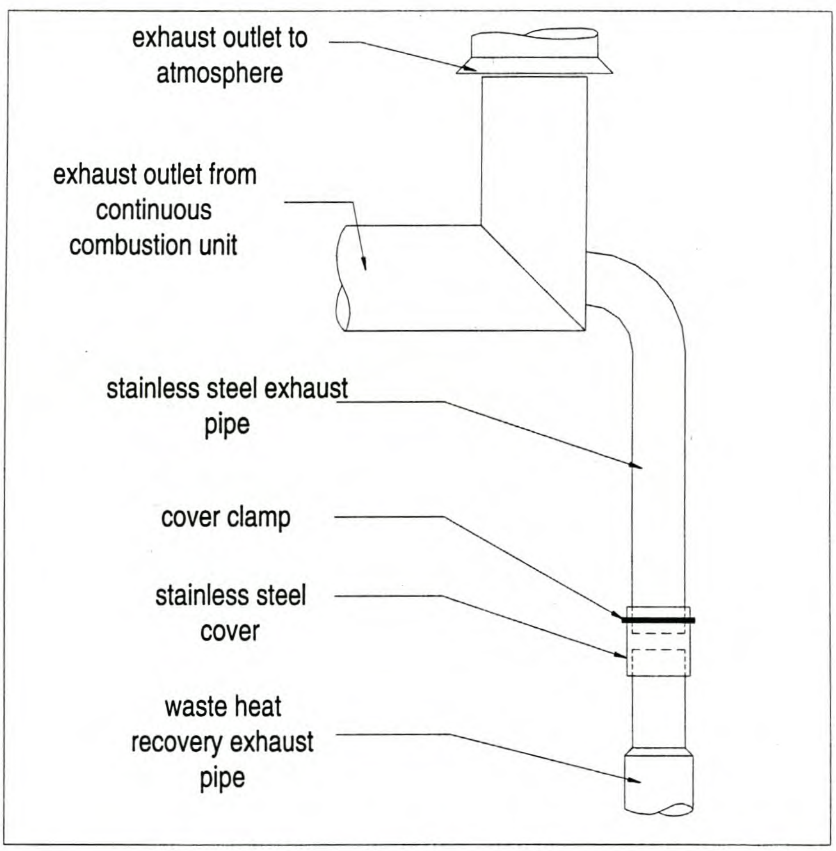

Figure 5.6: Exhaust pipe link of the outlet of continuous combustion unit to the waste heat revovery system

The final modification made was to connect the diesel tank to the fuel supply of the continuous combustion unit as it was previously operational on the gas fuel only. This required connecting the diesel supply that required to be put at a height of at least 2m above the combustion unit according to the manufacturers specifications. This height is important because diesel is drawn by gravity to the combustion unit. A number of calibration experiments were done and these are presented in chapter 7.

The cooling water out of this combustion unit was also used as the inlet water for the HRSG. This choice was motivated by two main factors as mentioned below.

- Water entering the HRSG can also be preheated by the cooling water running through the engine as shown by Zhang (1997) who used it for adsorption cooling system for automobile waste heat recovery system.
- By regulating the water flow rate through the cooling jacket around the combustion unit can monitor the water inlet temperature.

5.4 Water Connection to Waste Heat Exchanger

The connection for the water supply from the cooling water outlet of the continuous combustion unit to the waste heat exchanger is shown schematically in figure 5.7 below. A T-piece was installed into the cooling water outlet pipe of the continuous combustion unit. Two gate valves are installed to allow the user to regulate the amount of water flowing into the waster heat exchanger and also the amount of water being drained. It should always be ensured beforehand that the main valve, which is the one draining all the cooling water, is initially fully open so as not to cause pressure build up in the cooling water outlet pipe.

Then the gate valve leading to the waste heat exchanger is slowly opened. If required the main valve can be closed partially to divert more water in the waste heat exchanger. Hence by adjusting the two water valves shown schematically in figure 5.7 below, the required mass flow rate into the waste heat exchanger can be achieved.

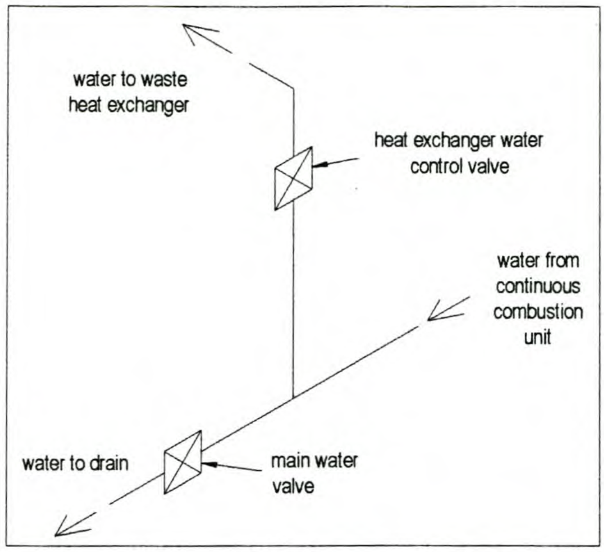

Figure 5.7: Water stream connection to waste heat exchanger

5.5 The Steam Separator

When the exhaust gases are initially diverted to the waste heat exchanger the water within is heated but during the heating process the water undergoes heating and eventually boiling. When the water in the superheater region of the heat exchanger is boiling, the liquid/vapour mixture water is spurted out in jets, which extend quite far. Because of safety reasons primarily a steam separator was designed and built. Also this would facilitate further experiments for the determination of the maximum mass flow rate of water than can be superheated for some other exhaust gas source. With the steam separator it is possible to measure the condensate flow rate hence also gives an indication of the amount of steam formed. The steam separator is shown schematically in figure 5.8 below.

The steam separator is made entirely of stainless steel because of its low thermal conductivity and also because it is corrosion resistant. The steam separator consists of an inlet, which can be screwed directly onto the waste heat exchanger exit, a U-tube at the bottom for condensate exit and an open end on top for steam exit. It also has a port where a thermocouple can be inserted to measure the temperature of the steam coming out of the heat exchanger.

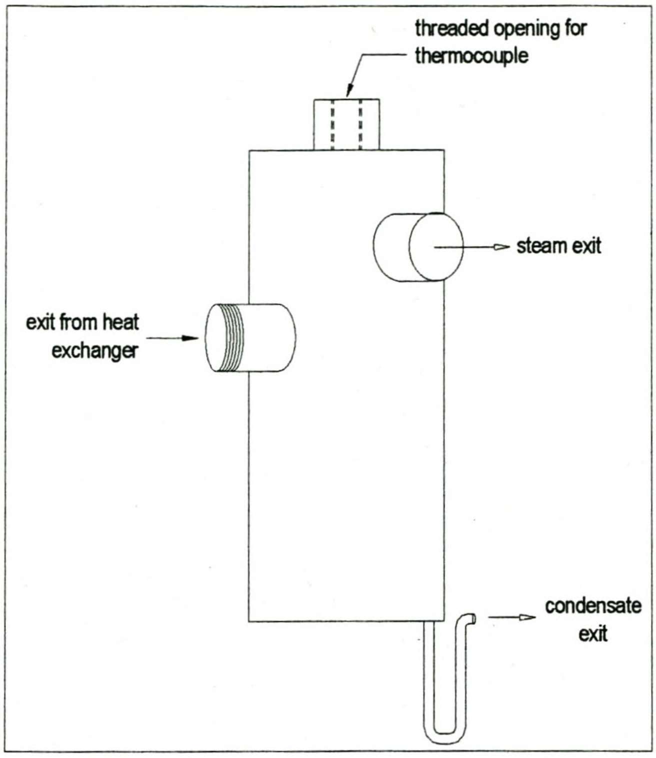

Figure 5.8: Steam separator

The opening on side will allow for the steam to be further superheated is any other further heat exchanger is added to the system or for supplying same to a turbine for work output, hence a complete waste heat recovery unit with useful work generation will be complete.

6. THE WASTE HEAT EXCHANGER

To extract waste heat from exhaust gases a waste heat exchanger is required. This is the most important component used in this project, the aim of which is to extract as much heat as possible from the exhaust gases. The efficiency of the Rankine cycle will be partly limited by the efficiency with which heat is extracted from the exhaust gases. In the design of the waste heat exchanger the following had to be kept in mind:

- The minimum temperature in the exhaust gas stream is limited to the temperature at which condensation will take place.
- Pressure drop in the exhaust gas stream should be kept to a minimum.
- Care should be taken while selecting he materials involved because of the high temperatures in the exhaust gas stream limit and the corrosive nature of water at elevated temperatures.

6.1 Specification and Preliminary Calculations

6.1.1 Specifications of the finned tube heat exchanger

Several heat exchangers as mentioned in chapter 5 earlier were investigated, however a cross-flow, finned copper tube heat exchanger proposed by Stevens (1999) was purchased. The specifications are presented in Table 6.1 below.

Manufacturing company	YUCON COIL
Company model number	3SC-254/254/10FS/10R Cu/Cu/SS
Pipe material	Copper
Fin material	Copper
Housing	Stainless Steel
Width	$W =$ 254mm
Height	$H =$ 254mm
Length	$L =$ 254mm
Number of rows	$N_r =$ 10
Number of pipes per row	$N_p =$ 10

Table 6.1a: Heat exchanger specifications and details

Air side finned surface	
Fin pitch	$P_f =$ 3.175mm
Fin thickness	$T_f =$ 0.14mm
Transversal tube pitch	$P_t =$ 19.05mm
Longitudinal tube pitch	$P_l =$ 33.0mm
Water side	
Pipe outer diameter	$OD_t =$ 9.53mm
Wall thickness	$t_t =$ 0.41mm

Table 6.1b: Heat exchanger specifications and details

The heat exchanger was tested at 2580kPa, for any leakages by the manufacturing company.

6.1.2 Preliminary calculations

The preliminary calculations performed on the finned tube waste heat exchanger prior to purchase are briefly presented below. For a more detailed calculation the reader is referred to appendix D.

In the calculations it is assumed that the properties of exhaust gases are similar to air at the same temperature.

The properties of air and water are calculated using the equations in appendix B.

Consider the preheater section.

The conditions of the fluids are presented in table 6.2 below. The lowest temperature to which the exhaust gases would condense was found to be 52.1°C but because of the presence of water molecules 100°C is chosen as a safer lowest temperature limit for the exhaust gas to cover for cooling losses at pipe walls.

Mass flow rate		
Water	$m_w =$	0.015 kg/s
Exhaust gas	$m_{exh} =$	0.1632 kg/s
Temperature		
Water inlet	$T_{w,in} =$	35°C
Exhaust gas inlet	$T_{exh,in} =$	145°C

Table 6.2: Preheater fluid conditions

Also it is expected that at 100°C water condensate will start forming and the other constituents present in the exhaust gases will dissolve and hence leading to formation of acidic solution, which might corrode the pipes and the heat exchanger.

Average air temperature in preheater section $= (T_{air,in} + T_{air,out}) /2$ .....(6.1)

$= (141°C + 100 °C) /2$

$= 120.5°C.$

Properties of air at 120.5°C,

$c_{p,air} =$	1.017×10^3 J/kg.K
$\mu_{air} =$	2.359×10^{-5} kg/m.s
$k_a =$	0.035 watt/m.K
$Pr_{air} =$	0.687

Table 6.3: Properties of air at average temperature in preheater section

Water will leave the preheater section as saturated liquid. At atmospheric pressure the saturation temperature of water is 100.24°C.

Average water temperature in preheater section $= (T_{w,in} + T_{w,out}) /2$ .....(6.2)

$= (35°C + 100.24°C) /2$

$= 67.62°C.$

Properties of water at 67.62°C

ρ_w =	968.45 kg/m^3
$c_{p,w}$ =	4.204 x 10^3 J/kg.K
μ_w =	3.279 x 10^{-4} kg/m.s
k_w =	0.673 watt/m.K
Pr_w =	2.049

Table 6. 4: Properties of water at average temperature in preheater section

Consider a small control volume between the centerlines of two adjacent tubes and two adjacent fins.

Calculating the heat transfer coefficient on the air side

Minimum flow area, $A_{cv,c}$ $= (Pt\text{-}OD_t).(P_f\text{-}t_f)$ .... (6.3)

$= 4.338 \times 10^{-5}\ m^2$

Frontal area, $A_{cv,fr}$ $= P_t \,.\, P_f$ .... (6.4)

$= 8.065 \times 10^{-5}\ m^2$

Area ratio, σ_a $= A_{cv,c} / A_{cv,fr}$ .... (6.5)

$= 0.544$

Fin surface area exposed to air stream, $A_{cv,f}$ $= 2.\ ((P_l \,.\, P_t) - (\pi \,.\, Od_t^2)/4)$ .... (6.6)

$= 9.749 \times 10^{-4}\ m^2$

Tube surface area exposed to air stream, $A_{cv,t} = (Pf - tf)\ .\ (2\pi.OD_t)$ .... (6.7)

$= 1.656 \times 10^{-4}\ m^2$

Total area exposed to air stream, $A_{cv,a}$ $= A_{cv,f} + A_{cv,t}$ .... (6.8)

$= 1.141 \times 10^{-3}\ m^2$

Hydraulic diameter of control volume, d_{hyd} $= (4 \,.\, A_{cv,c} \,.\, P_l) / A_{cv,a}$ .... (6.9)

$= 3.386 \times 10^{-3}\ m^2$

Air mass velocity through minimum free flow area of the core,

$$G_{ai,c} = m_{exh} / (A_{fr} . \sigma_a) \quad \text{........(6.10)}$$
$$= 4.649 \text{ kg} / (m^2 \text{ s})$$

Corresponding Reynolds number, $Re_{exh,i} = G_{ai,c} . d_{hyd} / \mu_a$ .... (6.11)

$$= 667.3$$

From Fraas et al. (1965) , we read for Re = 667.3,

$$St.Pr^{2/3} = 0.012 \text{ and } f_{appi} = 0.0315$$

Substituting Pr = 0.687, we have,

$$St = 0.02$$

Hence heat transfer coefficient on the exhaust gas side can be calculated using the definition of the St number,

$$St = h_{exh} / (G_{ai,c} . c_{p,a}) \quad \text{........ (6.12)}$$

$$h_{exh} = 92.294 \text{ W/(m}^2\text{.K)}$$

Calculating the heat transfer on the water side

Reynolds number, $Re_w = 4.m_w /(\pi . ID_t . \mu_a)$ .... (6.13)

$$= 5.622 \times 10^3$$

Re > 2500, therefore we have a turbulent flow hence according to equations B.6 and B.7 we have,

$f_{D,w} = 0.037$, and $Nu_w = 28.31$, respectively.

Therefore we can calculate the heat transfer coefficient on the water side,

$$h_w = Nu_w . k_w / ID_t \qquad \text{........ (6.14)}$$
$$= 2264 \text{ W/(m}^2\text{.K)}$$

Calculating the overall heat transfer coefficient for each control volume

$$UA_{cv} = \left(\frac{1}{h_{w,i} A_{cv,w}} + \frac{t_t}{k_{copper} A_{cv,w}} + \frac{1}{h_{exh,i} A_{cv,a}} \right)^{-1} \qquad \text{........ (6.15)}$$

UA_{cv} =0.0057 W/K

Consider the first two rows of pipes.

$$UA = UA_{cv} . N_{pipes} . L_t / P_f \qquad \text{........ (6.16)}$$
$$= 90.402 \text{ W/K}$$

Effectiveness of the heat exchanger

$$C_{max} = m_{exh} . c_{p,a} \qquad \text{........ (6.17)}$$
$$= 166.012 \text{ W/K}$$

$$C_{min} = m_w . c_{p,w} \qquad \text{........ (6.18)}$$
$$= 62.841 \text{ W/K}$$

$$C = C_{min}/C_{max} \qquad \text{........ (6.19)}$$
$$= 0.379$$

Number of transfer units, $N = UA/C_{min}$ .... (6.20)

$= 1.439$

From table 3.5.1 in Kröger (1998), effectiveness for two unmixed fluids is given by,

$$e = \frac{\left(1 - e^{-C\left(1-e^{-N}\right)}\right)}{C} \qquad \text{........ (6.21)}$$

$$= 0.663$$

Therefore, approximate heat transfer in the first two rows of pipes is,

$$Q \quad = e \,.\, m_w \,.\, c_{p,w} \,.\, (T_{exh,i} - T_{w,i}) \qquad \text{........ (6.22)}$$

$$= 4594W$$

Amount of heat energy required in preheater section to heat the incoming water to produce saturated liquid,

$$Q_{preheater} \quad = m_w \,.\, c_{p,w} \,.\, \Delta T \qquad \text{........ (6.23)}$$

$$= 0.015 \text{ x } 4204 \text{ x } (100\text{-}35)$$

$$= 4099W$$

Hence it can be concluded that the water will have reached saturated liquid stage when it exits the second row of pipes inside the heat exchanger.

Next let us consider the boiler section where the saturated liquid is supplied with latent heat and exits as saturated vapor. For the calculations, the boiler section was divided into nineteen control volumes such that the steam quality change between two subsequent control volumes is 0.05. An average heat transfer coefficient in the water stream in the boiler section was determined.

A detailed presentation of the calculation result is given in appendix D.

The procedure adopted is as presented by Lock (1996) and is presented briefly below.

Average air temperature in the boiler section $= (T_{air,in} + T_{air,out})_{boiler} /2$ .. (6.24)

$= (486°C + 141 °C) /2$

$= 313.5 °C$

Consider the water / water vapour stream

The heat flux density is separated into a forced convection component, q_{FC} and a nucleate boiling component, q_B.

The forced convection component in each control volume is calculated as follows,

$$q_{FC} = h_A (T_w - T_{sat}) , \qquad \text{........ (6.25)}$$

where,

$$h_A = \frac{k_L}{D} . \mathrm{Re}_L^{0.9} . \mathrm{Pr}_L . F(\mathrm{Re}_L , x) \qquad \text{........ (6.26)}$$

where,

$$F(\mathrm{Re}_L, x) = \frac{F_1(x)}{F_2(\mathrm{Re}_L, \mathrm{Pr}_L)} \qquad \text{........ (6.27)}$$

$$F_1(x) = 0.15[\chi + 2.\chi^{0.32}] \qquad \text{........ (6.28)}$$

where,

$$\chi = \left(\frac{\rho}{\rho}\right)^{0.5} . \left(\frac{\mu}{\mu}\right)^{0.1} . \left(\frac{x}{(1-x)}\right)^{0.9} \qquad \text{........ (6.29)}$$

and, $F_2(\mathrm{Re}_L ,\mathrm{Pr}_L)$ is calculated as follows,

$\mathrm{Re}_L > 1125$: $F_2 = 5.\mathrm{Pr}_L + 5.\ln (1+5.\mathrm{Pr}_L) + 2.5.\ln (3.1 \times 10^{-3}.\mathrm{Re}_L^{0.81})$ .... (6.30a)

$60 < \mathrm{Re}_L < 1125$: $F_2 = 5.\mathrm{Pr}_L + 5.\ln [1+\mathrm{Pr}_L (9.6 \times 10^{-2}.\mathrm{Re}_L^{0.58} -1)]$..... (6.30b)

The Prandlt number is defined by $\mathrm{Pr}_L = \nu_L / \kappa_L$ and the Reynolds number by,

$$\mathrm{Re}_L = \frac{G_D}{\mu_L}(1-x) \qquad \text{........ (6.31)}$$

Next, the nucleate boiling component of the heat flux is calculated as follows,

$$q_B = K(T_w - T_{sat})^3 \qquad \text{........ (6.32)}$$

where,

$$K = 1.89\times10^{-14}.\frac{g^{1/2}.h_{fg}.k_L.\rho_L^{17/8}.c_{cL}^{19/8}.\rho_V^{1/8}}{\sigma^{9/8}.(\rho_L-\rho_V)^{5/8}.T_{sat}^{1/8}} \qquad \text{........ (6.33)}$$

From the steam tables (Çengel and Boles, 1989), we find at 100kPa,

$$T_{sat\ @100kPa} = 100^{o}C.$$

Assume that the wall temperature of the pipe is equal to the temperature of the exhaust gases, therefore $T_w = 313.5^{o}C$.

The total heat transfer coefficient on the water side, h_w, in each control volume is calculated using the equation,

$$h_w = \frac{(q_{FC}+q_B)}{(T_w+T_{sat})} \qquad \text{........ (6.34)}$$

The average heat transfer coefficient on the water side was found to be 1869.5kW/(m^2.K).

Next consider the exhaust gas side. The heat transfer coefficient on the air side is calculated by first evaluating the properties of air at an average temperature of 313.5°C. Then the calculation is made similar to that in the preheater section and using equations 6.1 to 6.12.

The Reynolds number for the exhaust gases, using equation 6.9, is found to be 526.7, and the corresponding values obtained from figure B1, is

$$St.Pr^{2/3} = 0.013 \text{ and } f_{appi} = 0.034$$

The average heat transfer coefficient on the exhaust gas side was found to be 81.68W/(m^2.K) (see appendix D for details).

Finally using equation 6.13 with the above heat transfer coefficients calculated for three rows of pipes in the waste heat exchanger we find, if an efficiency of 65% is assumed,

The log-mean temperature difference in the boiler section is calculated as follows

$$LMTD_{boiler} = \frac{(T_{exh,out} - T_{water,in}) - (T_{exh,in} - T_{water,out})}{\ln[(T_{exh,out} - T_{water,in})/(T_{exh,in} - T_{water,out})]} \qquad \text{...... (6.35)}$$

$$= \frac{(141-100)-(486-100)}{\ln\left[\frac{(141-100)}{(486-100)}\right]}$$

$$= 154.505\text{K}$$

For the remaining 8 rows,

$$UA_{boiler} = 585.35\text{W/K}$$

Thus amount of heat transferred from exhaust gases to the water/steam side is

$$Q_{trans,boiler} = eff_{boiler} \cdot m_{exh} \cdot UA_{boiler} \cdot LMTD_{boiler} \qquad \text{........ (6.36)}$$

$$= 9594\text{W}.$$

Amount of heat transfer that takes place is less than the amount of heat required to cause complete conversion of the saturated liquid water at 100°C. The quality of the exiting liquid/vapour mixture is calculated as follows,

$$Q_{trans,boiler} = m_w \cdot x_{exit} \cdot h_{fg\ @\ 100kPa} \qquad \text{........ (6.37)}$$

$$x_{exit} = 9594/(0.02 \times 2250.10^3)$$

$$= 0.284$$

Therefore a steam quality of 0.284 should be expected at the exit of the waste heat exchanger for the given water mass flow rate of 0.02 kg/s and an exhaust mass flow rate of 0.1632kg/s and the water inlet temperature is 35°C.

Even though the calculations appeared to show that the production of superheated steam could not be achieved with a heat exchanger it was nevertheless purchased because of its low cost and also to give an insight about the criteria for a proper waste heat exchanger selection. It was found subsequently that in fact superheated steam could be achieved using the proper combination of exhaust and water mass flow rates.

6.2 Preliminary Tests on the Waste Heat Exchanger

After the waste heat exchanger was purchased preliminary tests were performed. The preliminary tests carried out was to determine the characteristics of the heat exchanger and, also to calibrate the pressure drop across the waste heat exchanger for given exhaust gas mass flow rate.

6.2.1 Test description

The test was performed in the induced draft wind tunnel in the laboratory. It is similar to the one described by Kröger (1998). Induced draft is applied on the finned side to draw air and the flow rate is determined by measuring the pressure drop across the nozzle situated within the wind tunnel. Hot water was fed into the tubes that eventually heats up the air on the finned side of the heat exchanger. For a detailed description of the test apparatus the user is referred to Kröger (1998).

These tests are also intended to give a calibration for the air flow rate. This is done by measuring the pressure drop across the heat exchanger on the air side for different mass flow rates of air.

The results obtained are presented in the next section.

6.2.2 Results for waste heat exchanger in the wind tunnel

The detailed results are presented in appendix D.

This characteristic equation of the heat exchanger is presented in graphical form in figure 6.1.

The characteristic curve for the heat exchanger was found to be

$$Ny = a\ Ry^{\ b} \text{ where } a = 455.59 \text{ and } b = 0.5833.$$

This equation is presented in graphical form in figure 6.1 below.

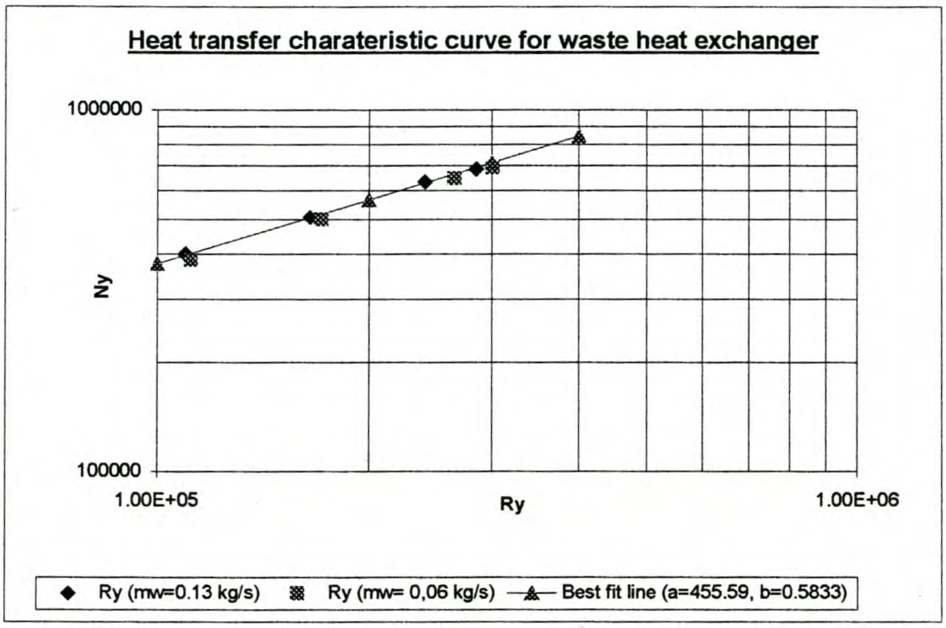

Figure 6. 1: Characteristic curve for the waste heat exchanger

The pressure drop across the heat exchanger for different mass flow rates and different Reynolds numbers are shown in figures 6.2 and 6.3 below respectively. A best fit curve was then fitted through these points and they are shown on the curve. It can noted that the graphs differ for different water mass flow rates. This might be attributed to the fact that as the mass flow rate of water decreases the heat transfer on the water side is reduced and hence the air temperature change is affected. For the test that was carried out later the pressure drop was found to be in the regions of 800Pa and the mass flow rate can be obtained using the extrapolated best fit curves as the curves are close to each for such Reynolds numbers.

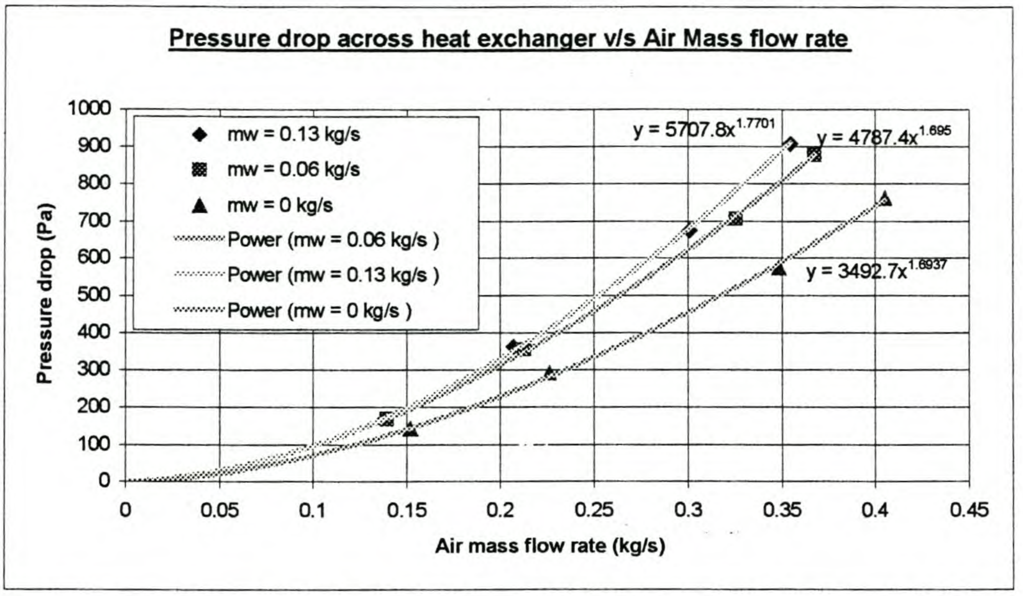

Figure 6. 2: Pressure drop on the air- side for various mass flow rate and water flow rates

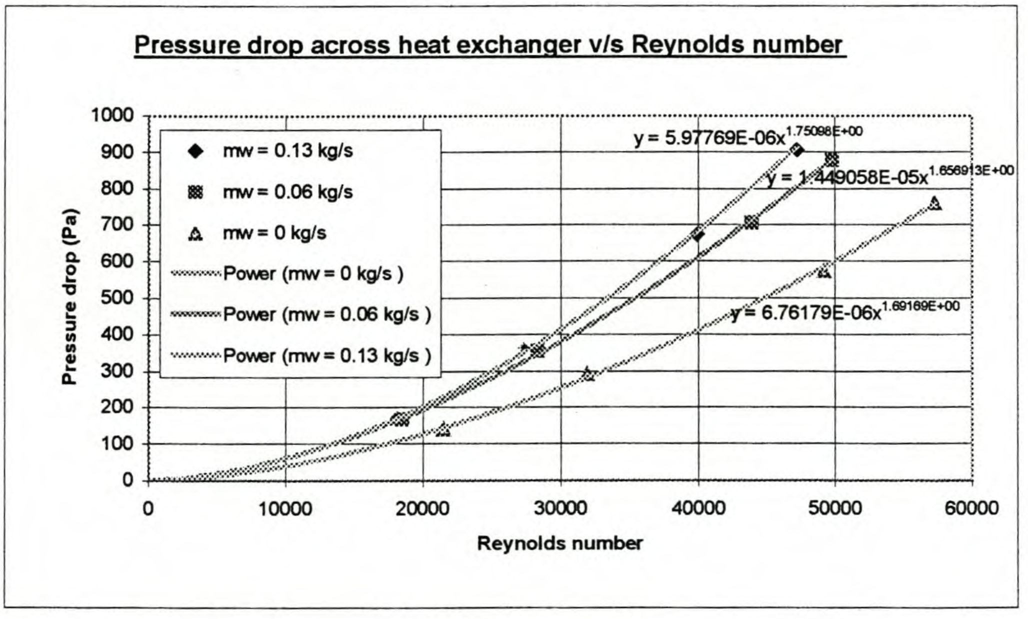

Figure 6. 3: Pressure drop on the air side v/s Reynolds number and water temperature.

It is however recommended that the graph of the pressure drop versus the Reynolds number be used for determining the mass flow rate of the exhaust gases as the

Reynolds number also considers the effect of viscosity change of air at different temperatures.

7. THE CONTINUOUS COMBUSTION UNIT

As mentioned briefly in chapter 5, the Continuous Combustion Unit was used as an alternative source for exhaust gases. The Continuous Combustion Unit can be run on both gaseous fuel and liquid fuel. However, the exhaust gas temperatures achieved by combusting propane (gaseous), which is the initial fuel that needs to be used before switching to any liquid fuel were found to be quite high. Such high temperature exhaust gases, if fed directly to the available waste heat exchanger made essentially of copper might damage the latter. Therefore, a number of varying mixture air-fuel ratios were tested to produce the desired exhaust gas temperatures to be used in the waste heat exchanger. In the following sections the procedures adopted before supplying the exhaust gases to the waste heat exchanger and the temperatures obtained using different air-fuel ratio mixtures are presented.

7.1 Procedure for Obtaining the Required Exhaust Gas Temperatures

Before switching the Continuous Combustion Unit to the diesel fuel, it first needs to be run on a gaseous fuel for at least 10 minutes to bring it to normal operating condition (P.A. Hilton Ltd., 1997). The liquid fuel is then progressively fed and the gaseous fuel is simultaneously closed. Eventually, the Continuous Combustion Unit is operated on the liquid fuel only. It was initially found that the exhaust gas temperatures were too high while accomplishing the above procedure, and cannot be fed directly into the waste heat exchanger. Therefore, initially the valve in the exhaust gas stream leading to the waste heat exchanger is closed and hence all the exhaust gases are purged directly into the atmosphere. After the correct fuel-air ratio and fuel flow rates have been set and the exhaust gas temperatures are in the desired range, the exhaust fan in the concrete combustion chamber is switched on and the valve in the exhaust gas stream is slowly opened. The low pressure in the waste heat recovery pipe network causes the exhaust gases to be diverted to the waste heat exchanger. To further reduce the temperature of the exhaust gases through the waste heat exchanger, ambient air was introduced into the exhaust gas stream to dilute and hence lower the temperature of the exhaust gases.

It will be noted that the temperatures of the exhaust gases are well below the melting point (1084°C) of copper (Callister, 1997). The reason for limiting the exhaust gas temperature was because one copper pipe inside the heat exchanger was damaged during testing and was repaired by soldering (with silver). Too high temperatures cause the weld to melt (400-500°C) and the heat exchanger to leak at the welded joints.

7.2 Settings of the Continuous Combustion Unit to Produce Exhaust Gases for Desired Operating Conditions

In this section, the temperatures obtained with different air-fuel ratios are given. The desired temperature at which the exhaust gases are to be fed into the waste heat exchanger is about 350°C. To achieve this an air-fuel (diesel) ratio of 9.35:1 is required. The air mass flow rate and the gas fuel should be set to 55kg/h and 4kg/h respectively. The temperature of the exhaust gases will be approximately 540°C. After approximately 10minutes, when the operating conditions have been reached the liquid fuel supply is progressively increased and the gaseous fuel supply progressively reduced. It takes approximately 30seconds to switch completely to the liquid fuel. The final fuel flow is 12kg/h and the air flow rate can then be adjusted to 65kg/h and the temperature reached in the exhaust gases is 550°C. After the flame is stabilized the valve leading to the waste heat recovery is slowly opened half way. (If opened completely, the low pressure due to the exhaust fan will also suck in fresh air from the main exhaust outlet of the continuous combustion unit).

The temperature of the exhaust gases into the waste heat exchanger was measured to be approximately 360°C. Heat is lost to the surrounding and to fresh air leaking through other valves (which are not air tight) within the pipe network and tapping holes for pressure and temperature measurements. In table 7.1 below, a typical flow setting is given which can be used to generate superheated steam. It is very important to maintain the same thermal input into the combustion unit when the fuel is switched from gaseous to liquid. It is also, important to let the system reach stable operating conditions. Note that the liquid fuel flow meter on the continuous combustion unit is calibrated for kerosene and to obtain the corrected flow rate for gas oil (diesel) the correction curve given in the experimental operating and maintenance manual (P.A. Hilton Ltd, 1997) should be used.

Initial gaseous fuel mass flow rate	4kg/h
Final liquid fuel mass flow rate	12kg/h
Cooling water mass flow rate	1400kg/h
Initial air mass flow rate	55kg/s
Air inlet temperature	33°C
Exhaust gas exit temperature (gaseous fuel only)	550°C
Exhaust gas exit temperature (liquid fuel only)	540°C
Cooling water inlet temperature	20°C
Cooling water exit temperature	40°C

Table7. 1:A typical setting for required exit conditions.

7.3 Operating conditions at varying air fuel ratios

In this section the determination for the fuel flow setting required for stoichiometric combustion is presented.

In figure 1.5 below the temperature variation for different air-fuel ratios is presented in graphical form.

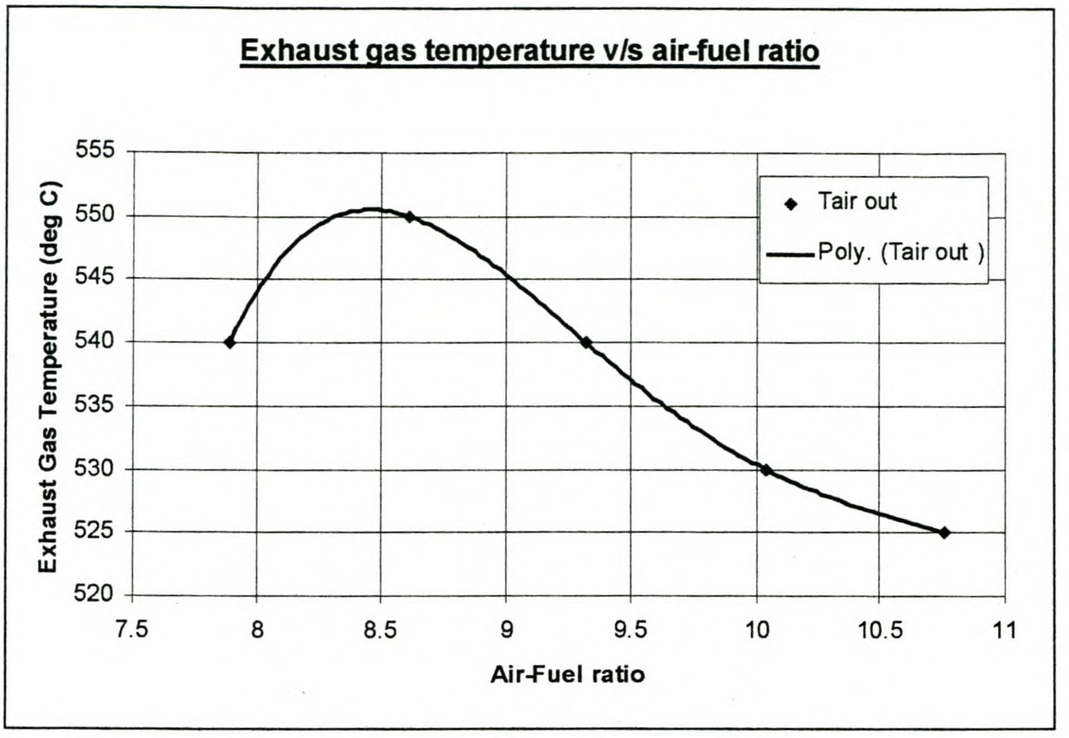

Figure 7. 1: Temperature of the exhaust gases for different air-fuel ratios

A polynomial fit was applied to determine the stoichiometric mixture. It can be seen that the stoichiometric mixture lies close to an air fuel ratio of 8.5 (highest temperature).

A quick reference guide is given in appendix I, for operating the continuous combustion unit.

It is however recommended that, the combustion is initially performed on a slightly lean mixture to ensure that the flame does not die and then if a the temperature of the exhaust gases is not enough the air flow rate can be reduced or the fuel flow rate can be increased.

8. THE INTERNAL COMBUSTION ENGINE

In this chapter the typical operating conditions of the internal combustion engine in the test cell number 7 are described. This was the only engine that was used for experimental evaluation of the waste heat recovery system.

8.1 The Internal Combustion Engine

The internal combustion engine used for experiment in this project was a C-10 series Caterpillar truck engine. The specifications of the C10 series are as presented in table 8.1 below.

Bore	125.0mm
Stroke	140.0mm
Displacement	10.3L
Aspiration	Turbocharged with ATAAC[1]
Rotation (from flywheel end)	Counterclockwise
Cooling system[2]	10.2L
Lube oil system	36L
Weight, net dry (approx) with standard equipment	932kg
Cylinder arrangement	In-line
Type of combustion	Direct injection
Firing order	1-5-3-6-2-4

Table 8. 1: Caterpillar C-10 diesel truck engine specification

[1] Air-to-Air AfterCooling.

[2] Engine only.

8.2 Readings for the C-10 Engine

During testing the Caterpillar C-10 engine was run in the test cell number 7. The readings taken on the engine are presented in this section. The exhaust gases were used in the ESL for experimental evaluation of the waste heat recovery system. The results obtained at the waste heat recovery unit are presented in chapter 9. The readings taken were temperature of the exhaust gases, inlet air temperature, the engine power delivered, the inlet and exit temperatures of the coolant liquid (50% water and 50% coolant fluid). Other readings were the fuel consumption, the power delivered and pressures at various points. These are summarized in table 8.2 below.

Speed	1214rpm
Torque	990.94Nm
Fuel flow rate	28.46kg/h
TEMPERATURE	
Inlet air	24.65°C
Air before intercooler	89.51°C
Air after intercooler	47.99°C
Coolant in	86.73°C
Coolant out	91.66°C
Exhaust gas before turbocharger	611.22°C
Exhaust gas post turbocharger	559.53°C
PRESSURE	
Ambient pressure	100.6kPa
Gauge pressure after booster (air intake)	66.93kPa

Table 8.2: Operating conditions on Caterpillar C-10 diesel truck engine

8.3 Determining the Power Output and the Mass Flow Rate of Air.

Power out (kW) of the engine is given by,

$$P_{out} = \frac{2\pi.N.T}{60.1000} \qquad .....(8.1)$$

where N = Speed of rotation (rpm) and T = Torque (Nm) Substituting the values for T and N from table 8.2 in the above equation we find, P_{out} = 125.9kW. The total mass flow rate of exhaust gases air was found to be 0.0188kg/s per litre capacity of engine (see appendix A). Therefore, for the 10.3 litre capacity engine,

$$\text{Total exhaust gas mass flow rate} = 0.0188 \times 10.3 = 0.18722\text{kg/s} \qquad .....(8.2)$$

8.4 Power Balance on the Engine

Calorific value of fuel = 42780kJ/kg

$$\text{Power from fuel} = \dot{m}_{fuel} \,.\, \text{Calorific value} = (28.46 / 3600) \,.\, 42780 = 338.20\text{kW}. \qquad .....(8.3)$$

Energy loss rate in exhaust gases,

$$Q_{exh} = \dot{m}_{exh} \,.\, c_{pexh}{}^{1} \,.\, (T_{exh\ out} - T_{air\ in}) = 0.18722 \,.\, 1.07\,(611.22 - 47.99) = 112.83\text{kW} \qquad .....(8.4)$$

---

[1] $c_{p\,exh}$ is obtained from appendix E

The coolant as mentioned above is mixture of coolant liquid which is Ethylene glycol and water. Water and ethylene glycol are mixed in a ratio of 1:1. However the amount of energy lost in the cooling water was not determined separately as this would require implementing a coolant mass flow measurement instrument (standard nozzle). Thus the energy transferred to the coolant and other losses are treated as a single unit. Also, the main interest in this project is to determine the amount of waste energy extracted from the exhaust gases.

Energy to the coolant and other losses in the system,

$$Q_{other} = \text{Total energy from fuel} - (\text{Energy in exhaust gases} + \text{Engine power output}) \quad \text{.....(8.5)}$$

$$= 338.20 - (112.83 + 125.9)$$

$$= 99.47\text{kW}$$

The energy distribution in the Caterpillar C-10 truck diesel engine is summarized in figure 8.1 below.

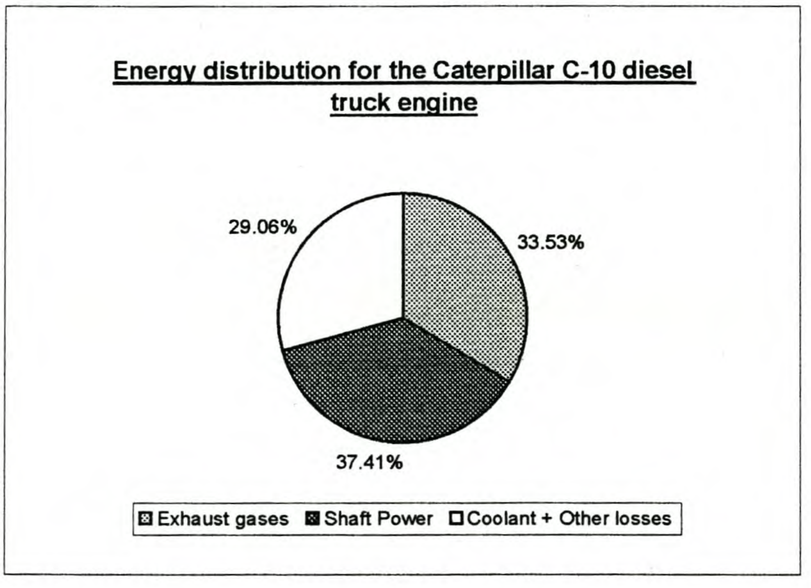

Figure 8.1: Power distribution for the Caterpillar C-10 diesel truck engine

The exhaust gases were used to perform an experimental evaluation of the waste heat exchanger and the results are presented in the next chapter.

9. PRESENTATION AND DISCUSSION OF RESULTS

In this chapter, the results obtained in the final stage of this project are presented. These results should give an indication of whether the experimental work does in fact reflect what was expected from the theoretical calculations and hence indicate whether it is possible to extract enough waste heat from the exhaust gases for the production of electrical power.

Two sets of complete results are presented in the next section. For the first set of results, exhaust gases from the continuous combustion unit were used and for the second set of readings the exhaust gases were from the internal combustion engine.

Due to shortage of time further complete sets of test results could not be obtained.

9.1 Presentation of Results

The results for the experiment with the exhaust gases originating from the continuous combustion unit are presented in section 9.1.1 and the results obtained with the exhaust gases from the internal combustion engine are presented in section 9.1.2.

For the case of exhaust gases being generated from the continuous combustion unit the settings on the continuous combustion unit is also included to the readings at the waste heat recovery unit. For the second set of results only the reading at the waste heat recovery unit is presented. The engine settings are presented in chapter 6.

9.1.1 Exhaust gases from the continuous combustion unit

The settings and readings of the continuous combustion unit are presented in table 9.1 below and the readings at the waste heat recovery apparatus is presented in table 9.2.

Water	
Mass flow rate	1400kg/h
Inlet temperature	20°C
Exit temperature	36°C
Air / Exhaust gases	
Air inlet mass flow rate	65kg/h
Air inlet temperature	35°C
Exhaust gas exit temperature	540°C
Fuel (Diesel)	
Indicated mass flow rate	12kg/h
Corrected mass flow rate	7.11kg/h

Table 9.1 : Settings and readings on the continuous combustion unit

Water	
Inlet Mass flow rate	0.004kg/s
Water inlet temperature	36°C
Steam exit temperature	160°C
Mass of condensate	None
Exhaust gases	
Inlet temperature	360°C
Exit temperature	168°C
Pressure drop	20Pa

Table 9. 2 : Readings at the waste heat recovery apparatus (exhaust gases from continuous combustion unit)

9.1.2 Exhaust gases from the internal combustion engine

The reader is referred to section 8.2 for the engine settings and running conditions during the experiment. The readings taken at the waste heat recovery unit is presented in table 9.3 below.

Water	
Inlet Mass flow rate	0.006kg/s
Water inlet temperature	21°C
Steam exit temperature	157°C
Mass of condensate	None
Exhaust gases	
Inlet temperature	212°C
Exit temperature	189°C
Pressure drop	270Pa

Table 9. 3 : Readings at the waste heat recovery apparatus (exhaust gases from IC engine)

9.2 Analysis and Discussion of Results

9.2.1 Exhaust gases from continuous combustion unit

Determining the mass flow rate of exhaust gases

For a pressure drop of 20Pa, corresponding Reynolds number = 7000 (obtained from figure 6.3)

Average exhaust gas temperature, $T_{exh,av}$ = (360 + 170)/2 ..... (9.1)

= 265°C

Viscosity of air at 265°C, $\mu_{exh} = 2.82 \times 10^{-5}$ Pa.s

Air mass flow rate is given by the following equation derived by substituting, $\dot{m}_{exh} = \rho\ A\ v$ into equation (B.2), such that,

$$\dot{m}_{exh} = (\pi.Re_{exh}.d_{hyd}.\mu_{exh})/4, \qquad \text{..... (9.2)}$$

where, d_{hyh} = 3.38×10^{-3}m (calculated from the geometry of the heat exchanger in appendix D)

Substituting the values in the above equation we find, $\dot{m}_{exh} = 0.055$ kg/s.

Performing an energy balance per unit time, we have,

Heat gained by water stream,

$$\begin{aligned} Q_{water} &= \dot{m}_w .(c_{pw}\Delta T_w + h_{out} - h_{in}) \qquad \text{..... (9.3)} \\ &= 0.0042 \times (4.205 \times (100\text{-}36) + 2798.169 - 419.04) \\ &= 11.12\text{kW} \end{aligned}$$

Heat lost by exhaust gas stream,

$$\begin{aligned} Q_{exh} &= \dot{m}_{exh} .\ c_{pexh} .\Delta T_{exh} \qquad \text{..... (9.4)} \\ &= 0.055\ .\ 1.07\ .\ (360\text{-}168) \\ &= 11.24\text{kW} \end{aligned}$$

Efficiency of heat transfer, E $= 11.12/11.24\ .\ 100$
$= 98.93\%$

9.2.2 Exhaust gases from the IC engine

For a pressure drop of 270Pa, we read from chart 6.3, corresponding Reynolds number is 31200.

Average exhaust gas temperature, $T_{exh,av}$ $= (212 + 189)/2$ ..... (9.5)
$= 200.5°C$

Viscosity of air at 200.5°C, μ_{exh} = 2.58×10^{-5} Pa.s

Using equation (9.2) again we find the mass flow rate of exhaust gases,

$$\dot{m}_{exh} = 0.22\text{kg/s.}$$

Therefore, the rate of energy lost by the exhaust gases (using 9.4),

$$Q_{exh} = 0.22 \times 1.07 \times (212 - 133.4)$$
$$= 18.50\text{kW}$$

Rate of energy gained by water inside waste heat exchanger (using 9.3),

$$Q_w = 0.0061\ .(\ 4.205\ (100\text{-}21) + 2790.25 - 419.04)$$
$$= 16.49\text{kW}$$

Efficiency of heat transfer E = (16.46/18.50) .100
= 88.97 %

It should be noted that the delivery temperature of the exhaust gases at the engine was nearly 560°C but it degraded to 212°C at the waste heat recovery unit due to the long sections of bare pipes.

The amount of energy extracted as a percentage of fuel consumed is calculated in the next section.

The product of the rate of fuel consumption and the calorific value gives rate at which energy is given off as a result of fuel combustion.

Therefore,

Rate at which energy is being given off = $\dot{m}_{fuel}$. Calorific value
= (28.46 / 3600) . 42780
= 338.2kW.

Fraction of fuel energy supply rate recovered in the waste heat exchanger unit is given by the ratio of the rate at which energy is transferred to the water stream in the waste heat recovery unit to the rate at which energy is given off as a result of fuel combustion.

Therefore,

$$\text{Percentage of heat recovered} = (16.49 / 338.2) \cdot 100 = 4.9\%$$

9.3 Comments

It can be seen form the results that superheated steam was produced in both cases. Therefore the main aim of this project that included the production of superheated steam was achieved.

It can also be noticed that the heat exchange between the exhaust gas stream and the water stream was very good as heat was transferred with efficiencies of 98.93% and 88.97% for the case of exhaust gases generated from the continuous combustion unit and the internal combustion engine respectively.

The valves inside the concrete exhaust gas chamber are not air-tight. Therefore, some of the exhaust gases do leak out in the case of exhaust gases being produced by the IC engine and fresh air is sucked in when the exhaust gases from the continuous combustion unit are used.

It is also important to mention that if the exhaust gas supply is closer to the waste heat recovery unit, a better quality of exhaust gases will be available from a temperature point of view.

Finally it should be noted that the water mass flow rates used in the above presented cases are not the optimal flow rates. It is still possible to have higher mass flow rates however, a compromise will then have to be reached between the degree of superheated steam produced and the desired water mass flow rate.

The percentage of heat recovered is presented in figure 9.1 below.

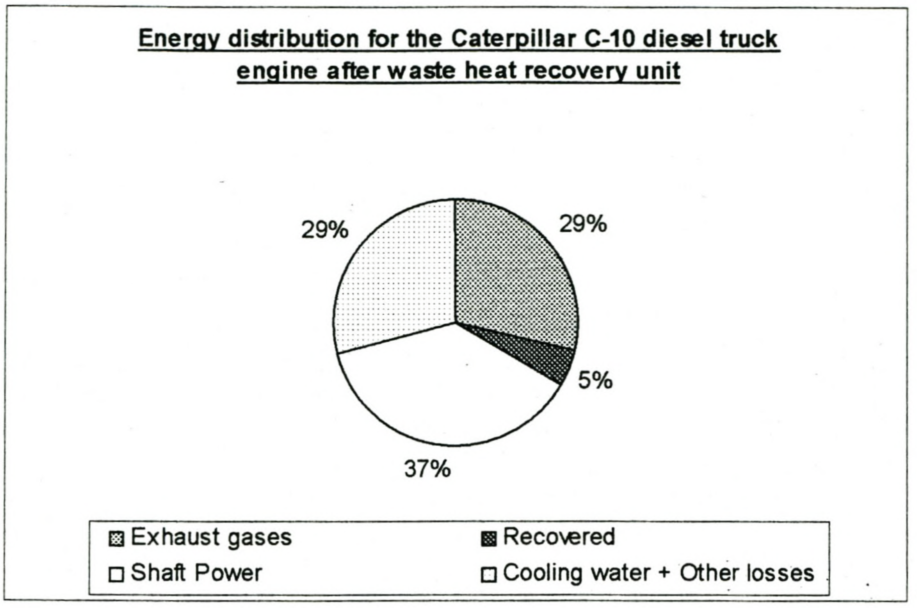

Figure 9. 1 : Power distribution for the Caterpillar truck engine after waste heat recovery

10. CONCLUSIONS

The first step in the project was to carry out preliminary studies to determine the feasibility of a small scale waste heat recover project. A spreadsheet program was written to determine the feasibility of the project. It was found that there is adequate energy which can be extracted from a typical diesel truck engine exhaust to generate superheated steam to operate a Rankine cycle, and hence, generate useful mechanical energy and therefore increase the thermal efficiency of the fuel used. The preliminary calculations were performed on the results obtained from a 6litre diesel powered engine.

In this project, one of the main objectives was to set up a waste heat recovery test facility and that included a system of pipe network from the exhaust gas outlet pipes from the *Centre for Automotive Engineering* test cells to the *Energy System Laboratory*. This was successfully designed, built and implemented. A valve control system was required to control the flow of the exhaust gases, i.e. it should be able to either divert the exhaust gases to the waste heat recovery exhaust gas pipe network or to purge the gases directly into the concrete exhaust duct. This was achieved by designing simple disc in pipe valves, which were interlinked such that at any time at least one valve is open and hence no exhaust gas is trapped inside the pipes.

An auxiliary source of exhaust gases was also coupled to the exhaust gas pipe network of the waste heat recovery system. The auxiliary source was a continuous combustion unit, which was commissioned by Schwack (2000) to be operated on gaseous or liquid fuel for this project. The liquid fuel used was gas oil (diesel). This included a fuel supply line and a fuel tank that needed to be connected to the Continuous Combustion Unit.

Various heat exchangers were investigated for the extraction of waste heat from the exhaust gases. Three main heat exchanger arrangements were investigated and finally a conventional copper-finned copper pipe cross-flow heat exchanger was selected and purchased. The model number as per its manufacturer is 3SC-254/254/10FS/10R Cu/Cu/SS.

The waste heat exchanger was then fitted with transition pieces to allow same to be implemented into the exhaust gas pipe network. The unit (waste heat exchanger and transition pieces) was then tested and calibrated for pressure drop and heat transfer in the wind tunnel. The characteristic curve of the waste heat exchanger was then determined. It was found that the characteristic curve for the waste heat exchanger is given by $Ny = a\ Ry^b$. This can be used for future tests. The pressure drop at varying Reynolds numbers were plotted and were used to determine the air flow rate through the waste heat exchanger in the final waste heat recovery set up.

The heat exchanger unit was then installed in the waste heat recovery apparatus. A water supply and steam separator was then added onto the waste heat recovery system to complete the waste heat extraction unit.

Experimental results were performed using exhaust gases from a continuous combustion unit and from a 10litre diesel truck engine, which was then available for testing. Superheated steam was obtained in both cases. In the case of exhaust gases originating from the continuous combustion unit, exhaust gases were supplied at a temperature of 360°C and superheated steam at 160°C was obtained with a heat transfer efficiency of 98.93%. For exhaust gases from the internal combustion engine, the exhaust gas supply temperature at the waste heat exchanger was 212°C and superheated steam at 157°C was produced with a heat transfer efficiency of 88.97%.

Therefore, it can be concluded that the aims of the project were achieved and the results show that there is ample room for further research in the field of waste heat recovery on small-scale level and optimization of the experimental apparatus already set up in the *Energy Systems Laboratory*.

11. RECOMMENDATIONS

Based on the conclusions and findings of the project the following recommendations can be made:

- The waste heat exchanger can be further optimized and a more efficient one selected. Examples of other types of heat exchangers that can be investigated are multiple pass or plate heat exchangers. Also a longer heat exchanger as opposed to the presently wider heat exchanger can also be investigated. It is suspected that with a longer heat exchanger there will be even better room for the heat to be transferred on the air side.

- The properties of the exhaust gases were assumed to be the same as that of air at the same temperature in the design phase of the heat exchanger. Further research in the exhaust gas composition and properties should be carried out and the use of an exhaust gas composition analyzer would be very advantageous as it will enable the composition of the exhaust gases to be known precisely.

- The waterside of the waste heat exchanger should be pressurized to obtain a better heat transfer and also a higher energy content of the exiting steam. This would require a more enhanced design, meeting the relevant safety standards.

- The pinch point is the main limiting factor for pressure that can be applied to the water used inside the waste heat exchanger. The use of alternative working fluid should also be considered. The use of organic Rankine cycle is a very attractive alternative and a wide variety of information can be obtained in the literature.

- More precise measurement apparatus should also be employed to obtain better understanding of the energy transfer within the system. These include a continuous temperature measurement of the water and exhaust gases during experimentation. An increased number of thermocouples and pressure transducers usage with continuous data logging into a PC is also recommended.

- Insulations on the pipe network would further heat losses from the relatively long pipe sections.

- Fouling effect of the waste heat exchanger can be reduced by using a dust /ash filter before the waste heat exchanger.

- More tests should be performed using exhaust gases from the test engines, as they are capable of delivering higher volumes of exhaust gases than the continuous combustion unit.

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APPENDIX A

ENGINE PARAMETERS FOR THE 6 LITRE AND 10 LITRE DIESEL ENGINES

A.1 Engine Parameters for the 6l Diesel Engine (Koorts, 1998)

The engine parameters of the 6l diesel engine investigated in this proposal report are as follows(Koorts, 1998):

Number of cylinders	6
Total displacement	5987 cc
Average exit temperature	560 °C
Air-Fuel Ratio	14.7 : 1
Exhaust gas pressure	8 kPa gauge

Table A.1 : Design engine parameters

Calculating the intake mass flow rates

(Per litre capacity and at an engine speed of 2600 rpm)

Mass of air intake , $m = pV/RT$ .....(A.1)

$= (101300 * 0.001)/(287 * 300)$

$= 0.001176$ kg per litre capacity and revolution per second

Therefore at a speed of 2600 rpm and a capacity of 5.987 litres we have,

Total air intake $= (0.001176 \cdot 5.987 \cdot 2600)/(60 \cdot 2)$ .....(A.2)

$= 0.1526$ kg/s

Adding the fuel intake at an air to fuel ratio of 14.7 : 1 total intake of air and fuel can be calculated as,

Total intake mass flow rate $= 0.1526\,(14.7 + 1)/14.7$ .....(A.3)

$= 0.1632$ kg/s

Therefore applying conservation of mass we obtain the exhaust gas mass flow rate to be 0.1632kg/s.

A.2 Engine Parameters for the 10l Diesel Engine

The parameters of the engine used for the test performed on the waste heat recovery apparatus are given in tables 8.1 and 8.2.

Applying the same procedure as in section A.1 we find the exhaust gas mass flow rate to be 0.1872 kg/s.

APPENDIX B

GRAPHS AND COMMON EQUATIONS

In this appendix the common equations and graphs used for preliminary calculations are presented.

Reynolds number

Flow in a circular pipe: $Re = \rho V d/\mu$ (Sayers, 1992) .....(B.1)

Flow in any other geometry: $Re = \rho V d_{hyd}/\mu$ (Sayers, 1992) .....(B.2)

where, d_{hyd}, the hydraulic diameter is given by

d_{hyd}= 4 * Cross sectional area/Wetted Perimeter (Kays and Crawford, 1993) ....(B.3)

Prandlt Number

$Pr = \mu c/k$ .....(B.4)

Nusselt number

By definition: $Nu = hd/k$ (Kays and Crawford, 1993) .....(B.5)

For a turbulent flow:

$$Nu = \frac{(f_D/8)(Re-1000)Pr\left(1+(d/L)^{0.67}\right)}{1+1.27(f_D/8)^{0.5}\left(Pr^{0.67}-1\right)}$$ (Kröger, 1998) .....(B.6)

where f_D, the friction factor is given by,

$f_D = (1.82 \log_{10}Re - 1.64)^{-2}$ (Kröger, 1998) .....(B.7)

for the following ranges , $2300 < Re < 10^6$, $0.5 < Pr < 10^4$, and $0 < d/L < 1$.

Properties of air were determined using the following equations (Kröger, 1998)

Specific heat capacity

$$c_{pa} = 1.045356 \times 10^{3} - 3.161783 \times 10^{-1}\ T + 7.083814 \times 10^{-4}\ T^2 - 2.705209 \times 10^{-7}\ T^3 \quad \text{(J/kgK)} \tag{B.8}$$

Dynamic viscosity

$$\mu_a = 2.287973 \times 10^{-6} + 6.259793 \times 10^{-8}\ T - 3.131956 \times 10^{-11}\ T^2 + 8.15038 \times 10^{-15}\ T^3 \quad \text{(kg/m.s)} \tag{B.9}$$

Thermal conductivity

$$k_a = -4.937787 \times 10^{-4} + 1.018087 \times 10^{-4}\ T - 4.627937 \times 10^{-8}\ T^2 + 1.250603 \times 10^{-11}\ T^3 \quad \text{(W/mK)} \tag{B.10}$$

Properties of water were determined using the following equations (Kröger, 1998)

Specific heat capacity

$$c_{pw} = 8.15599 \times 10^{3} - 2.80627 \times 10\ \ T + 5.11283 \times 10^{-2}\ T^2 - 2.17582 \times 10^{-13}\ T^3 \quad \text{(J/kgK)} \tag{B.11}$$

Dynamic viscosity

$$\mu_a = 2.414 \times 10^{-5} \times 10^{-(247.8/T-140)} \quad \text{(kg/m.s)} \tag{B.12}$$

Thermal conductivity

$$k_a = -6.14255 \times 10^{-1} + 6.9962 \times 10^{-3}\ T - 1.01075 \times 10^{-5}\ T^2 + 4.74737 \times 10^{-12}\ T^4 \quad \text{(W/mK)} \tag{B.13}$$

APPENDIX C

CALCULATION OF PARAMETERS FOR RANKINE CYCLE AT VARYING PRESSURES

C.1 Procedure for Calculating the Various Parameters of the Rankine cycle at Different Pressures

In this appendix the equations used for the calculation of the parameters are given. The results are also presented in section 5.4 of the main report.

The specific volumes and the enthalpy values were read directly from tables (Cengel and Boles, 1983).

Pump efficiency is chosen to be 0.7 and the turbine efficiency 0.85, based on the efficiencies of commercially available pumps and turbines.

The low pressure (before pump) $= P_1$ ………(C.1)

High pressure (after pump) $= P_2$ ………(C.2)

Work done by pump, $w_{pump} = v_1(P_2-P_1) / \eta_{Pump}$ ………(C.3)

Enthalpy of fluid after pump, $h_{2'} = h_1 + w_{pump}$ ………(C.4)

Next, the highest temperature reached by superheated steam or the liquid/vapour mixture is determined.

The notations used are shown in the diagram below.

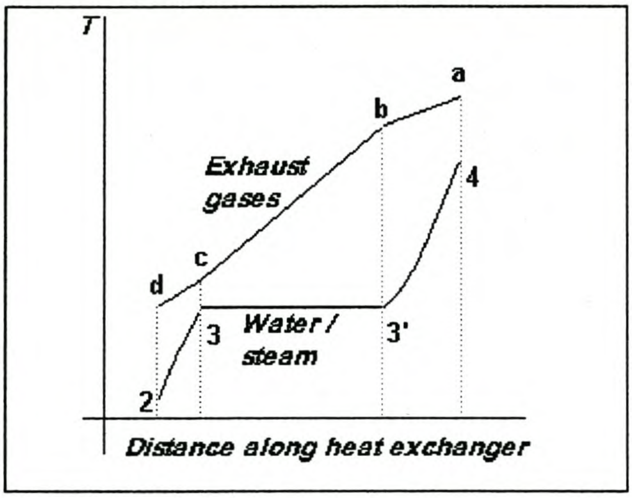


Figure C.1: Exhaust gases and water /steam temperatures along waste heat exchanger

Subscripts a, b, c, and d are used for the exhaust gas stream such that T_a and T_b are the inlet and exit temperatures of the exhaust gas in the superheater section, T_b and T_c are the inlet and exit temperatures of the exhaust gas in the boiler section and T_c and T_d are the inlet and exit temperatures of the exhaust gas in the preheater section of the waster heat exchanger. The notation used for the water stream are numbers 1 to 4 such that 1 is at the preheater inlet, and 4 is at the superheater exit of the waste heat exchanger. The temperatures were calculated with the aid of an Microsoft Excel (Version 7.0) spreadsheet program using the method described below.

Assumed exhaust gas exit temperature from the preheater section is 100°C (T_d). Also assume an initial water mass flow rates of 0.02 kg/s. The exhaust gas mass flow rate as shown is 0.1632 kg/s as found from the specifications of 6-litre diesel engine in appendix A.

Water enters the preheater section at a temperature of 35°C (T_2) and leaves as saturated liquid.

Performing an energy balance on the preheater section we have,

$$m_w . c_{p,w} . (T_3 - T_2) = \eta_{preheater} . m_{exh} . c_{p,exh} (T_c - T_d) \qquad \text{.........(C.5)}$$

Now $T_3 = T_{3'} = T_{sat@\,P}$,

Therefore

$$T_c = \frac{m_w . c_{p,w} (T_3 - T_2)}{\eta_{preheater} . m_{exh} . c_{p,exh}} + T_d \qquad \text{.........(C.6)}$$

Next, we ensure that the amount of heat that can be extracted from the exhaust gases is enough to evaporate all the saturated liquid.

Amount of energy available from exhaust gases,

$$Q_{exh,\,remaining} = \eta_{boiler} . m_{exh} . c_{p,exh} (T_a - T_c) \qquad \text{.........(C.7)}$$

Amount of energy required to convert the saturated liquid to saturated vapour,

$$Q_{w,\ required} = m_{w}.\ (h_3 - h_2) \qquad \text{………(C.8)}$$

Now, if $Q_{exh\ ,\ remaining} > Q_{w,\ required}$

Performing a heat balance on the boiler section we have,

$$m_{w}.\ (h_{3'} - h_3) = \eta_{boiler}\ .m_{exh}.\ c_{p,exh}\ (T_b - T_c) \qquad \text{……(C.9)}$$

such that the inlet exhaust gas temperature in the boiler section will be,

$$T_b = \frac{m_w.(h_{3'} - h_3)}{\eta_{spheater}.m_{exh}.c_{p,exh}} + T_c \qquad \text{………(C.10)}$$

Finally, an energy balance can be applied to the superheater section to calculate highest superheated temperature reached as follows,

$$m_{w}.\ (h_4 - h_3) = \eta_{sprheater}\ .m_{exh}.\ c_{p,exh}\ (T_a - T_b) \qquad \text{………(C.11a)}$$

Rewriting with h_4 as subject of formula we have,

$$h_4 = \frac{\eta_{sprheater}.m_{exh}.c_{p,exh}(T_a - T_b)}{m_w} + h_3 \qquad \text{………(C.11b)}$$

Having obtained the enthalpy of the superheated steam for each pressure the respective superheated steam temperature, T_4 can be looked up from superheated steam table (Çengel and Boles, 1983).

Otherwise, if the amount of energy available is not enough to provide the required latent heat to the water stream, i.e. if $Q_{exh\ ,\ remaining} > Q_{w,\ required}$, we can calculate the steam quality, x leaving the boiler section as follows:

Setting up a heat balance we have,

$$m_{w}.\ .x.\ (h_3 - h_2) = \eta_{boiler}\ .m_{exh}.\ c_{p,exh}\ (T_a - T_c) \qquad \text{………(C.12a)}$$

or putting x as the subject of formula we can rewrite the above as,

$$x = \frac{\eta_{boiler}.m_{exh}.c_{p,exh}.(T_a - T_c)}{m_w.(h_{3'} - h_3)} \qquad \text{.........(C.12b)}$$

The maximum temperature, T_4 then reached will be the saturation temperature at that pressure.

Now $s_5 = s_4$, therefore,

Work out from turbine, $w_{turb} = \eta_{turb} (h_4 - h_5)$ ....(C.13)

Therefore, actual enthalpy of steam out of turbine,

$$h_{5'} = h_4 - w_{turb} \qquad \text{.........(C.14)}$$

Performing an energy balance on the waste heat exchanger,

Heat lost by exhaust gas = Heat gained by water

The net work output of the system, $w_{nett} = w_{turb} - w_{pump}$ .... (C.15)

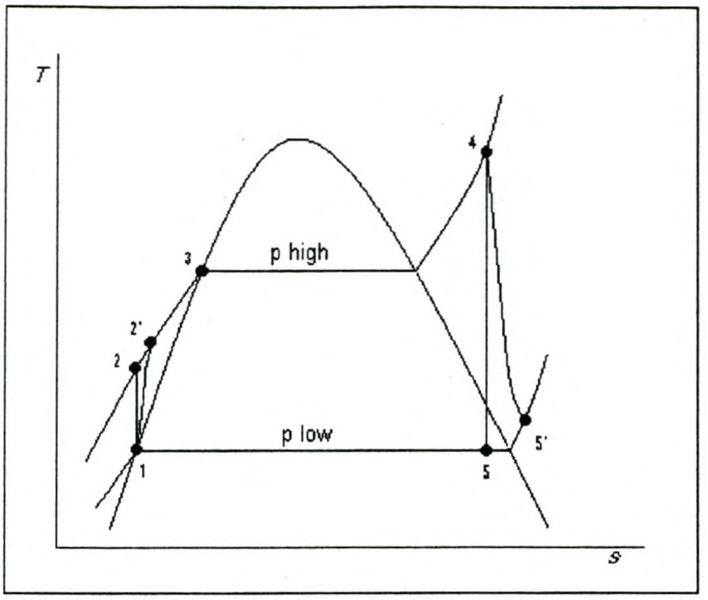
Finally the temperature difference, ΔT_{pinch} at the pinch point and the cycle efficiency are calculated as follows,

$$\Delta T_{pinch} = T_c - T_2, \text{ and ,} \qquad \text{.......... (C.16)}$$

$$\text{Cycle efficiency} = w_{nett} / \text{Heat recovered} \qquad \text{.......... (C.17)}$$

The numbering notation used for calculations in the Rankine cycle diagram of the above-mentioned parameters, at different pressures are shown below.

The preliminary results obtained are presented in the next section. It can be seen from the results obtained, that, for reasonable heat transfer to take place a pressure of about 800kPa is required when the pinch point temperature difference is approximately 10°C.



FigureC.2: The Rankine cycle

C.2 Values of Parameters at Different Pressures

The fixed parameters are the parameters that are selected to remain constant or to be achieved. The variable parameters are the parameters that change at different operating pressures.

Fixed parameters

WATER SIDE

$T_{w,low} = T_2$	35°C
$p_{w,low}$	5.63kPa
c_{pw}	4.18kJ/(kg.K)
Specific vol	1.01E-03m^3/kg
H_1	146.68kJ/kg

Efficiencies

Pump, η_{pump}	0.7
Turbine, η_{turb}	0.85
Preheater, $\eta_{preheater}$	0.6
Boiler, η_{boiler}	0.6
Superheater, $\eta_{sprheater}$	0.6

EXHAUST GAS SIDE

m_{ex}	0.1632kg/s
c_{pex}	1.07kJ/(kg.K)
$T_{ex,high} = T_a$	493°C
$T_{ex,low} = T_d$	100°C

Table C.1: Fixed parameters in the preliminary calculations

Variable parameters

Property	Unit	Parameters			
$p_{w,high}$	kPa	500	600	700	800
m_w	kg/s	0.015	0.015	0.015	0.015
w_{pump}	KJ/kg	0.713	0.858	1.002	1.146
$h_{2'}$	KJ/kg	147.39	147.54	147.68	147.83
h_3	kJ/kg	640.23	670.56	697.22	721.11
$h_{3'}$	kJ/kg	2748.7	2756.8	2763.5	2769.1
T_3	°C	151.86	158.85	164.97	170.43
T_c	°C	169.93	174.12	177.78	181.05
$Q_{exh,remaining}$	kJ	33.85	33.41	33.03	32.68
$Q_{w,required}$	kJ	31.63	31.29	30.99	30.72
T_b	°C	471.79	472.79	473.60	474.25
h_4	kJ/kg	2896.84	2897.96	2899.03	2900.10
T_4	°C	219.68	222.34	224.94	227.46
s_4	kJ/kg	7.1425	7.0626	7.0114	6.9380
h_5	kJ/kg	2190.00	2150.00	2140.00	2120.00
w_{turb}	kJ/kg	600.82	635.76	645.18	663.09
$h_{5'}$	kJ/kg	2296.03	2262.19	2253.86	2237.02
w_{nett}	kJ/kg	600.10	634.91	644.18	661.94
Net power out	kW	9.00	9.52	9.66	9.93
ΔT_{pinch}	°C	18.07	15.27	12.81	10.62
Heat Recovered	kJ/kg	2749.45	2750.42	2751.35	2752.28
Cycle efficiency	%	21.83	23.08	23.41	24.05

Table C.2: Values of variable parameters from preliminary calculations.

These results are also presented in graphical form in figures 4.1 to 4.3.

APPENDIX D

WASTE HEAT EXCHANGER CALCULATIONS

D.1 Introduction

In this appendix the major three heat exchangers that were investigated are briefly described and a detailed calculation of the final chosen waste heat exchanger is presented.

The major three heat exchangers that were investigated are as follows,

1. A three stages waste heat exchanger vertically mounted on each other as proposed by Koorts (1998). The three units consist of a preheater, a boiler and a superheater.
2. A three stages unit with a shell and tube arrangement for the preheater.
3. A finned tube heat exchanger incorporating the preheater, boiler and the superheater in one unit, such that subcooled liquid enters the unit and leaves as superheated steam.

In the calculations the common values used for the parameters are as follows

Water mass flow rate, m_w = 0.015kg/s (appendix B)
Exhaust gas mass flow rate, m_{exh} = 0.1630 kg/s (appendix A)
Pressure on water side, P_{high} = 800kPa (appendix B)

Section in waste heat exchanger	*Preheater*	*Boiler*	*Superheater*
Inlet water temperature	35.0°C	170.43 °C	170.43 °C
Exit water temperature	170.43 °C	170.43 °C	227.46 °C
Inlet exhaust gas temperature	181.05 °C	474.25 °C	493.0 °C
Exit exhaust gas temperature	100.0 °C	181.05 °C	474.25 °C
Average water /steam temperature	102.72°C	170.43 °C	198.94 °C
Average exhaust gas temperature	140.53 °C	322.34 °C	483.62 °C

Table D. 1: Temperatures of the fluids from each section of the waste heat exchanger

D.2 The Three Stages (Counterflow) Waste Heat Exchanger

D.2.1 The preheater section

Mean water inlet temperature, $T_{w,av,pr}$ = 102.72°C

Properties of water at mean temperature of 102.72°C.

Thermal conductivity, $k_{w,pr}$ = 0.682192 W/(m^2.K)
Viscosity, $\mu_{w,pr}$ = 2.71216 x 10^{-4} Pa.s
Specific heat capacity, $c_{p,w,pr}$ = 4217.8 kJ/(kg.K)

Mean exhaust gas temperature, $T_{exh,av,pr}$ = 140.53 °C
Properties of exhaust gas at mean temperature of 140.53 °C, assuming it to be te same as that of dry air at the same temperature.

Thermal conductivity, $k_{exh,pr}$	34.588 x 10^{-3} W/(m^2.K)
Viscosity, $\mu_{exh,pr}$	2.33945 x 10^{-5} Pa.s
Specific heat capacity, $c_{p,exh,pr}$	1016.7 kJ/(kg.K)

Table D.2: Properties of water at mean temperature of 102°C

Heat exchanger specifications

Simple counterflow arrangement (Annular pipe arrangement)

Tube internal diameter, $d_{t,inner}$	101.6mm (4')
Tube thickness, t_1	2mm
Material	Mild Steel
Conductivity, k_{steel}	44 W/(m.K)
Shell internal diameter, $d_{s,inner}$	127.0 mm (5')

Table D.3: Preheater geometrical specification

Consider the exhaust gas side (Tube side),

Using equation B.1 , we find

$Re_{exh,pr} = 887422.5$, therefore the flow is turbulent

Therefore, using equations B.7 and B.6 respectively, we have,

$f_D = 0.018$ and $Nu = 124.6$

Using equation B.5, we calculate the heat transfer coefficient on the exhaust gas side to be,

$h_{exh} = \underline{42.42\ W/(m^2.K)}$

Next, consider the water side

The hydraulic diameter is given by

$$
\begin{aligned}
d_{hyd} &= \text{Inner diameter of shell} - \text{Outer diameter of inner tube} \\
&= (127 - 105.16)\ .\ 10^{-3}\ m. \\
&= 0.021\ m
\end{aligned}
$$

Substituting , d_{hyd} in equation B.2 we have,

$Re_w = 3353$,

The flow is again turbulent therefore, using equations B.7 and B.6 again we find,

$f_D = 0.044$ and $Nu = 14.4$

The heat transfer coefficient on the water side can be computed using B.5,

$h_w = \underline{116.42\ W/(m^2.K)}$

For the simple counterflow arrangement we have, per unit length of heat exchanger, For the simple counterflow arrangement we have, per unit length of heat exchanger,

$$UA = \left[\left(\frac{1}{\pi}\right)\cdot\left(\frac{1}{h_{exh}.d_{t,inner}} + \frac{\ln\left(\frac{d_{t,inner} + 2t_{pipe}}{d_{t,inner}}\right)}{2.k_{steel}} + \frac{1}{h_w.\left(d_{t,inner_1} + 2t_{pipe}\right)}\right)\right]^{-1} \quad \text{....(D.1)}$$

Substituting the values we obtain,

UA = <u>2.596 W/(m.K)</u>

The next step in the analysis is to estimate the temperature change that takes place in the two fluid streams. The calculation performed are briefly presented in the next section, however, the reader is referred to Stoeker (1989) for a detailed explanation.

First calculate the heat capacities, W, for each fluid as follows,

$$W_{exh} = c_{p,exh} \cdot m_{exh}, \quad \text{....(D.2)}$$

and,

$$W_w = c_{p,w} \cdot m_w, \quad \text{....(D.3)}$$

for the exhaust gas and water stream respectively.

To find the water gas exit temperature proceed as follows,

Define, a variable D as follows:

$$D = UA.\left(\frac{1}{W_w} - \frac{1}{W_{exh}}\right) \quad \text{....(D.4)}$$

Substituting the appropriate values for UA, W_{exh} and W_w we have, $D = 0.0254$. Then the exhaust gas exit temperature is given by,

$$T_{w,out} = T_{w,in} - \left[\left(T_{w,in} - T_{exh,in}\right)\left[\frac{1 - e^D}{\left(W_w / W_{exh}\right) - e^D}\right]\right] \quad \text{....(D.5)}$$

The water exit temperature is calculated for increasing heat exchanger length and is shown in graphical form below.

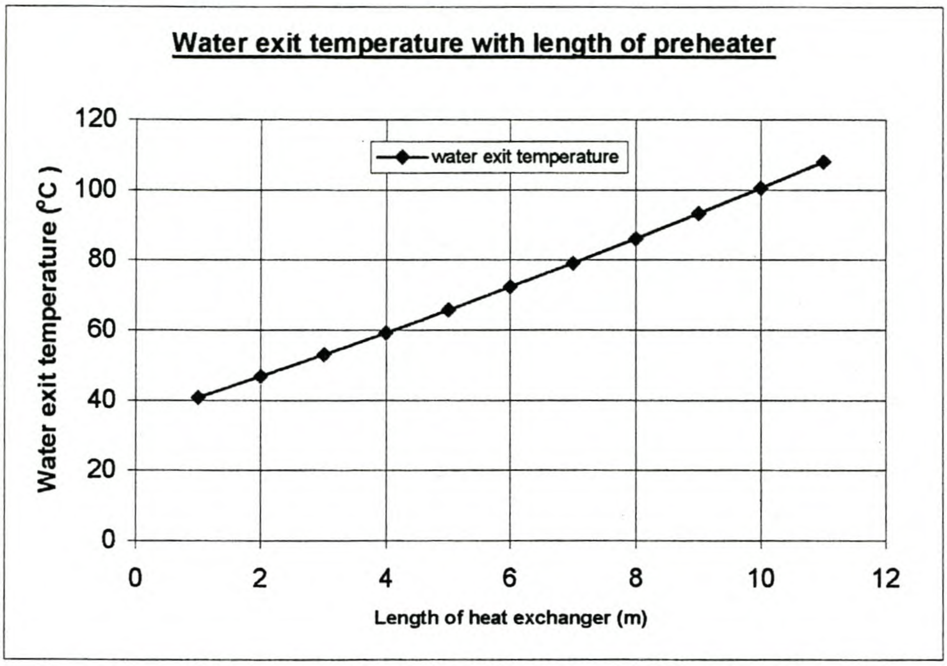


Figure D. 1: Water exit temperature with varying preheater length

It can be seen that to obtain the required exit temperature of water (170.43°C), more than 10m of such a simple counterflow heat exchanger will be required. Such a long heat exchanger is clearly not feasible from a cost effective, space and heat losses point of view.

The boiler and superheater proposed were not investigated further. Instead a shell and tube arrangement for the preheater was investigated and this is presented in the next section.

D.3 The Shell and Tube Preheater

The geometry of the heat exchanger investigated as preheater was a shell and tube arrangement one and the geometry is as shown below in table D.5 below.

Shell inner diameter, $d_{sh,in}$	0.106m
Pipe inner diameter, $d_{p,in}$	0.01m
Pipe wall thickness, t_{pipe}	2mm
Number of pipes , n_{pipe}	8
Tube geometry	
Internal diameter of pipes, id_{pipe}	10mm
Wall thickness, t_{pipe}	2mm
Shell geometry	
Internal diameter, id_{shell}	106mm

Table D. 4: Shell and tube preheater specifications

The fluid inside the pipe is chosen to be water and that on the shell side is exhaust gases.

The properties of water and exhaust gases are the same as calculated in the previous section.

The geometry considered has the following specifications:

Length of heat exchanger investigated, $L_H = 3$m

Consider the water side (tube side)

Mass flow rate through each pipe, $m_{pipe,w} = m_w/N_{pipes}$ ….(D.6)

$= 0.015 / 8$

$= 1.875 \times 10^{-3}$ kg/s.

Reynolds number inside tube is calculated using equation B.1, and is found to be $Re_w = 880.23$. The flow is therefore laminar and the Nusselt number is 3.657 (Kays and Crawford, 1993).

The heat transfer coefficient inside the pipes are therefore obtained using equation B.5 is, h_w=249.478 W/(m^{2}K).

Next consider the exhaust gas side.

The hydraulic diameter, d_{hyd} = 0.044m (using equation B.3)

Therefore, using equation B.2 we find he Reynolds number on the shell side to be Re_{exh}=2.042 x 10^5. The flow is turbulent and using equations B.6 and B.7 respectively we have,

f_D=0.016 and Nu_{exh} = 291.77.

The heat transfer coefficient on the shell side is then calculated using equation B.5 and,

h_{exh}= 227.556 W/(m^{2}K).

The overall heat transfer coefficient per 3 metres length of heat exchanger is obtained by

$$UA = \left[\left(\frac{1}{\pi . L_H}\right)\left(\frac{1}{h_{exh} . d_{t,inner}} + \frac{\ln\left(d_{t,inner} + 2t_{pipe} \big/ d_{t,inner}\right)}{2.k_{steel}} + \frac{1}{h_w . \left(d_{t,inner_1} + 2t_{pipe}\right)}\right)\right]^{-1} . N_{pipes}$$

....(D.7)

= 42.884W/(m.K).

Then the same procedure outlined by Stoeker (1983) and that is used above for the simple counterflow method is adopted. The intermediate value obtained is

D (as defined by equation D.4) = 0.419

For a heat exchanger with an average efficiency of 60% (based on commercially available heat exchangers) the value of D as found above should be multiplied by the efficiency.

Hence, $D = 0.252$, and the temperature profile for water along the heat exchanger on the water side is found by applying equation D5 and it is shown graphically in figure D2 below.

It can be seen that the length of heat exchanger required to achieve a required temperature rise of from 35°C to 170.43°C would again require approximately 7.2m. This again is not a feasible from a cost and increased heat losses point of view.

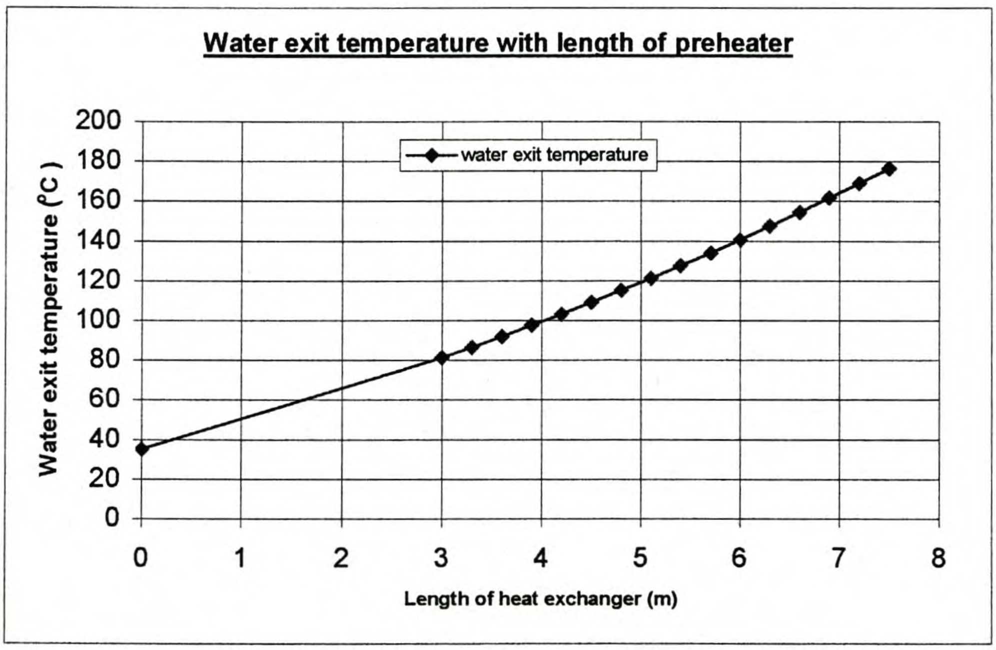


Figure D.2: Water temperature profile along a shell and tube waste heat exchanger

After analysis the results obtained it was found that a shell and tube heat exchanger is not sufficient because of the low heat transfer coefficients on the air side. Therefore a finned heat exchanger was next investigated.

D.4 Copper Finned Copper Pipe Waste Heat Exchanger

This was the final design investigated and purchased. The specification for the heat exchanger and the calculation for the preheater section are presented in section 6.1. The procedure for the calculation of the heat transfer coefficient on the water side for the boiler are also explained in section 6.1 and the results obtained is presented figure D.3 below in graphical form.

The main varying parameters on which the heat transfer coefficient depends inside the pipe is presented in below.

It was found from the preliminary calculations that 3 rows of such waste heat exchanger would suffice for preheating the entering liquid water to saturated liquid.

The boiler section was then investigated by dividing the pipe into 19 control volumes such that the quality of the mixture changes by 0.05 in each control volume.

The procedure adopted is presented in chapter 6 and the equations used are 6.24 to 6.37. For a more detailed explanation the reader is referred to Lock (1996).

The main parameter that also remain constant because of the assumption of a constant temperature of the wall is the nucleate boiling component, $q_{NB} = K.(T_{wall} - T_{sat})^3$ and has a value of 3.895×10^8 W/(m^2.K).

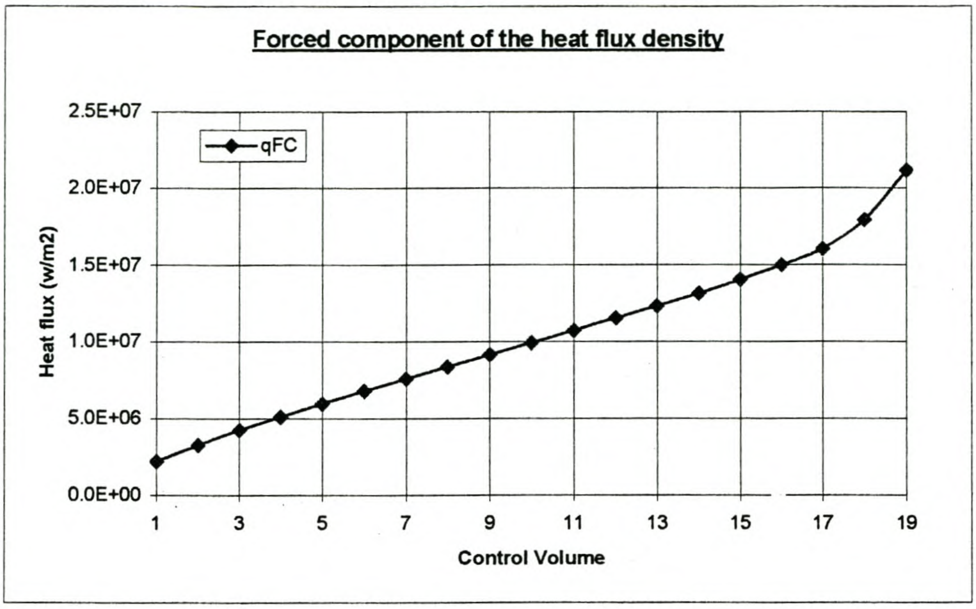


Figure D. 3: Forced component of heat flux density in each control volume

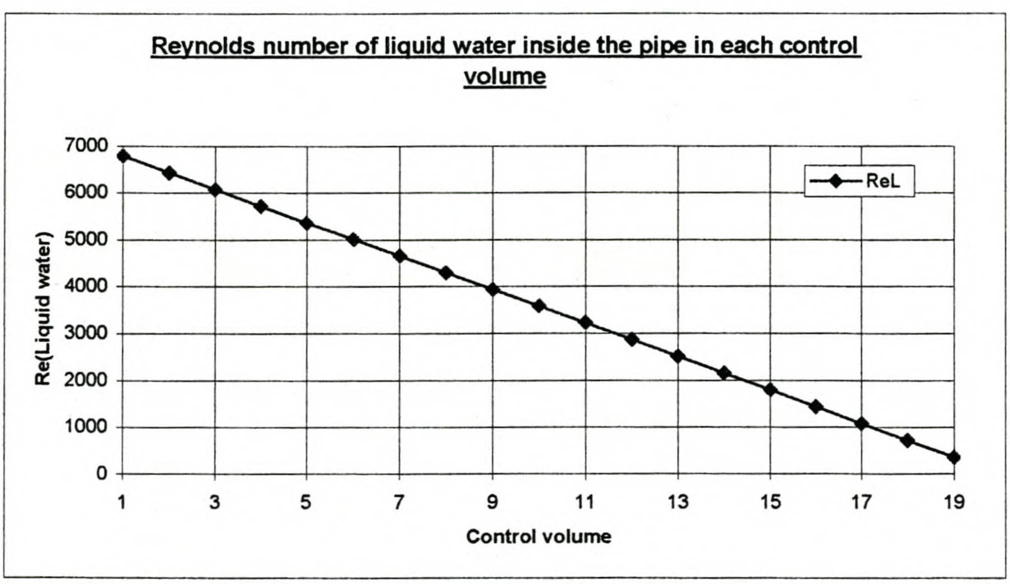


Figure D. 4: Reynolds number of liquid water for each control volume

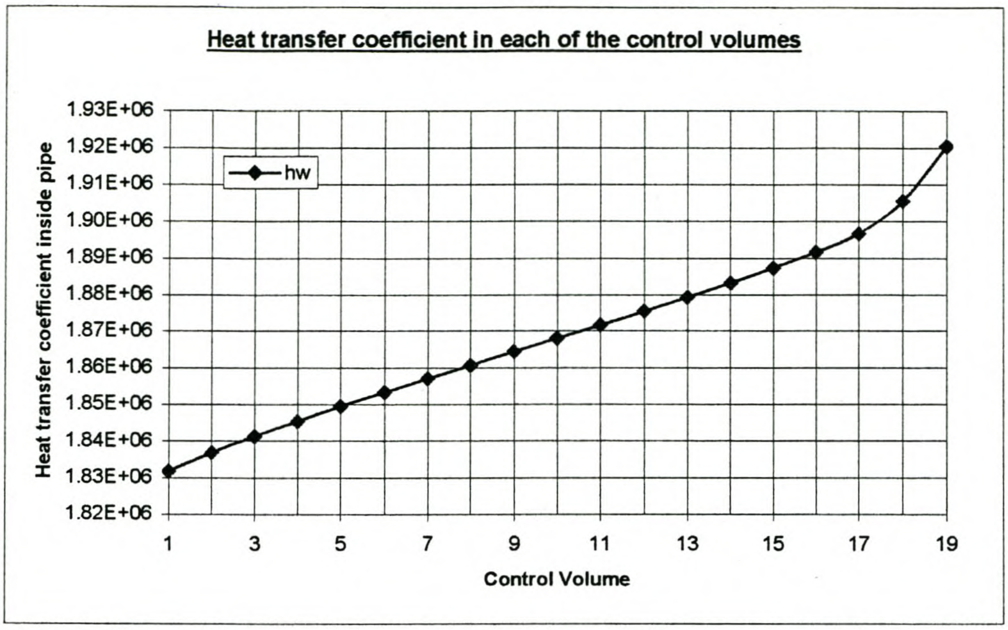


Figure D. 5: Total heat transfer coefficient in each control volume

Finally the average heat transfer in the boiler section of the waste heat exchanger is calculated by averaging the heat transfer coefficients in the 19 control volumes. The average heat transfer coefficient was found to be 1869.467 kW/(m^2.K).

APPENDIX E

EXHAUST GAS PROPERTIES

E.1 Exhaust Gas Density

In this appendix it is assumed that complete combustion of the diesel fuel takes place i.e. a stoichiometric mixture is combusted in the engine. The mass fraction of a constituent is calculated by dividing the mass of the constituent by the total mass of one mole of exhaust gases. The molar mass of exhaust gases is 30.717 g/mol as shown in section 4.2.2. The mass of a constituent is calculated by multiplying the relative molecular mass of the constituent by the mole fraction obtained by balancing the chemical combustion equation (equation 4.2).

Gas	*Mass fraction*	*Density*	*Volume Fraction*
CO_2	0.196	1.2	0.163
H_2O	0.083	1.8	0.046
N_2	0.72	0.5863	1.228
		Specific volume	*1.437*

Table E. 1: Specific volume of exhaust gas

Therefore, the density of the exhaust gas = 1 / Specific volume ....(E.1)

= 1 / 1.437

= <u>0.696 kg/ m^3</u>.

E.2 Specific heat capacity of exhaust gas

Again the specific heat capacity of the exhaust gases is calculated by adding the product of the mass fraction and the specific heat capacity of each constituent.

Gas	*Mass fraction of gas*	*c_p (kJ/kgK)*	*Mass fraction x c_p*
CO_2	0.196	0.846	0.166
H_2O	0.083	1.872	0.156
N_2	0.720	1.039	0.748
		Total c_p (kJ/kgK)	**1.07**

Table E. 2: Specific heat capacity calculation

Therefore the specific heat capacity of the exhaust gases is 1.07kJ/(kgK) of gas.

APPENDIX F

THERMAL EXPANSION OF EXHAUST GAS PIPE NETWORK

The first stage of the project involved setting up the piping layout to divert the exhaust gases from the exhaust exit (into the exhaust channel) to the *Energy Systems Laboratory* situated above the Engine test cells. Preliminary calculations, which are also presented below, shows that the expansivity of the relatively long steel pipes are quite considerable.

Coefficient of thermal expansion of mild steel, α = 12.10^{-6} m/°C (Gieck, 1989)

Average temperature of exhaust gases in side the pipes, T_{av} = 500°C

Average room temperature, T_{room} = 20°C

Temperature change, ΔT $= T_{av} - T_{room}$ ....(F.1)

$= 480°C$.

Per unit length of pipe,

Expansion of pipe, Δl = $(\alpha.\Delta T) \times$ original length ....(F.2)

$= (12 \times 10^{-6} \times 480) . 1$

= 0.00576m per metre length of pipe.

For the 3000 mm pipe section

Average length of pipe, l = 7.8m

The longest section of the pipe network (7.8m) will expand by

= 0.00576. x 7.8 m ....(F.3)

= 0.045m

= 44mm

The expansions calculated in other sections of the piping system is summarised in table F.1 below.

Section	*Original length(mm)*	*Expansion(mm)*
1.	7800	44
2.	3000	17.28
3.	4470	25.75

Table F. 1: Expansion in the three major sections of the exhaust gas piping

As can be seen there is considerable expansion that should be expected in the piping system.

To account for these considerable thermal expansions of the pipes inside the ESL, axial expansion compensators (also referred to as bellows) were used in the two long sections of the pipe network.

APPENDIX G

PRESSURE DROP APPROXIMATION IN THE PIPE NETWORK

In this appendix the average expected pressure drops in the pipe network are calculated.

G.1 Tapping point at existing exhaust gas pipe exit into the exhaust gas chamber

At the point where exhaust gas is tapped from the existing engine exhaust there is a contraction in radii.

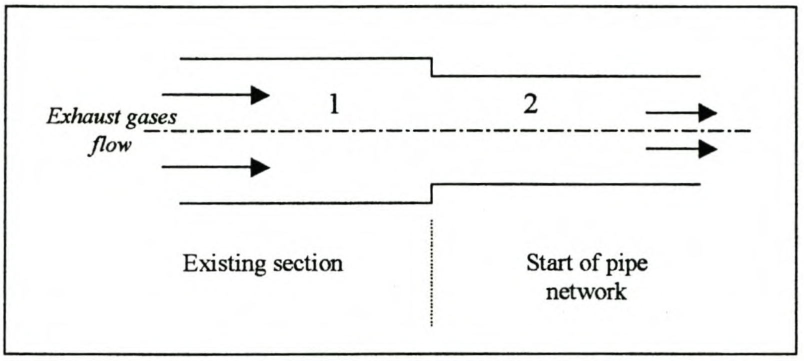


Figure G. 1: Pipe contraction at the initial tapping point inside exhaust chamber

The maximum size of the pipe was limited by the 200mm holes drilled into the floor in the ESL.

Pressure drop due to this contraction is calculated using equation presented by Kröger (1998) and restated below is,

$$p_1 - p_2 = \frac{\rho v_2^2}{2}\left[\left(1-\sigma_{21}^2\right)+K_c\right] \qquad \text{...... (G.1)}$$

where,

the area contraction ration, $\sigma_{21} = A_2/A_1$,

and, the contraction coefficient, K_c can be obtained from Kröger (1998)

The relevant value used and the calculated results are as follows,

$\sigma_{21} = 0.78$

$\rho = 0.696$ kg/m^3 (appendix E)

$K_c = 5.5 \times 10^{-2}$

$v_2 = 26.57 m/s$

Substituting the above values into eq. G.1, we get

$$\Delta p = p_2 - p_1$$
$$= 106 Pa$$

G.2 In the longer straight pipe sections

The Reynolds number of the exhaust gases is, $Re_{exh} = 83652$ (see appendix H).
Using equation B.7 we find, $f_D = 1.867 \times 10^{-2}$.

The pressure drop per unit length of pipe is given by,

$$\Delta p = f_D \cdot \left(\frac{\rho . v^2}{2}\right) \cdot \left(\frac{L}{d}\right) \quad \text{...... (G.2)}$$

$$= 65.2 Pa \text{ per metre length of pipe}$$

Therefore, for a total length of pipe of 16m the pressure drop will be,

$$\Delta p = (65.2 \times 16)\ Pa \quad \text{.....(G.3)}$$
$$= 1043 Pa$$

For each elbow, equivalent pipe length = 3 (Energy Research Institute, 2001)

Number of elbows in E.S.L. = 2

Therefore additional pressure drop due to elbows inside the ESL,

$$\Delta p = (65.2 \times 3 \times 2) Pa \quad \text{.....(G.4)}$$
$$= 391 Pa$$

Adding all the pressure drops we obtain a total approximate pressure drop of,

$$\Delta p_{total} = (1043 + 391 + 106)\ Pa \quad \text{.....(G.5)}$$
$$= \underline{1540 Pa}$$

This pressure drop is experienced inside the ESL and as it can be seen is very low. Therefore, the back pressure exerted on the engines is relatively small. The low pressure inside the pipes due to the exhaust gas extraction fan located at the exit of the pipe network. This negative pressure inside the pipes will further reduce the back pressure exerted on the engines being tested or the combustion unit when it is being used as an alternative source of exhaust gases.

APPENDIX H

PIPE INSULATION THICKNESS

In this appendix the calculation of the thickness of insulation required for exhaust gas supply is investigated.

Pipe Specifications

Material	Mild Steel
Conductivity, k	43W/ (m^2.K)
Outside diameter	101.6mm (4 inches)
Inside diameter	95.6mm
Wall thickness	4mm

The heat transferred across the wall is given by the heat equation :

$$Q_{transferred} = UA\ \Delta T \qquad \text{....(H.1)}$$

Where, the temperature difference, ΔT of the exhaust gases is given by

$$\Delta T = T_{in} - T_{out} \qquad \text{....(H.2)}$$

and based on the inner surface area one can also write,

$$UA = U_i\ A_i = \frac{1}{\left(\frac{1}{h_i A_i}\right) + \left(\frac{\ln\left(\frac{r_o}{r_i}\right)}{2\pi k_{steel} L}\right) + \left(\frac{\ln\left(\frac{(r_o + t)}{r_o}\right)}{2\pi k_{insulation} L}\right) + \left(\frac{1}{h_o A_o}\right)} \qquad \text{....(H.3)}$$

Where,

r_i and r_o are the inner and outer radii of the supply pipe respectively,

t is the thickness of the supply pipe,

and $k_{insulation}$ and k_{steel} are the conductivities of the insulation and the supply pipe respectively.

Determination of the heat transfer coefficient, h_i, inside the exhaust pipe

Reynolds number for the exhaust gas, $Re_{exh} = (\rho\ v\ d_i\ /\ \mu)$ ....(H.4)

Now mass flow rate, $m_{exh} = \rho\ A\ v$ ....(H.5)

$$\rho\ v = m_{exh} / A$$

Therefore, the Reynolds number can also be written as

$$Re = m_{exh}\, d_i / A_i\, \mu$$

$$= 4\ m_{exh} / d_i\, \pi\, \mu. \qquad \text{....(H.6)}$$

Substituting the values obtained from appendix A, we get,

$$Re = (4 \times 0.1632) / (0.106 \times \pi \times 2.671 \times 10^{-5})$$

$$= 73392$$

(At a temperature of 500 ^{o}C the viscosity of air = 2.671 x 10^{-5} kg / ms.)

Also $\quad Pr = 0.7078$ (Assumed to be approximately equal to that of air)

$\quad d/L = 0.0106$ (per meter length)

The flow is therefore turbulent, and therefore we can use equations. B.6 and B.7 to calculate the heat transfer coefficient on for he exhaust gases

These restated are,

$$Nu = \frac{\left(f_D/8\right)(Re-1000)Pr\left[1+\left(d/l\right)^{0.67}\right]}{1+12.7\left(f_D/8\right)^{0.5}\left(Pr^{0.67}-1\right)}, \qquad \text{....(B.6)}$$

for the following conditions,

$$2300 < Re < 10^6,$$

$$0.5 < Pr < 10^4,$$

and $\quad 0 < d/L < 1$

where, $\quad f_D = (1.82 \log Re - 1.64\ )^{-2}$ ....(B.7)

Substituting the values, we have,

$$f_D = 1.92 \times 10^{-2}$$

And

$$Nu = \frac{\left(1.92.10^{-2}/8\right)(73392-1000)Pr\left[1+(0.106)^{0.67}\right]}{1+12.7\left(1.92.10^{-2}/8\right)^{0.5}\left(0.7078^{0.67}-1\right)}$$

$$= 151.04$$

Hence the heat transfer coefficient is finally obtained using equation. B1.

$$h_i = Nu \,.\, k / d \qquad \text{....(B.1)}$$

Using equation B.10, we find the conductivity of air, k_{air} at average temperature of 500°C = 0.04038W/(m.K).

Therefore,

$$h_i = 151.04 \times 0.04038 / 0.106$$

$$= 57.53 W / m^2 K.$$

Determination of the heat transfer coefficient, h_o, the outside of the pipe (for the case where there is no insulation)

Using equations presented by Holman (1992) for free convection from a horizontal pipe, we have,

If flow is turbulent,

$$h_o = 1.24\,(\Delta T)^{1/3} \qquad \text{....(H.7)}$$

and if flow is laminar,

$$h_o = 1.32(\Delta T / d)^{1/4} \qquad \text{....(H.8)}$$

Substituting the values into the above equations we have,

For, the turbulent case: $h_o = 1.24\,(500\text{-}25)^{1/3}$

$$= 9.68 W / m^2 K.$$

and, for the laminar case: $h_o = 1.32((500\text{-}25)/0.106)^{1/4}$
$= 10.80 W / m^2 K.$

The temperature profile obtained for the non-insulated case is obtained using the above calculated properties, and compared with that of using insulation of varying thickness.

The temperature profile along the pipe length is shown in figure H.1 below.

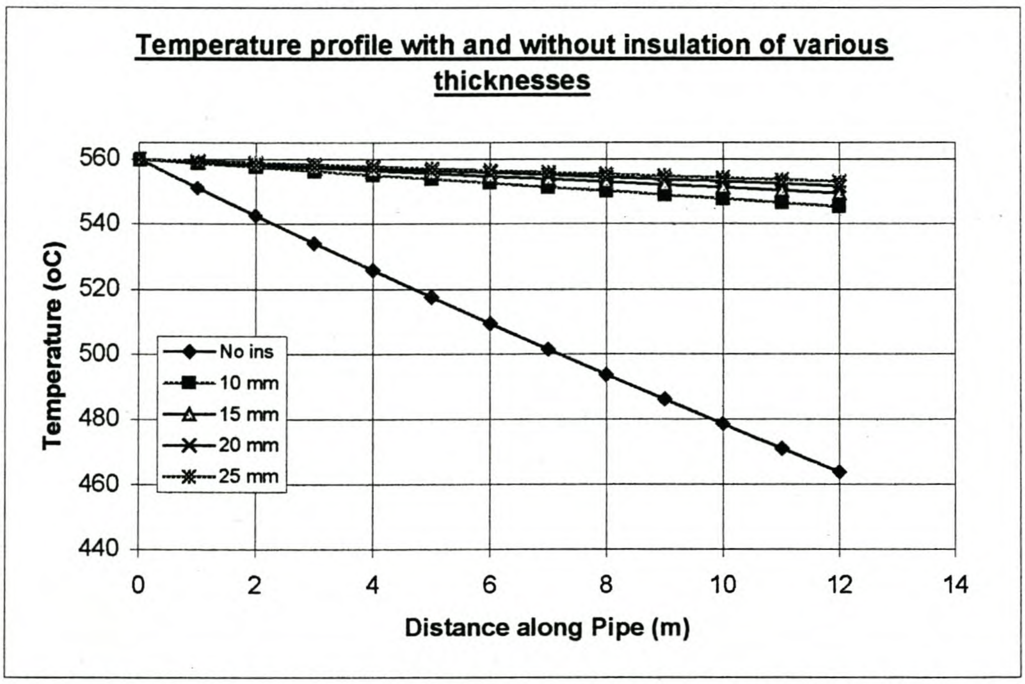


Figure H. 1: Temperature profile of exhaust gas temperature along pipe network

It can be seen that the temperature drop along the exhaust gas supply pipe is reduced satisfactorily by using 20mm of insulation. A thicker insulation layer does not improve the exhaust gas supply temperature to the waste heat exchanger much.

APPENDIX I

QUICK REFERENCE GUIDE FOR CONTINUOUS COMBUSTION UNIT OPERATION (Schwack,2000)

1. Ensure that all fuel valves are closed.
2. Zero the air flow rate using the calibration adjustment on the side of the unit
3. Drain any condensate in the combustion chamber by unscrewing the drain knob below the combustion chamber
4. Open water in the following order
    - Open main (RED) water valve at the waste heat exchanger completely
    - Open main (BLACK) water supply valve
    - Open secondary (RED) water supply valve completely
5. Switch on the mains (grid) at lead
6. Switch the mains (RED) switch on the Combustion unit
7. Push fan start button (GREEN)
8. Open air inlet to maximum (Position 8)
9. Open GAS bottle completely
10. Set AIR mass flow rate to **50 kg/h**
11. **Start combustion as follows**
    - Make sure reset switch if in the off (UP) position
    - Press the instrument reset switch button (PURPLE)
    - Set reset switch to on position (DOWN)
    - Wait for the solenoid to open (A 'TAK' sound is heard)
    - PRESS AND HOLD the black ignition switch
    - While still holding the ignition open the fuel knob and set fuel flow to 4-5 kg/h
    - After flame has been established hold the ignition switch for a further 5 seconds
12. **If ignition not successful**
    - Close the gas fuel knob on the combustion unit
    - Turn OFF the fuel supply at the MAIN supply (Gas bottle)
    - Open air flow valve to maximum (position 8)
    - Wait 2 minutes to purge any unburned fuel from combustion chamber
    - Repeat procedures 9 to 11

13. Shut down procedure

- Turn OFF the fuel supply at source (gas bottle)
- Switch off the reset switch
- Progressively move the air flow control to position 8 (Maximum)
- Wait for 2 minutes for all gases and any unburned fuel to purge
- Press stop (RED) button to switch off fan
- Switch mains switch off
- Disconnect mains
- Close the water valve in the following order BLACK and RED VALVES at main supply, then the BLUE on the panel.

Switching over to liquid fuel

IMPORTANT: Run the combustion unit on the gaseous fuel for at least **10 minutes** to allow the normal operating conditions to be reached.

1. Slowly OPEN the LIQUID fuel valve (bottom right on panel) and simultaneously slowly CLOSE the GAS fuel using the control valve on the unit (the colour of the flame will change from blue to bright yellow and the exhaust note might also change)

NOTE: The switching over will be achieved by progressively and simultaneously performing the above. The whole procedure should take about 30 seconds.

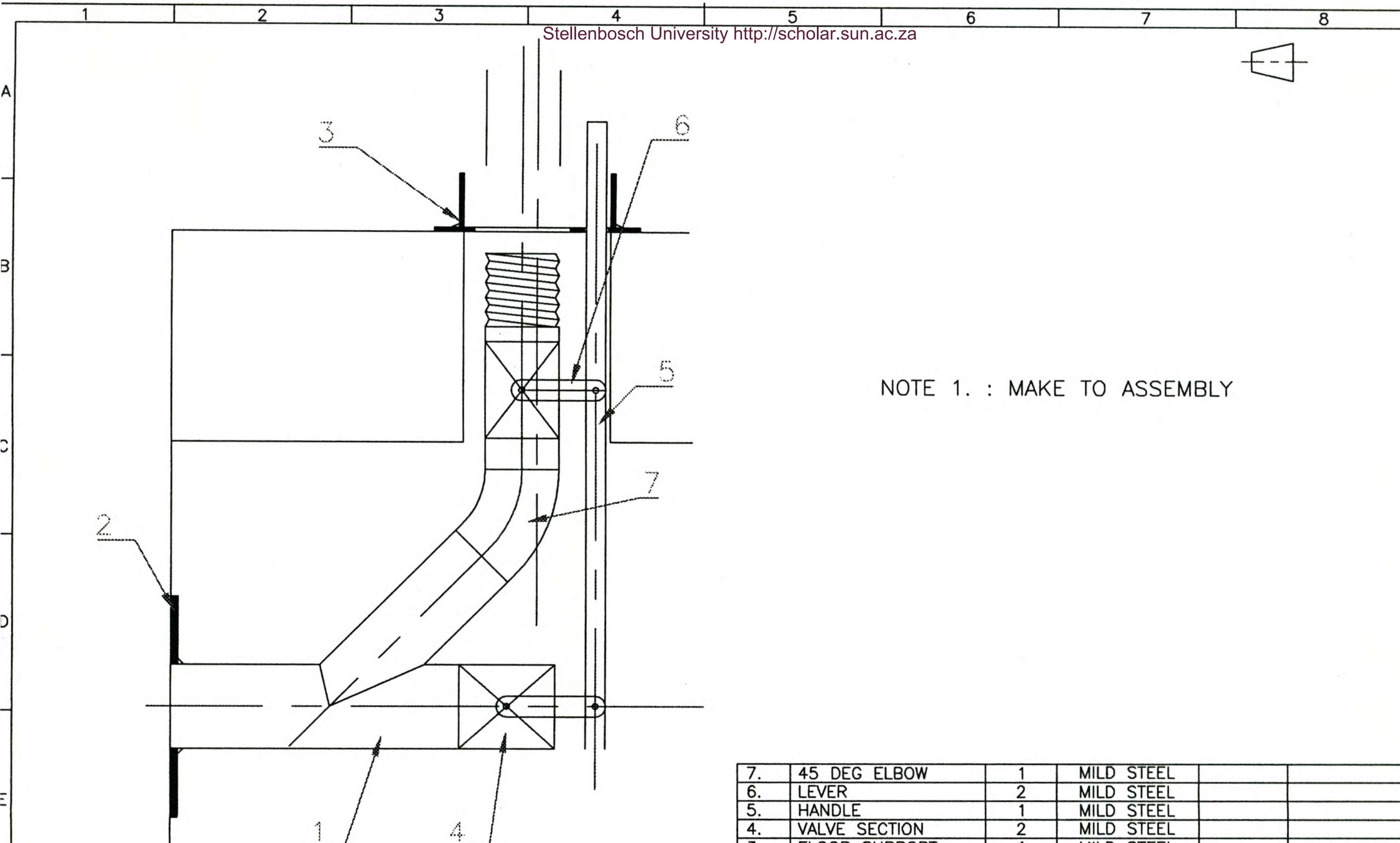
2. If GAS fuel is no longer required then turn OFF at the main supply (i.e. Gas bottle)

3. **Shut down procedure**

- Turn OFF the liquid fuel control valve on panel
- Switch off the reset switch
- Progressively move the air flow control to position 8 (Maximum)
- Wait for 2 minutes for all gases and any unburned fuel to purge
- Press stop (RED) button to switch off fan
- Switch electricity off at unit (RED) and mains.
- Disconnect mains
- Close the water valve in the following order BLACK and RED VALVES at main supply, then the BLUE on the panel.

APPENDIX J

DRAWINGS



NOTE 1. : MAKE TO ASSEMBLY

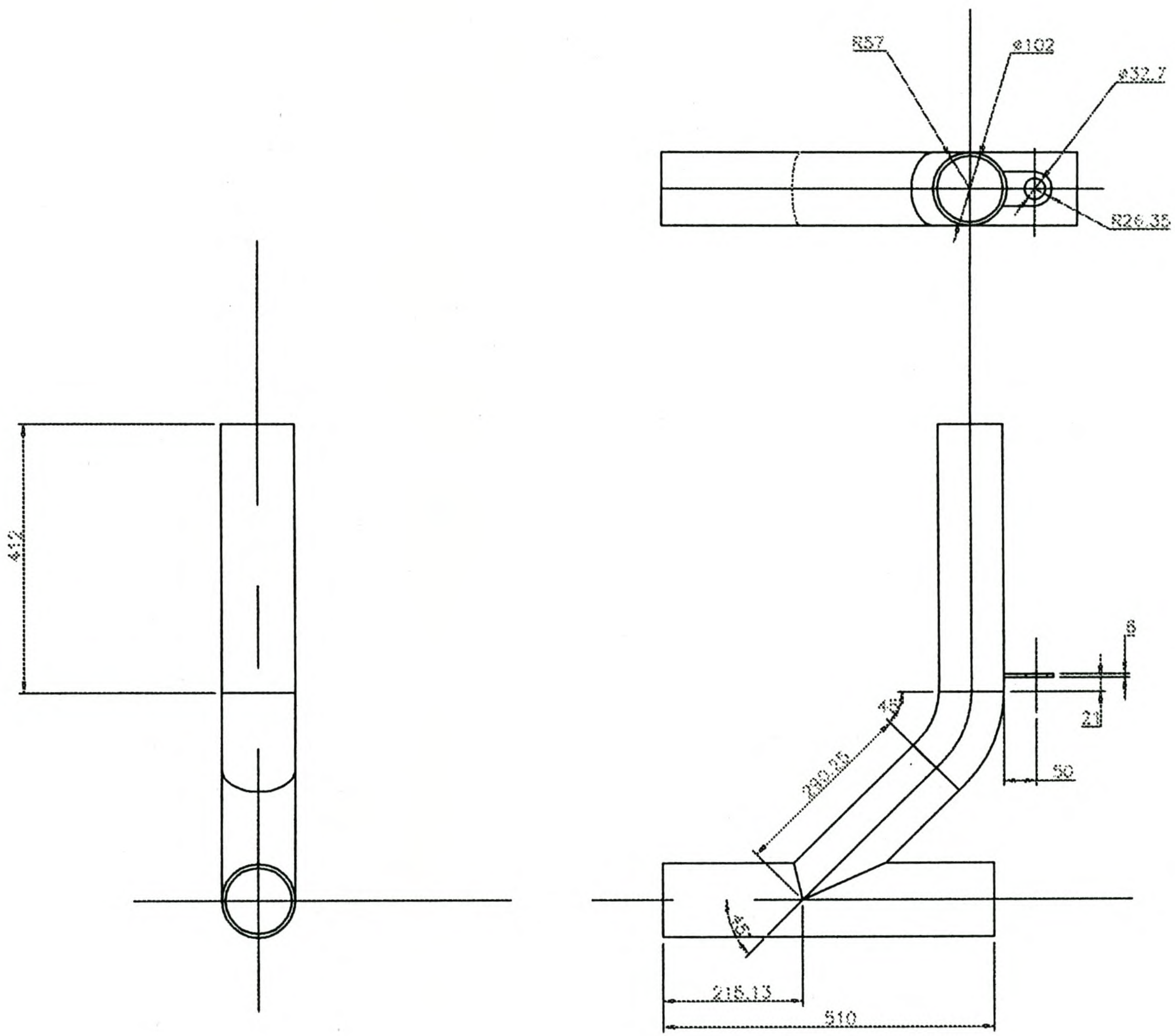
ITEM	DESCRIPTION	AMOUNT	MATERIAL	SPECS	REMARKS
7.	45 DEG ELBOW	1	MILD STEEL		
6.	LEVER	2	MILD STEEL		
5.	HANDLE	1	MILD STEEL		
4.	VALVE SECTION	2	MILD STEEL		
3.	FLOOR SUPPORT	1	MILD STEEL		
2.	WALL FLANGE	1	MILD STEEL	300x300	
1.	MAIN PIPING	1	MILD STEEL		

UNIVERSITY OF STELLENBOSCH

SCALE

MEAS IN mm

TITLE Assembly Drawing for Piping Arrangement

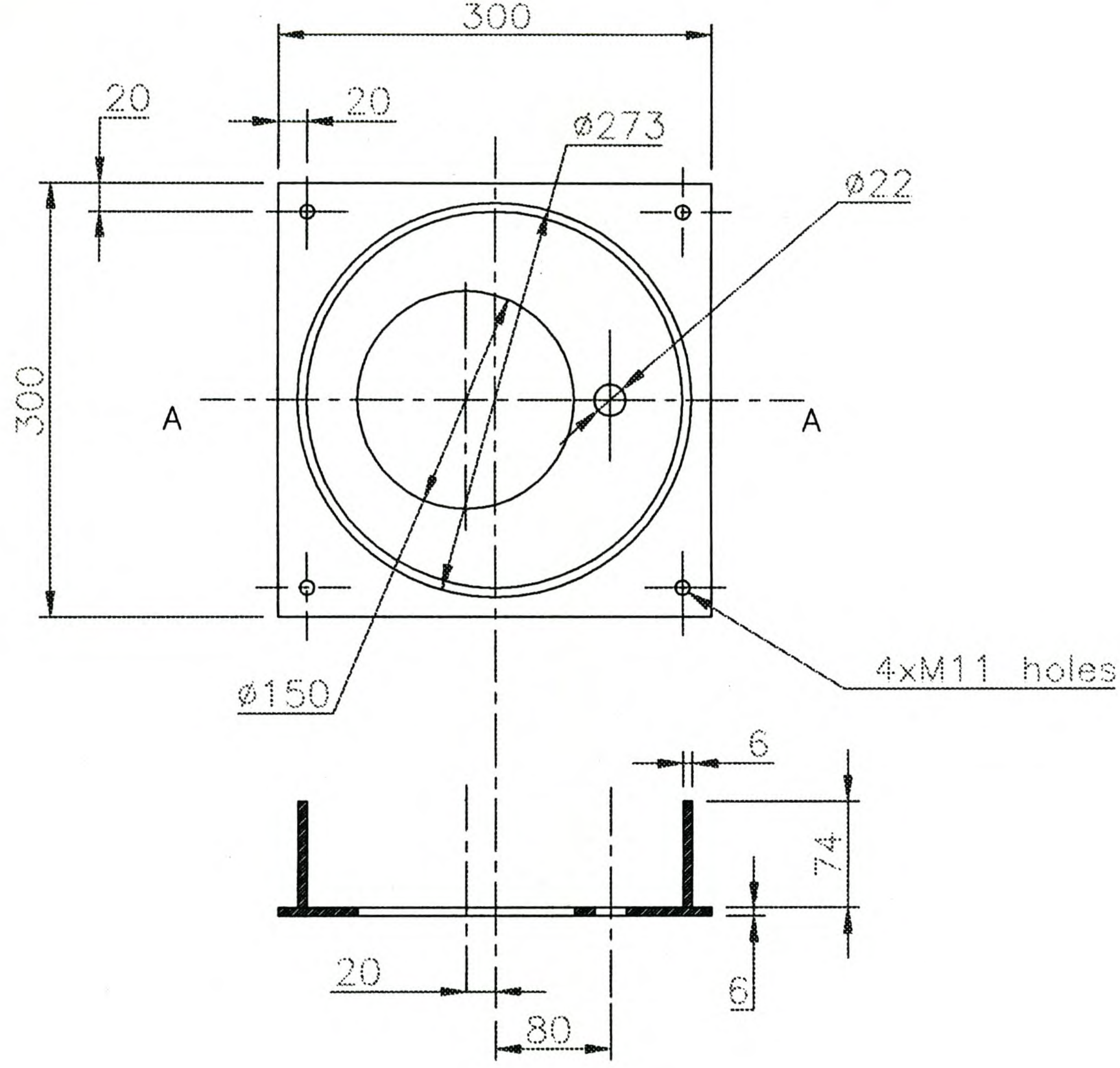


ITEM	DESCRIPTION	AMOUNT	MATERIAL	SPECS	REMARKS

UNIVERSITY OF STELLENBOSCH

SCALE
MEAS IN mm

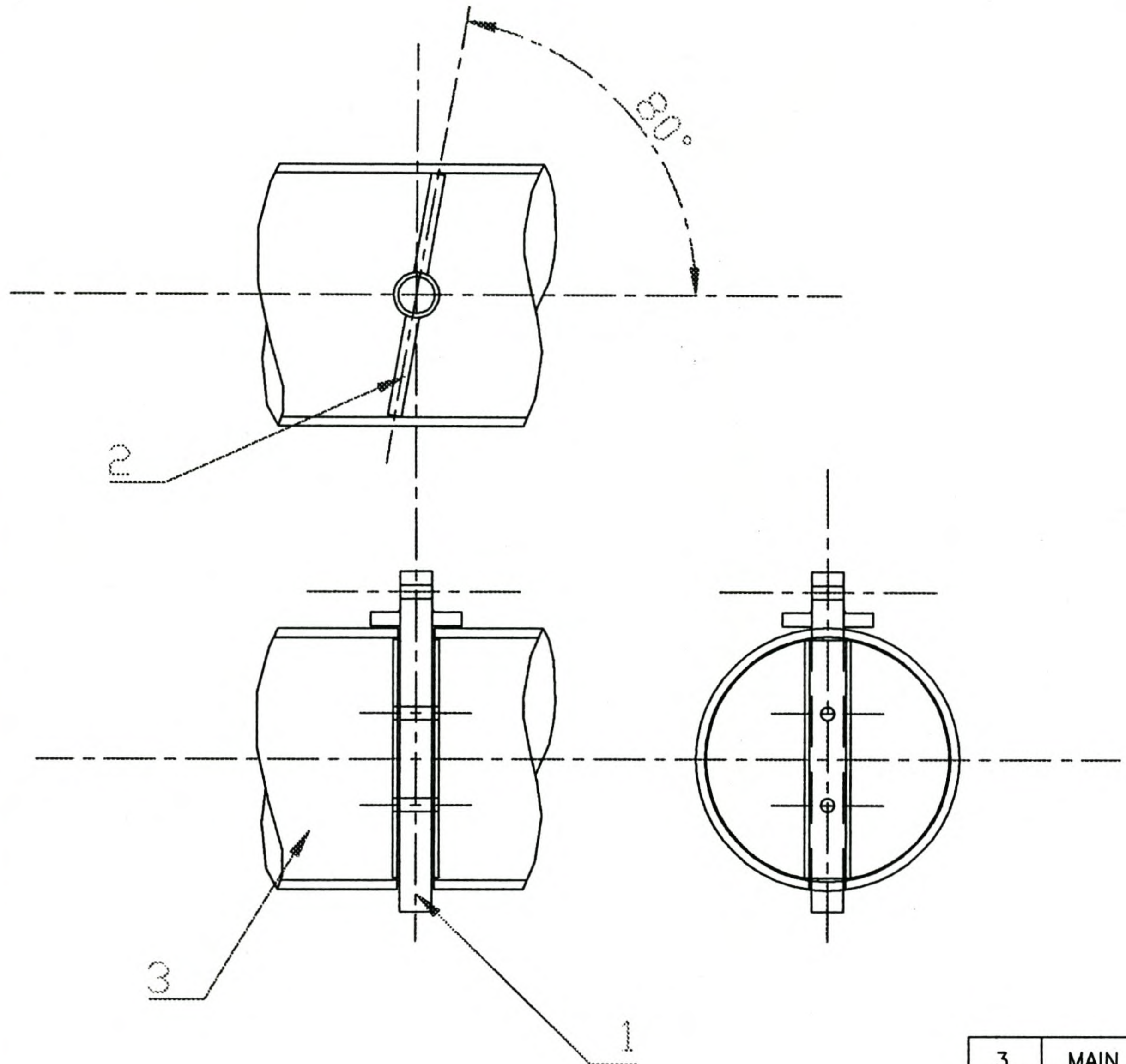
TITLE : MAIN PIPING AND HANDLE GUIDE



SECTION A–A

ITEM	DESCRIPTION	AMOUNT	MATERIAL	SPECS	REMARKS

UNIVERSITY OF STELLENBOSCH		SCALE MEAS IN mm	TITLE : FLOOR SUPPORT
STUDENT No. 9969578	DRAWN BY D. LOTUN	DATE 14.03.00	SHEET No. OF SHEETS | No.



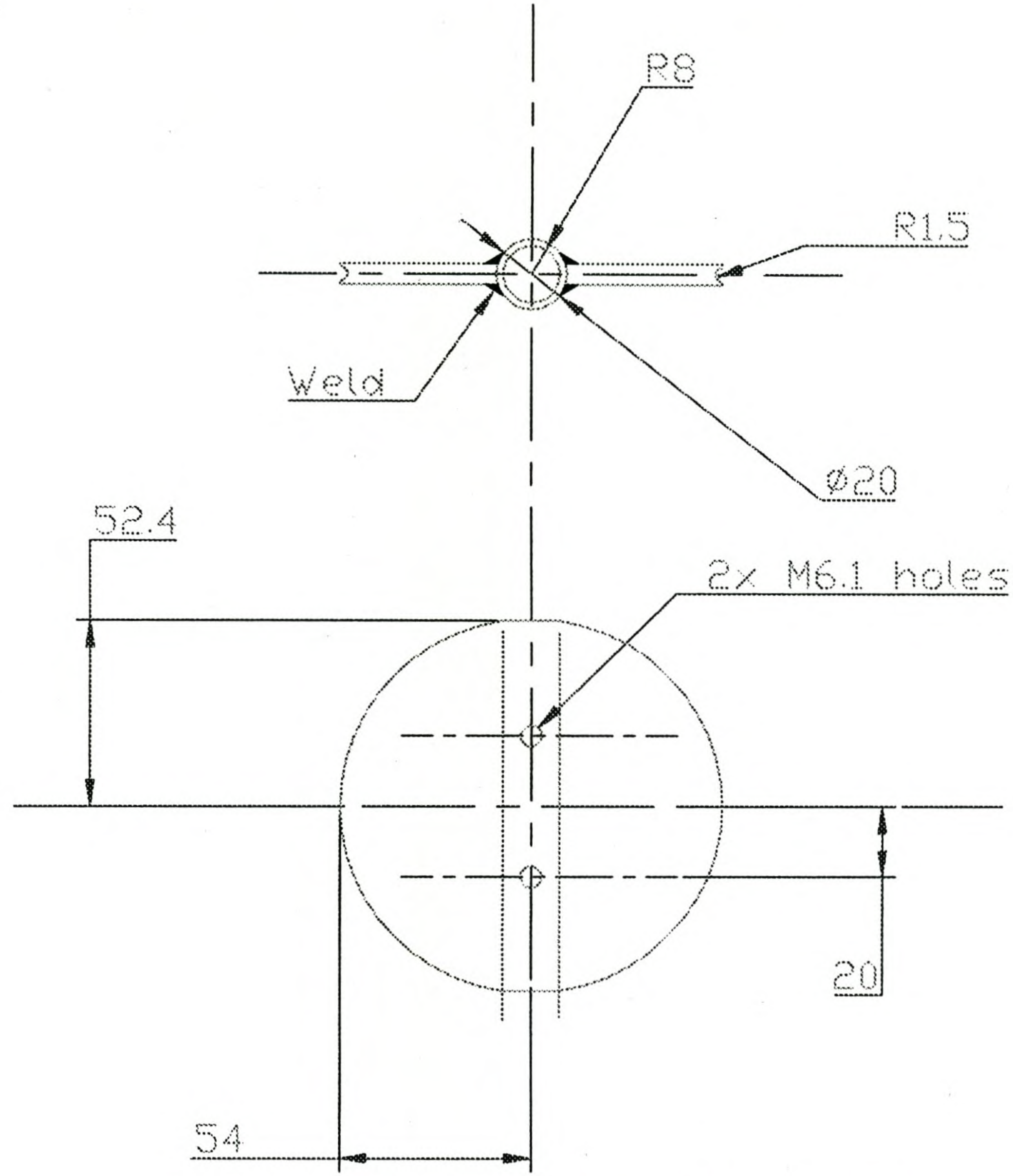
ITEM	DESCRIPTION	AMOUNT	MATERIAL	SPECS	REMARKS
3.	MAIN PIPE	1	MILD STEEL	Nom ø100	
2.	VALVE DISK	1	MILD STEEL		Ellipse
1.	AXIAL PIN	1	MILD STEEL	ø 16	

UNIVERSITY OF STELLENBOSCH

SCALE

MEAS IN mm

TITLE :VALVE ASSEMBLY DRAWING

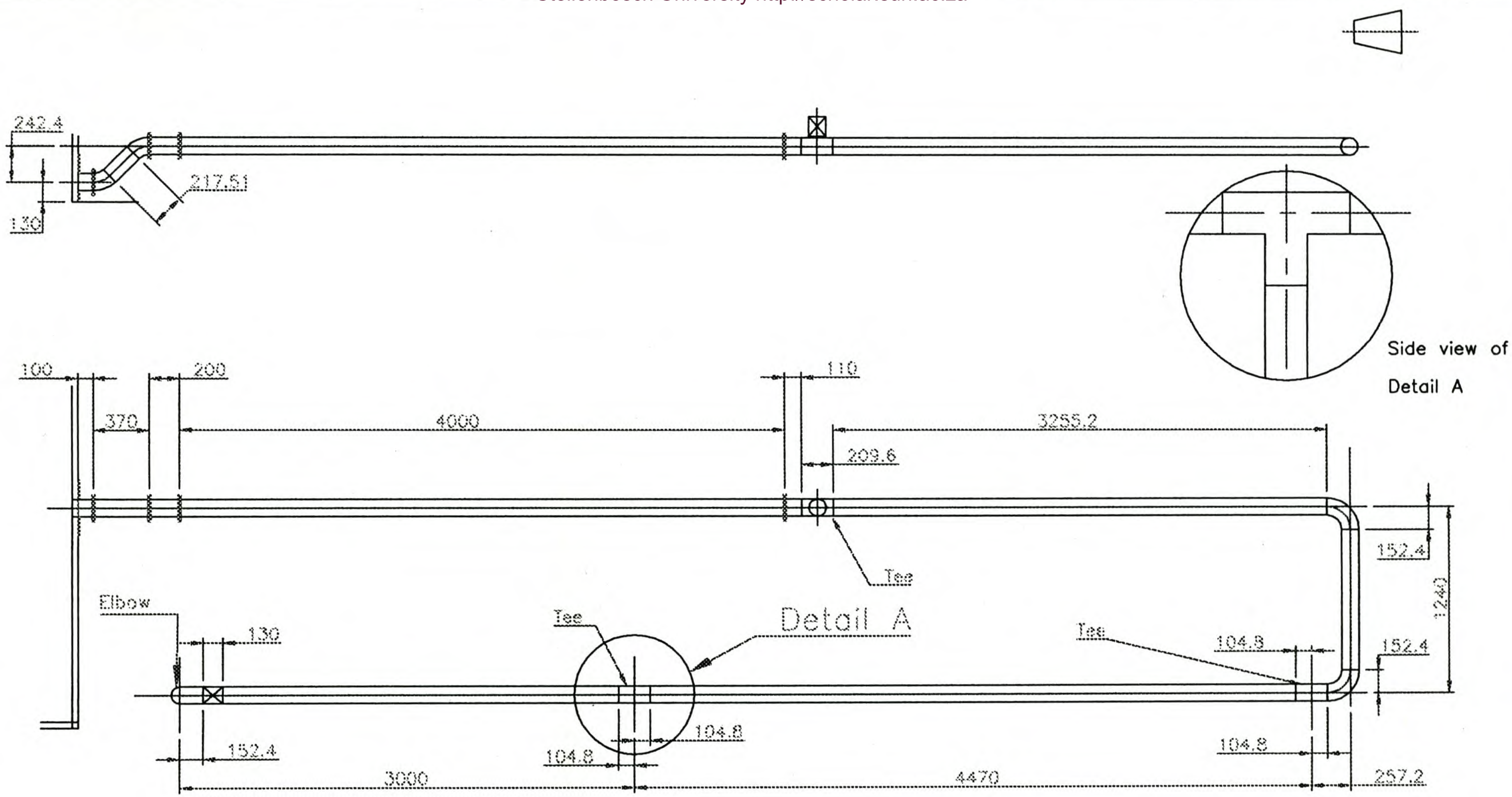


ITEM	DESCRIPTION	AMOUNT	MATERIAL	SPECS	REMARKS

UNIVERSITY OF STELLENBOSCH

SCALE
MEAS IN mm

TITLE **VALVE DISC**



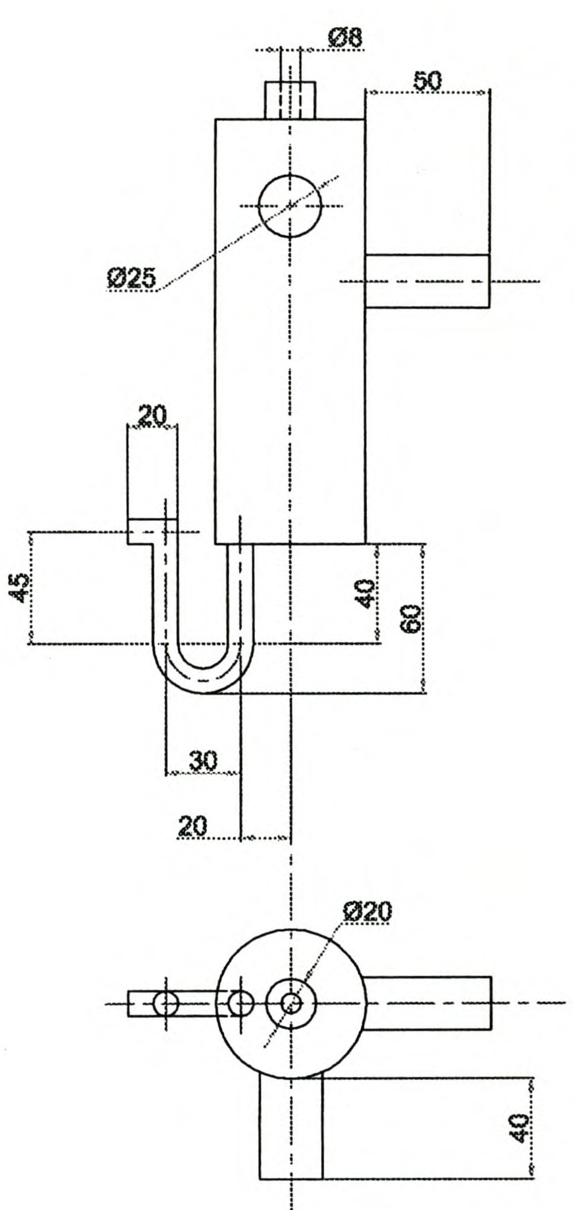
NOTE 1 : One additional valve is connected with EACH of the TWO Tees over the floor openings

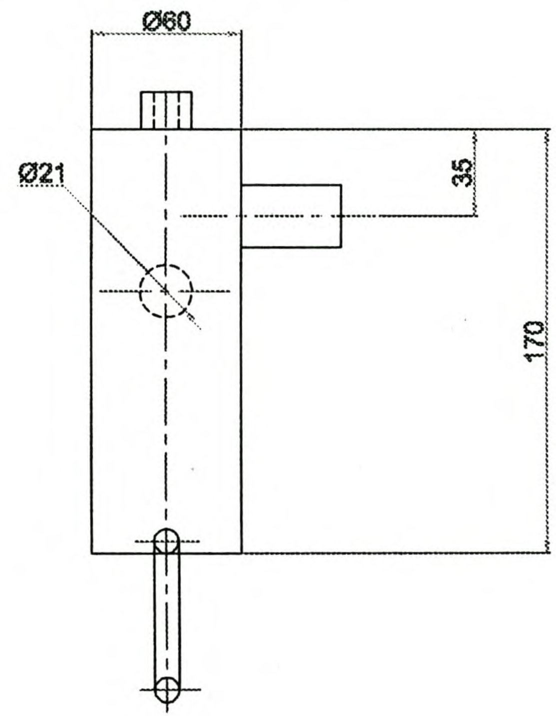
NOTE 2 : All pipes have an outside diameter of 114.3mm

ITEM	DESCRIPTION	AMOUNT	MATERIAL	SPECS	REMARKS

UNIVERSITY OF STELLENBOSCH

SCALE	TITLE : PIPING LAYOUT IN ESL		
MEAS IN mm			
STUDENT No. 9969578	DRAWN BY D. LOTUN		
DATE 14.03.00	SHEET No. OF SHEETS	No.	



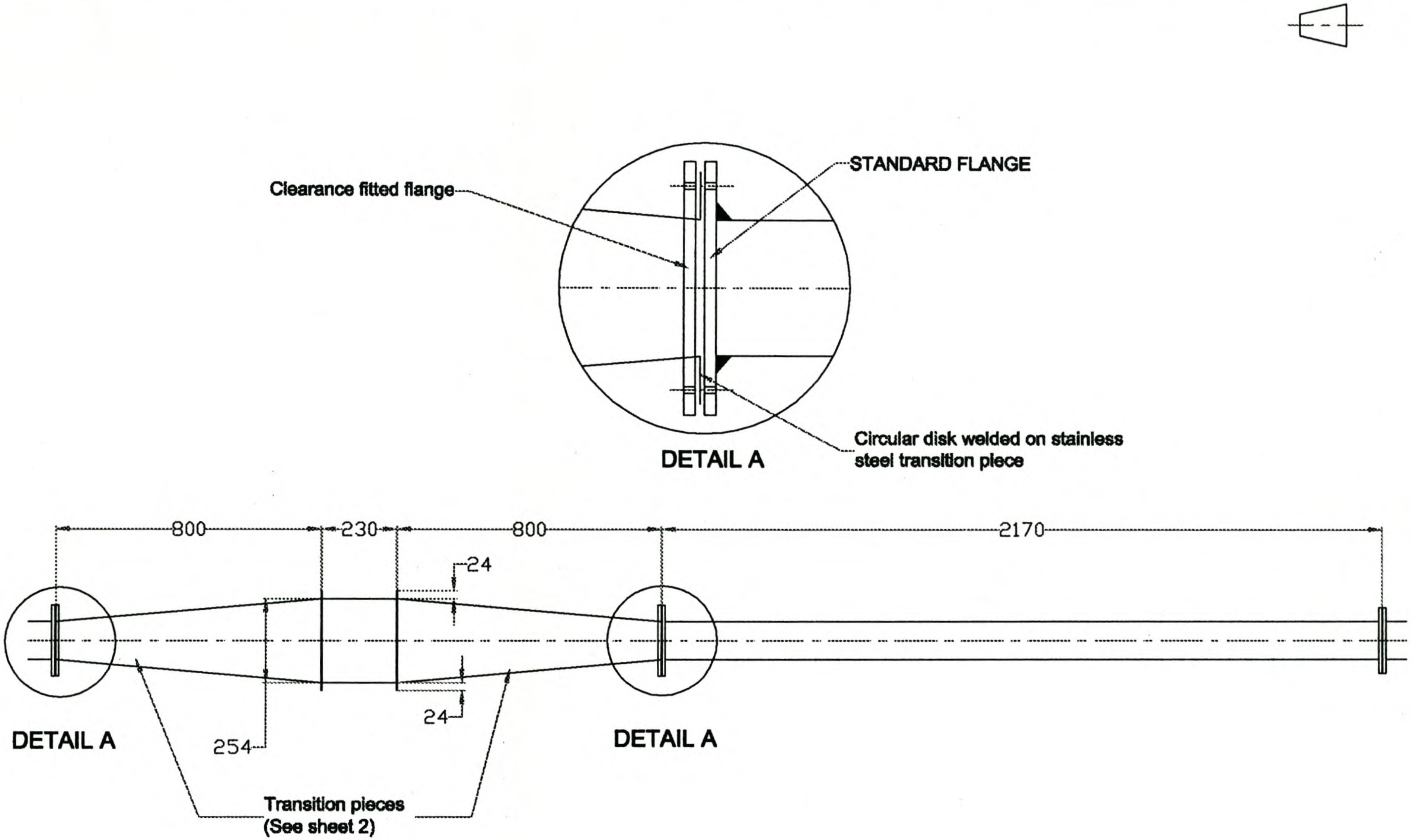


NOTE : THE Ø8mm HOLE ON TOP IS TO BE THREADED 1/8 NPT

ITEM	DESCRIPTION	AMOUNT	MATERIAL	SPECS	REMARKS

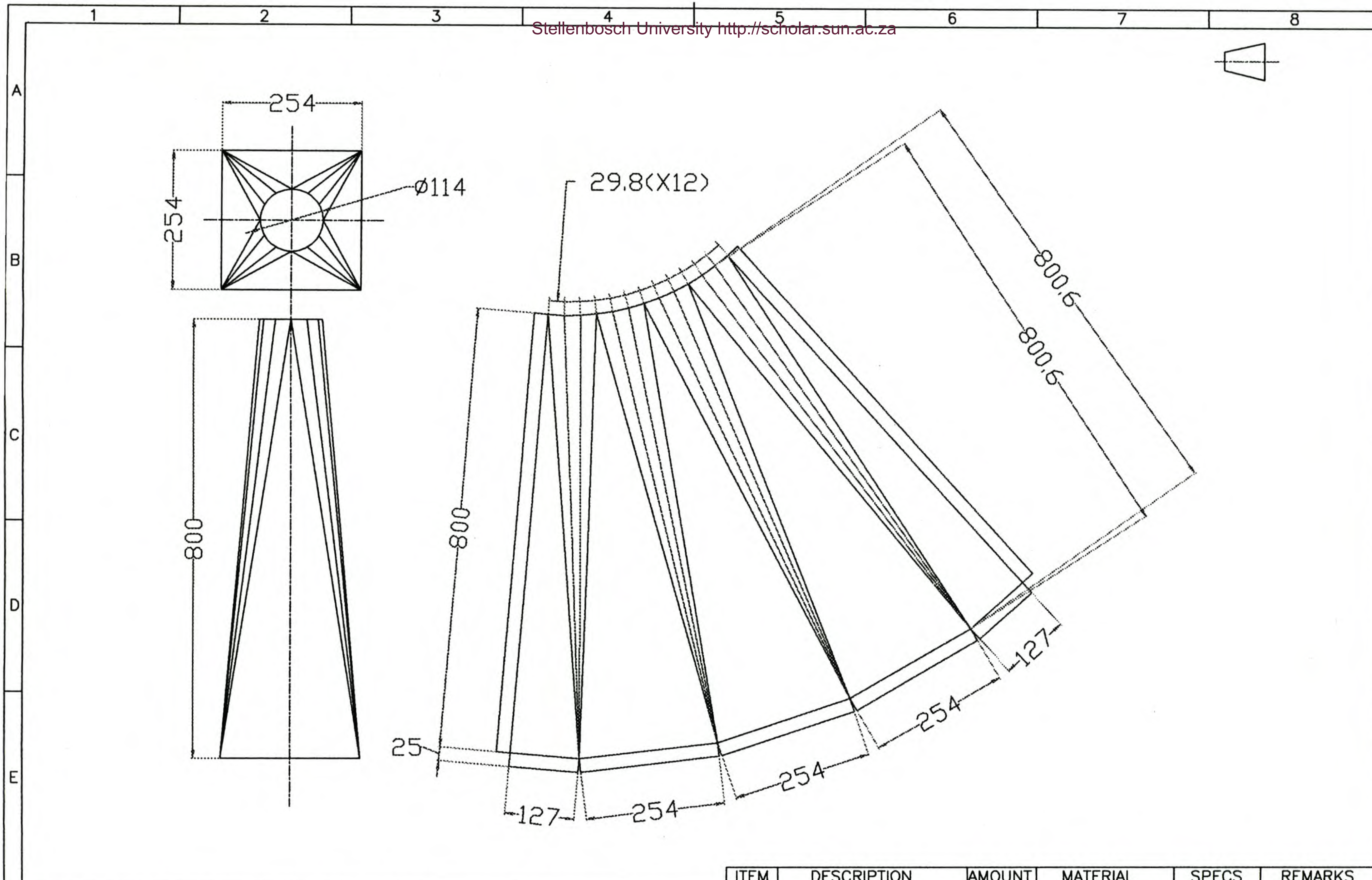
UNIVERSITY OF STELLENBOSCH

SCALE MEAS IN mm	TITLE : **STEAM SEPARATOR**



ITEM	DESCRIPTION	AMOUNT	MATERIAL	SPECS	REMARKS

UNIVERSITY OF STELLENBOSCH	SCALE MEAS IN mm	TITLE HEAT EXCHANGER PIPING CONNECTION
STUDENT No. 9969578 — DRAWN BY D. LOTUN	DATE 06.03.01	SHEET No. 1 OF 2 SHEETS — No.



ITEM	DESCRIPTION	AMOUNT	MATERIAL	SPECS	REMARKS

UNIVERSITY OF STELLENBOSCH

SCALE
MEAS IN mm

TITLE **DEVELOPMENT OF TRANSITION PIECE**