FINAL YEAR PROJECT REPORT

UNIVERSITY OF NAIROBI

DEPARTMENT OF MECHANICAL AND MANUFACTURING ENGINEERING

Project title:

DESIGNING A BOILER CHIMNEY HEAT RECOVERY SYSTEM AGAINST FOULING

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DECLARATION STATEMENT

We declare that any information in this report, except where indicated and acknowledged, is our original work and has not been presented before to the best of our knowledge.

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They were informative and cooperative whenever we required their technical assistance in their respective fields.

Finally we would like to appreciate our parents, siblings , and friends for their support throughout this project.

Karanja Samson Ngugi.

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ABSTRACT
An initial design of chimney heat recovery heat exchanger was provided. The design had a completely fabricated exchange core but an incomplete ducting system .

This report is based on the work undertaken to complete and test the gas to gas heat recovery system. This system was specifically designed for boiler chimney and therefore the systems ducting was designed to conform to the general boiler stack.

In the completion of the design, the major factor to consider was to design against fouling. The system was therefore designed with means of reducing fouling such as provision for easily replaceable particulate filter and quick washing system.
The project was hence done in the following manner.

1. Completing of the fabrication.
2. Research on ways of minimizing fouling .
3. Incorporating the ways arrived at in 1 above into the system design.
4. Testing of the model under forced convection condition.

The gases from a furnace were used to simulate industrial flue gases. The performance of the model was used to project the optimum of prototype.

LIST OF TABLES

LIST OF GRAPHS

LIST OF SYMBOLS AND ABREVIATIONS.

A_a: Air side total surface area.

A_c: Exchanger minimum free flow area.

A_{fr}: Heat exchanger frontal area.

A_g: Gas side total surface area.

A: Surface area of the heat exchanger surfaces.
C_p: Specific heat capacity of air at constant pressure.

F: Correction factor for the heat exchanger.

H_1: Height of the exchanger core.

h_i: Convective heat transfer coefficient of the hotter side.

h_o: Radioactive heat transfer coefficient.
K: Thermal conductivity of the exchanger material.
L: Length.
Q: Overall heat transfer rate.

R: Total thermal resistance from inside to outside flow.

r: Radius.

$T_{a\ in}$: Air inlet temperature.

$T_{a\ out}$: Air outlet temperature.

$T_{g\ in}$:Gasinlet temperature.

$T_{g\ out}$: Gasoutlet temperature.

Tr: Room temperature.

t : Plate thickness.

U: Overall heat transfer coefficient.
V: Flow velocity.

W: Width of the exchanger core.

LIST OF GREEK SYMBOLS.

β:The ratio of the heat transfer surface area of a heat exchanger to its volume is called the area density.

α:Ratio of heat transfer area on one side of a plate exchanger to total volume between the plate on that side.

θ: Time.

θ^*: Normalized time.

θ_d: Dwell time.

ε :Effectiveness .

$\dot{m}$: Mass flow rate.

ρ: Fluid density.

$\Delta T_{m.}$: The log mean temperature difference.

OBJECTIVE STATEMENT

The aim of this project was to recover heat lost through flue gases exhaust at the chimney stage taking a keen consideration of the effect of fouling especially at the core of the heat exchanger. Some research was done and the exchanger system designed and fabricated though not to completion. It was nevertheless tested specifically to determine its heat exchange effectiveness. However critical factors such as fouling were not keenly observed. The small plate spacing of the exchange core will allow for a substantial heat recovery. This obviously means the core will undergo fouling at a higher rate as compared to boiler tubes. This makes the exchanger to require more frequent maintenance than the normal boiler maintenance.

The objective was to review the design ensuring that fouling was reduced and that the maintenance practice on the exchanger does not adversely interfere with the normal operation of the boiler.

It was projected that the project will maintain its goal of recovering heat and hence its benefits towards energy management and at the same time maintain the smooth operation of the boiler.

The aim of this project can therefore be summarized as

1. Complete the fabrication of the heat recovery system and test.
2. Research on fouling effects for different fuels used in boilers.
3. Minimizing fouling and reduce maintenance requirements to avoid interference with the normal operations of the boiler.
4. Give the recommendations based on the prototype performance

CONTENTS

CHAPTER FIVE

CHAPTER SIX

CHAPTER SEVEN

CHAPTER ONE

1.0 INTRODUCTION

1.1. INDUSTRIAL WASTE HEAT.

This is heat lost in industries through ways such as discharge of hot combustion gases to the atmosphere through chimneys, discharge of hot waste water, heat transfer from hot surfaces. This energy loss can be recovered through heat exchangers and be put to other use such as preheating other industrial fluids such as water or air.

This project focuses on recovering heat that is lost through boiler chimney flue gas. The advantages of heat recovery include:

i). Increasing the energy efficiency of the boiler.

ii). Decreasing thermal and air pollution dramatically.

1.2 DESIGN CONSIDERATIONS

In the designing of the exchanger following factors were put to consideration.

1. The exchanger surface has to be the most efficient and suitable for gas-gas heat exchange.
2. The design has to consider the fouling effect of the flue gases.
3. The design has to allow for quick maintenance without interfering with the boiler operations.
4. The ducting design has to conform to the boiler chimney design.

Based on the above factors, the exchanger was designed to be of compact plate type. Various designs for the exchange core were considered including cylindrical type (ducts).

The plate type was found to be more efficient and simpler in design. It was also more suitable for gas - to gas heat exchange as it offers higher surface for heat transfer.

1.3 CHALLENGES TO RECOVERING LOW TEMPERATURE WASTEHEAT (HODGE B.K, 1990)

Corrosion of heat exchanger surface: as the water vapor contained in the exhaust gas cools some of it will condense and deposit corrosive solids and liquids on the heat exchanger surface. The heat exchanger must be designed to withstand exposure to these corrosive deposits. This generally requires using advanced materials, or frequently replacing components of the heat exchanger, which is often uneconomical.

Large heat exchanger surface required for heat transfer; since low temperature waste heat will involve a smaller temperature gradient between two fluid streams, larger surface areas are required for heat transfer. This limits the economy of heat exchangers.

Finding use for low grade heat: recovering heat in low temperatures range will only make sense if the plant has use for low temperature heat.

CHAPTER TWO

2.0LITERATURE REVIEW

2.1 INTRODUCTION

Heat exchangers are devices that facilitate the exchange of heat between two fluids that are at different temperatures while keeping them from mixing with each other. Heat transfer in heat exchangers involves convection in each fluid and conduction through the wall separating the two fluids. In order to account for the contribution of all the effects of convection and conduction, an overall heat transfer coefficient, U, is used in the analysis. Heat transfer rate depends on the temperature differences between the two fluids at the location and the velocity of the fluids (time of interaction) between the fluids.

2.2TYPES OF HEAT EXCHANGERS

Due to the different types of applications for heat exchanges, different types of hardware and different configurations of heat exchanges are required. This has resulted to different designs of heart exchangers which includes and not limited to.

2.2.1Double pipe heat exchanger (simplest heat exchanger)

Consists of two concentric pipes of different diameter. In application, one fluid passes through the pipe of smaller diameter while the other flows through the annular space between the two pipes. The flow of fluids can be arranged into:-

i). **Parallel flow.(***Cengel, 2002***)**

Both fluids (hot fluid and cold fluid) enter the heat exchanger at the same end and move in the same direction to leave at the other end as shown in the figure below.

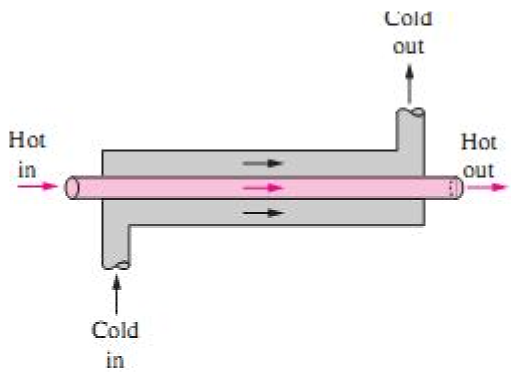


Fig a (i)

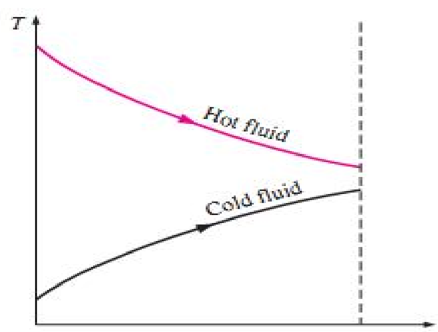


Fig a (ii)

Fig a. (i) shows the flow regimes while fig a (ii) shows the associated temperature profiles.

(**ii**). **Counter flow**(*Cengel, 2002*)

In these types of arrangement, the cold and hot fluids enter the exchanger at opposite ends and flow in opposite directions as shown in the figure below:

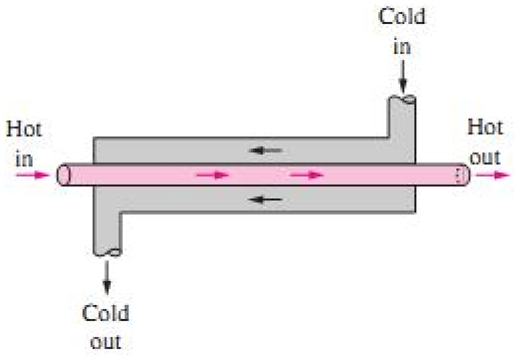


Fig b(i)

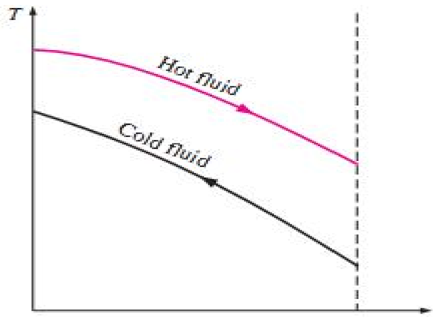


Fig b(ii)

Figure b (i) shows the flow regimes and figure b (ii) shows the associated temperature profiles.

2.2.2 The compact heat exchanger

This type of heat exchanger is designed to allow a large heat transfer surface area per unit volume. The ratio of the heat transfer surface area of a heat exchanger to its volume is called the area density β. Heat exchangers with β >700 are classified as compact heat exchanger e.g. car radiator, human lung am0ongest others. They allow high heat transfer rates between fluids in a small volume. They are therefore best suited for applications with strict limitations on the weight and volume of heat exchanger. They are mostly used in gas-to-gas and gas-to-liquid heat exchanger to counteract the low heat transfer coefficient associated with fluid flow with increased surface area. The two fluids in this type of heat exchangers move in directions perpendicular to each other, a flow configuration referred to as cross-flow. This type of flow may be classified as unmixed or mixed.

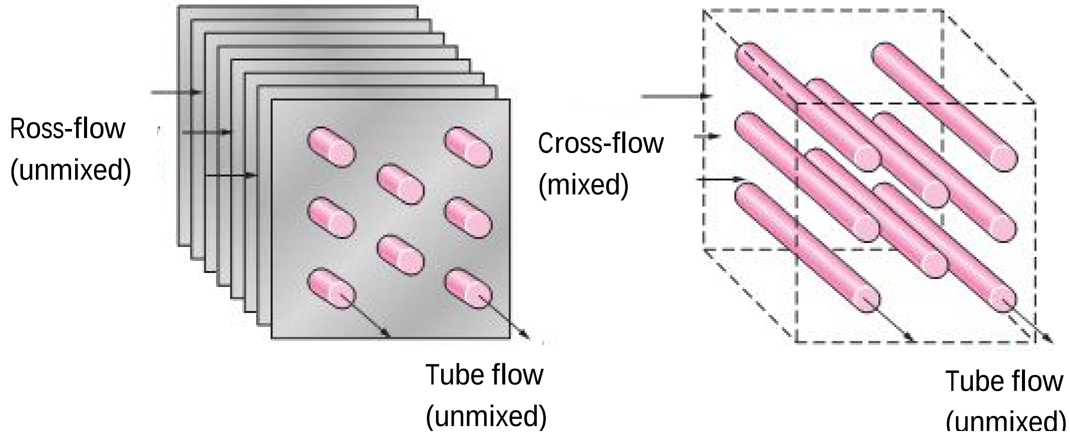
i). Unmixed flow

Plate fins force the fluid to flow through a particular inter-fin spacing and prevent it from moving in the transverse direction.

ii). Mixed flow

The fluid is free to move in the transverse direction. The presence of mixing can have adverse and significant effects on the heat transfer characteristics of a heat exchanger.

(Cengel, 2002)



(Ozisk, 1985)

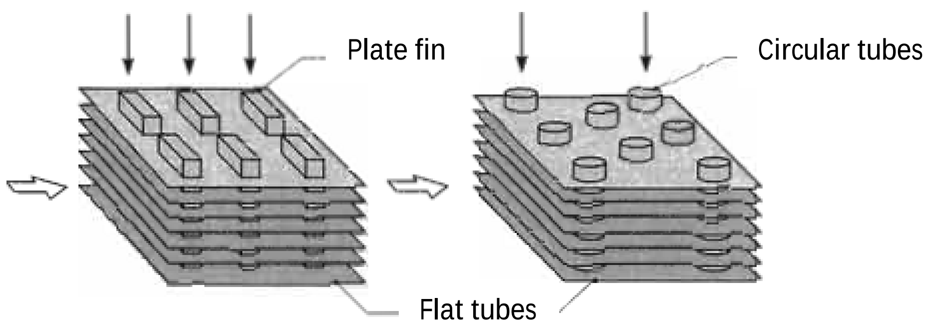


Fig c. compact heat exchangers

2.2.3 Shell and tube heat exchanger

Contains a large number of tubes packed in a shell with their axes parallels to that of the shell. One fluid flows through the tubes while the other flows through the shell but outside the tubes. Baffles' placed in the shell increases the flow time of the shell-side fluid by forcing it to flow across the shell thereby enhancing heat transfer in addition to maintaining uniform spacing between the tubes.These baffles are also used to increase the turbulence of the shell fluid. The tubes open to some large flow areas called header at both ends of the shell. These types of heat exchanger can accommodate a wide range of operating pressures and temperatures. They are easier to manufacture and are available at low costs. Both the tube and shell fluids are pumped into the heat exchanger and therefore heat transfer is by forced convection. Since the heat transfer coefficient is high with the liquid flow, there is no need to use fins. They can also be classified into parallel and counter flow types.

(Ozisk, 1985)

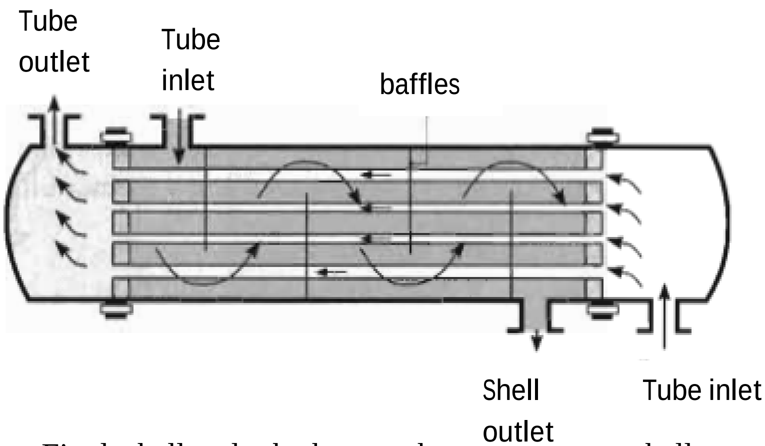


Fig d. shell and tube heat exchanger with one shell pass and one tube pass (cross-counter flow configuration)

2.2.4 Plate heat exchangers*(Ozisk, 1985)*

They are usually constructed of thin plates which may be smooth or corrugated. Since the plates cannot sustain as high pressure and or temperatures as circular tubs, they are generally used for small and low to moderate pressure/temperatures. Their compactness factor is also low compared to other types of heat exchangers. The plates can be arranged in such a way that there is cross-flow i.e. the hot and cold fluids flowing in directions perpendicular to each other to enhance the heat transfer characteristic.

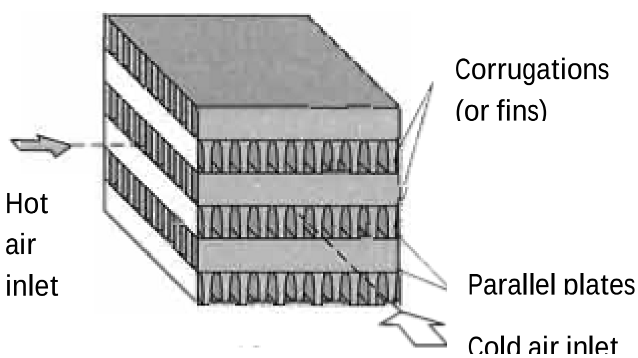


Fig e. plate type compact heat exchanger (cross flow)

2.2.5 Other technologies applied to waste heat recovery

1.2.51<u>Regenerators</u>

This is a type of heat exchanger where heat from the hot fluid is intermittently stored in a thermo storage medium before it is transferred to the old fluid. In this type of heat exchanger can be the same fluid. The fluid may go through an external processing step and then it is flowed back through the heat exchanger in the opposite direction for further processing

1.2.5.2<u>Recuperators.</u>

It is a counter-flow energy recovery heat exchanger used in industrial processes to recover waste heat.

2.2.5.3 Thermal wheel.

A rotary heat exchanger consists of a circular honeycomb matrix of heat absorbing material which is slowly rotating within the supply and exhaust air streams of an air handling system.

2.2.5.4 Economizer.

In case of process boilers, waste heat in the exhaust gas passed along a recuperator that carries the inlet fluid for boiler and thus decrease energy intake of the inlet fluid.

2.2.5.5 Run around coil.

Comprises 2 or more multi-raw finned tube coils connected to each other by pumped pipe work circuit.

2.3 OVERALL HEAT TRANSFER COEFFICIENT

In analysis of heat transfer in heat exchangers, various thermal resistances in the path of heat flow from the hot to cold fluid are combined.

Heat is first transferred from the hot fluid to the wall by convection, through the mass by conduction and from the wall to the cold fluid by convection. Any radiation effects are usually included in the convection heat transfer coefficients.

The total thermal resistance, R, for the whole system is given by:-

R= thermal resistance of inside flow + thermal resistance of the systems material + thermal resistance of outside flow

$$R = \frac{1}{A_i h_i} + \frac{1}{A_o h_o} + \frac{t}{kA_m}$$

Where h_i, h_o= heat transfer coefficients for inside and outside flow respectively

k= Thermal conductivity of the exchanger material

R= Total thermal resistance from inside to outside flow

t= Thickness of the heat exchanger material

$$A_m = \frac{A_o - A_i}{ln(\frac{A_o}{A_i})} = \text{ logarithmic mean area, } m^2$$

A_i, A_o= Inside and outside surface areas of the heat exchanger surfaces respectively.

Expressing the thermal resistance R as an overall heat transfer coefficient based on either the fluid inside or outside surface of the heat exchanger surface areas:-

$U_o = \frac{1}{A_o R}$ and $U_i = \frac{1}{A_i R}$

If the wall thickness is small and its thermal conductivity is high the material resistance may be neglected and hence the overall heat transfer coefficient becomes:-

$$U_t = \frac{1}{\frac{1}{h_i}+\frac{1}{h_o}}$$

In applications of heat exchangers, accumulation of deposits mostly from combustion, on the heat exchanger surface causes additional thermal resistance, a condition known as fouling. Effects of fouling are introduced in the heat transfer coefficient in the form of a fouling factor. The total thermal resistance then became:-

$$R = \frac{1}{A_i h_i} + \frac{F_i}{A_i} + \frac{t}{kA_m} + \frac{F_o}{A_o} + \frac{1}{A_o h_o}$$

Where F_i and F_o are the fouling factors on the inside and outside surfaces respectively.

2.4 FOULING FACTOR

The performance of heat exchangers usually deteriorates with time due to accumulation of deposits on the heat transfer surface. The accumulation of deposits leads to increased resistance to heat transfer and causes the rate of heat transfer in a heat exchanger to decrease. This accumulation of deposits on the heat transfer surface is known as fouling and the net effect of fouling is represented by a fouling factor, R_f, which is a measure of the thermal resistance introduced by fouling.

CHAPTER THREE

3.0FOULING

3.1 INTRODUCTION

Fouling is a general term that includes any kind of deposits of extraneous material that appears on the heat transfer surface during the lifetime of the heat exchanger.

Fouling reduce heat transfer across the exchanger surface hence reduces efficiency of the heat exchanger. The foulingdeposits also reduce flow cross-section area causing a pressure deferential across the heat exchanger which in turn increasing on the fan power required. It might also eventually block the heat exchanger.

Different kinds of fuel produce different degrees of fouling .most fuel produce just soft black soot that get deposited on the exchanger surface.

This can easily be removed by brushing and sand washing. However lower grade fuel oil(principally no.6.oil or resid)contain large quantities of alkaline sulfates and vanadium pentoxide that causes scaling due to their lower fusion temperatures.

The forms of fouling may therefore include

- Particulate fouling.
- Scaling/precipitation.
- Chemical/corrosion fouling.
- Solidification.

2.5 FORMS OF FOULING

2.5.1 Scaling/precipitation: - scaling/precipitation occur as a result crystallization of dissolved substance on to the heat transfer surface. These deposits can be removed by scratching or by cleaning via chemical treatment. This is the most common type of fouling.

Scaling/precipitation can be reduced by treating the fluid flowing past the heat exchanger before it reaches the heat exchanger surface.

2.5.2 Particulate fouling

This result from the accumulation of solid particles suspended in the process fluid onto the heat transfer surface. Such solid particles can be removed by use of filters to treat the process fluid before it reaches the heat exchanger surface.

2.5.3 Chemical /corrosion fouling

In this case, the surfaces are fouled by accumulation of the products of chemical reactions on the surfaces. This form of fouling can be avoided by coating the heat exchanger surfaces by glass. Heat exchanger surfaces can also be fouled by growth of algae in warm fluids (chemical fouling) which can be prevented by chemical treatment.

2.5.4 Solidification fouling

The crystallization of a pure liquid or one component of the liquid phase on a sub cooled heat transfer surface.

The mechanism of fouling is complicated and no reliable techniques are available but there are means of reducing fouling. The methods mostly used to reduce fouling include use of filters and increasing the fluid flow to ensure turbulent flow.

3.2 DESIGN AGAINST FOULING

It was our duty to consider the effect of fouling upon the heat exchanger performance during the desired operation lifetime and make provisions in our design for sufficient extra capacity to ensure that the exchange will meet process specifications upto shut down for cleaning.

We were also to consider the mechanical arrangements that are necessary to permit easy cleaning.

In our design, the following measures have been taken to reduce the rate of fouling.

I. Provision for particulate filters.
II. Introduction of turbulent flow upstream of the exchanger core.

3.2.1 Provision of particulate filters

At the entry of the flue gas duct is attached, a cone shaped duct to whose narrower end can be attached diesel particulate filter. The particulate filter is designed to remove fuel particulate matter (soot) from the fuel gases. The efficiency of the filter is inversely proportional to the pressure that is build up due to resistance to gas flow. It is therefore difficult to achieve 100 percent efficiency through filtration, as there must be a compromise between efficiency and pressure buildup .the best filters are therefore broad band filters that can filter particles of diameters between 0.2-150µm.

The filters can easily be removed through a door on the side of the side duct for cleaning.

3.2.2Introduction of turbulent flow upstream of the exchange core

The cone shape element at the gas-duct entry causes turbulence as it suddenly opens into the larger gas duct .this causes turbulence. This turbulent flow of air picks with it some of the particles that stick on the exchanger surface due to its drag effect. This helps to reduce on fouling.

The above filtration and turbulence only minimizes rate of fouling. But the fouling still takes place. This therefore implies that the exchanger will require maintenance (cleaning). There are various ways that could be used in cleaning the exchanger. In the design we consider using the following methods.

1. Blowing
2. Washing

The system was designed with a slit on the wall of the flue gases duct downstream of the exchanger .This allows the overhead water washing.

Pressurized water mixed with abrasives e.g. fine sand is used to remove soot that cannot be removed by blowing air past the exchanger. The abrasives help in scrubbing the surface.

Before washing, the particulate filter is removed and replace with alid to prevent water from entering the broiler.

During washing the waste water drains out of the system through the outlet ducts at the base of the flue gas inlet duct.

CHAPTER FOUR

4.0 THE HEAT EXCHANGER SYSTEM DESCRIPTION

4.1 COMPONENTS AND PROPERTIES

Every part of the system is described and explained in this section. From the previous report, the system was designed to have the following characteristics

TABLE 4

DESIGN PARAMETERS OF THE HEAT EXCHANGER.

Core dimensions	L_1 L_2 L_3	0.3m 0.2m 0.3m
Plate spacing	mm	3
Air temperature at inlet	°C	26
Air temperature at outlet	°C	200
Gas temperature at inlet	°C	450
Gas temperature at outlet	°c	200
A_{ft}.	m^2	0.06
A_{fr}	m^2	0.09
Core volume	m^3	0.018
Plate thickness	b mm	1.83
α_a	m^{-1}	615.555
α_g	m^{-1}	613.888
A_a	m^2	11.08
A_g	m^5	11.05
Effectiveness		0.75
U_a	W/m^2K	3.683
R		0.96
Number of passages	Air side Gas side	23 23

4.1.1 Funnel shaped duct

This is a short cone shaped duct that is welded of the entry to the flue gas duck upstream of the exchanger core**.**

It provides an end that can be covered by alid during cleaning to prevent water from entering the boiler. Small holes are left at its joint to the gas duct to allow water out.

During normal boiler operation gas filler can be put at this narrow end to trap carbon particles from reaching the exchanger core. It was constructed from mild steel sheet (16 gauge)

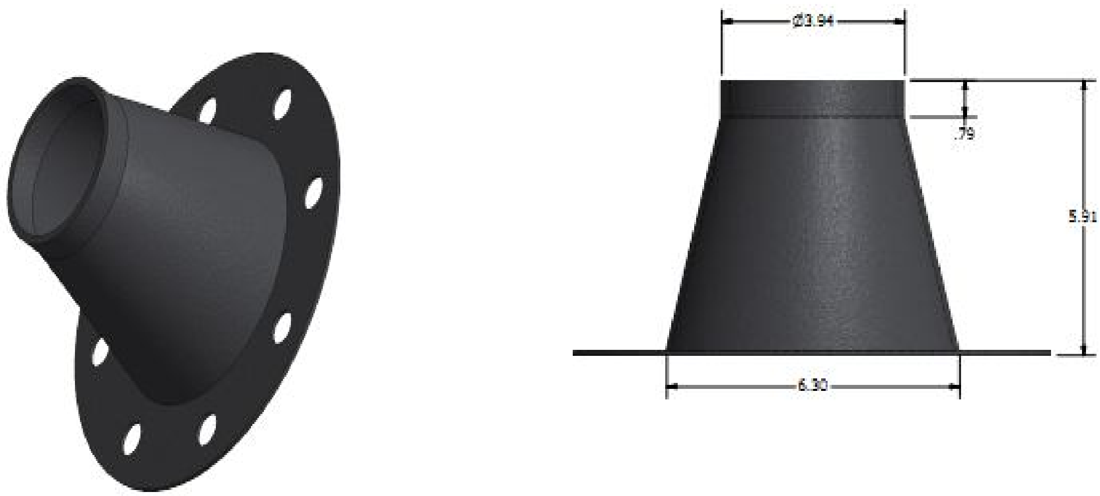
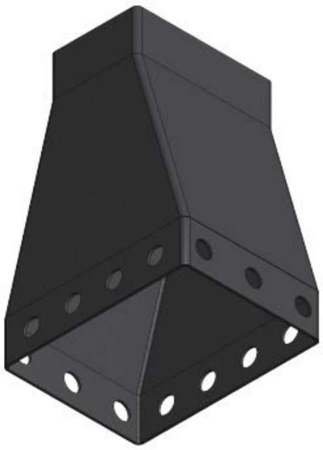


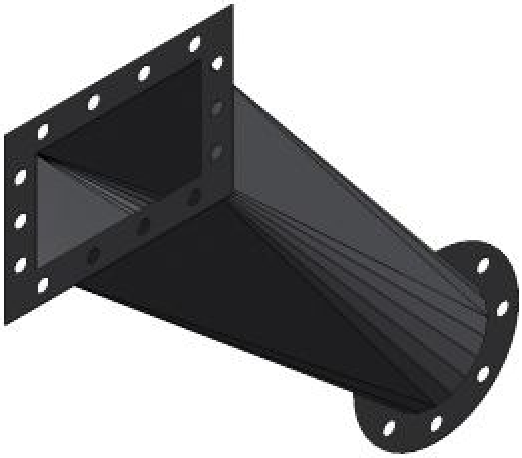
Fig f. Funnel shaped duct

4.1.2Ducts

The system has four ducts which are

a) Air inlet duct made from galvanized iron sheet (32 gauge)
b) Air outlet duct also made from galvanized iron sheet (32gauge) and is connected to the air fan.
c) Gas inlet duct constructed from mild steel sheet (16gauge) to be able to support the whole system and the rest of the chimney above.
d) Gas outlet duct: also constructed from mild steel sheet (16gauge) so as to hold the rest of the chimney above the exchange.





Air duct Flue gas duct

Figs.j: showing air duct and flue gas duct.

4.1.3Heat exchanger core:

It consists of 134 parallel aluminum plates (32gauge) of dimensions 0.3m by 0.2m enclosed in a mild steel flanged rectangular frame.

The plates were sealed alternately using silicon sealant. The sealant has good heat resistance properties. The sealant also has good elastic properties that ensure sealing even during plate expansion. Galvanized binding wires were used to hold the exchange core fit in the frame and at the centre by aluminum spacers that ensured even spacing between plates.

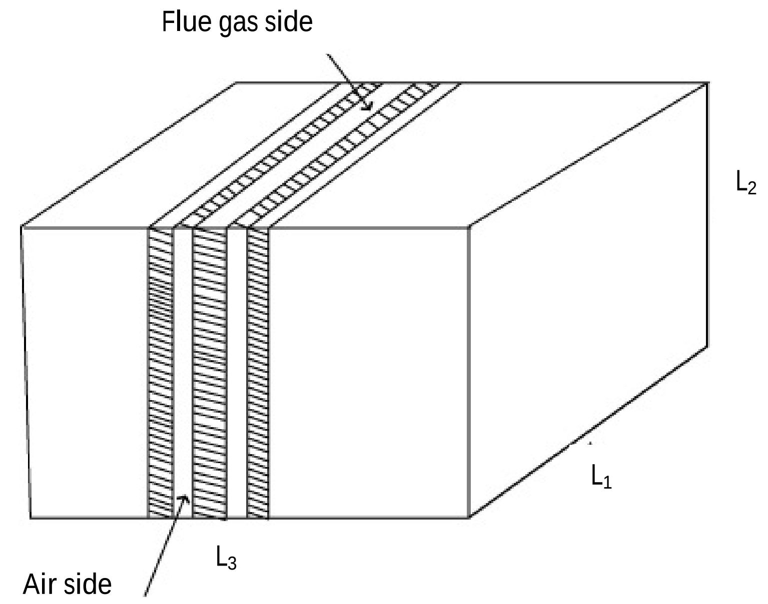


Fig 4.2 Heat exchanger core.

4.1.4 Draught system

If consists of a electric motor driven (centrifugal) fan which operate at constant speed of 1389 rpm. This fan was driven by a single phase synchronous motor.

CHAPTER FIVE

5.0MODEL TESTS, RESULTS AND ANALYSIS

The tests carried out were:-

i) Forced convection test with varying air flow

ii) Transient test

5.1 FORCED CONVECTION TEST FOR DIFFERENT AIR FLOW RATES.

- The heat exchanger was connected to the air inlet fan and the furnace.
- The air flow control flap was set at 75^0.
- Room temperature was measured and recorded as T_r.
- Air inlet fan was switched on and the air velocity was measured by use of an air velocity meter.
- The furnace was fired and temperatures of the gas at inlet and outlet were measured and recorded.
- The air control damper was then changed to 60^0, 45^0, 30^0, 15^0 and 0^0 and their respective temperature readings and air velocities taken and recorded.
- The results were recorded in table 3.1.

5.2 TRANSIENT TESTS

The objective here was to determine the time it took for the exit temperature to reach a constant value. (Time response).

5.2.1Procedure

- The air control valve was set at 0^0 and the air inlet fan switched on.
- Stop watch was set at 0.
- The air inlet fan was switched on and the furnace lit.
- Temperature readings in each duct were taken at 2minutes interval until the temperatures reached a steady value.
- The readings were recorded in table 5.3

5.3 RESULTS

Table 5a

Flow rate against temperatures.

Air controldamper angle	Average velocity Of air	T_{air}Inlet(^{0}C)	T_{air} (^{0}C)Outlet	$T_{fluegas}$Inlet(^{0}C)	$T_{fluegas}$Outlet(^{0}C)
75^0	0.3	22.7	205	480	249
60^0	0.395	22.7	194	482	253
45^0	0.49	22.	190	483	255
30^0	0.58	22.7	184	483	256
15^0	0.70	22.7	171	483	25
0^0	0.95	22.7	170	483	260

Table 5b

Air flow rate

Velocity near duct wall (m/s)	Velocity at duct center (m/s)	Angle of air control flap	Average velocity (m/s)
0.15	0.45	75	0.3
0.19	0.60	60	0.395
0.26	0.72	45	0.49
0.32	0.84	30	0.58
0.44	0.96	15	0.70
0.60	1.30	0	0.95

Other results.

Flue gas velocity =3.7m/s

Flue gas supplier duct diameter =15.24cm

Table 5c

Transient test

Time (s)	$T_{g\ in}$	$T_{g\ out}$	$T_{a\ in}$	$T_{a\ out}$
0	_	_	22.7	22.7
2	400	250	22.7	200
4	490	260	22.7	205
6	500	265	22.7	202
8	500	266	22.7	207
10	500	270	22.7	210

5.4Analysis.

5.4.1 Calculation of volume flow rate.

Flue gas volume flow rate $=\pi\ r^2.V$

$3.142 \times (\frac{15.24}{2})^2 \times 10^{-4} \times 3.7 = 6.749 \times 10^{-2}\ m^3/s$

Volume flow rate of air$A = l \times w \times V$

Area of duct = $(20 \times 21) \times 10^{-4}$

$4.20 \times 10^{-2}\ m^2$

Maximum volume flow rate of air

= Area of duct × velocity at $\Theta=0^0$

$= 4.2 \times 10^{-2} \times 0.95$

$= 3.99 \times 10^{-2}\ m^3/s$

As the air flow rate at fully open duct is still low compared to the flue gas flow rate, it was decided that the subsequent test be carried out at flap angle of 0^0 in the air duct.

5.4.2 Major parameters of interest;-

1. Overall heat transfer rate Q
2. Effectiveness ε

3. Overall heat transfer coefficient U

(1) Overall heat transfer rate (Q)

$Q = \dot{m}C_p\ (T_{aout} - T_{ain})$

Where M- Mass flow rate of air

C_p-Specific heat capacity of air at constant pressure

T_{aout}&T_{ain}- Temperatures of air at outlet and inlet of the heat exchanger respectively.

NOTE: C_p was taken at average/mean temperature

$[(T_{aout} + T_{ain}) \div 2]$

Mass flow rate of air is given by;

$\dot{m}$= density of air $\rho_a \times$ Duct area A. × Air velocity

Mass flow rate of flue gases is given by

$\dot{m}$= density of gas $\rho_g \times$duct area $A_g \times$ gas velocity V_g

Density of gas $\rho_g = \frac{P}{RT}$

Ambient conditions were: pressure=24.4 in Hg

Temperature=22.7 ^{0}C.=295.7

Pressure in meters of mercury= 0.61976mHg

Pressure in Pa is: 0.61976× 13600× 9.81= 82685.9Pa

$$\frac{82685.9}{287*295.7} = 0.9743 Kg/m^3$$

(2) Overall heat transfer coefficient (u)

$$U = \frac{Q}{FA\Delta T_m.} = ; \Delta T_m = \frac{(T_{g\ in} - T_{a\ out}) - (T_{g\ out} - T_{a\ in})}{ln(\frac{T_{g\ in} - T_{a\ out}}{T_{g\ out} - T_{a\ in}})}$$

(3) Effectiveness ε

$$\text{Effectiveness} = \frac{Actualheattransferrate}{Maximumheattransferratepossible}$$

$$\varepsilon = \frac{Cc\ (T_{\text{a out}} - T_{a\ in})}{Cmin\ (T_{\text{g in}} - T_{a\ in})}$$

5.4.3 Transient Test

Important parameters include

Dwell time Θ_d

Normalized time Θ^*

Time to reach steady state Θ

Dwell time is the time air is in contact with the heat transfer surface from entry to exit of exchanger core. It is given by: $\Theta_d = \frac{L}{V}$ where L= length of heat exchanger from inlet to outlet to exit.

$$V = \frac{M}{\rho Ac \times \text{number of passages}}$$

ρ =density of air

A_c=Exchanger minimum free flow area.

The normalized time is the ratio of time the air temperature takes to reach a constant value to the dwell time i.e.

$$\Theta^* = \frac{\theta}{\theta_d}$$

5.4.4 Sample calculations

Mass flow rate $\dot{m} = \rho AV$

$=0.9743 \times 0.042 \times 0.95$

$=0.04724$ kg/s

Q for flap angle $=0^0$

$=\dot{m}C_p\ (T_{aout} - T_{ain})$

$=0.03887 \times 1.005\ (170-22.7)$

$=5.7542 kW/m^2$

$U=\frac{Q}{FA\Delta T_m}$. Where F= is correction factor for cross flow (both fluids unmixed).

$$R=\frac{(T_{g\ in}-T_{g\ out})}{(T_{a\ out}-T_{a\ in})} \quad , P=\frac{(T_{a\ out}-T_{a\ in})}{(T_{g\ in}-T_{a\ in})}$$

Hence $R=\frac{(483-260)}{(170-22.7)}= 1.5139$, $P==\frac{(170-22.7)}{(483-22.7)}= 0.32$

For the above values of P and R, F=0.94 (from charts)

$$\Delta T_m= \frac{(T_{g\ in}-T_{a\ out})-(T_{g\ out}-T_{a\ in})}{ln\ (\frac{T_{g\ in}-T_{a\ out}}{T_{g\ out}-T_{a\ in}})}$$

$$=\frac{(483-170)-(259-22.7)}{\text{In}\ (\frac{483-170}{259-22.7})}$$

$=272.85^0C$

$$U=\frac{5.7832}{0.94 \times 0.042 \times 272.85}$$

$= 0.5369 kW/m^2K$

Effectiveness

$$\varepsilon =\frac{C_c\ (T_{a\ out}-T_{a\ in})}{C_{min}\ (T_{g\ in}-T_{g\ out})}$$

Dwell time

$\Theta_d = \frac{L_2}{V} =\frac{0.3}{0.95}= 0.316$ seconds, where L_2 is the length of the exchanger core from air inlet to air outlet.

Θ = 6min = 360 seconds

$$\frac{\Theta}{\theta_d} = \frac{360}{0.316} = 1140$$

Table 5d

DETERMINING CROSS-FLOW CORRECTION FACTORS

$\dot{m}=\rho UA$ kg/s	$T_{a\,in}\ {}^0C$	$T_{a\,out}\ {}^0C$	$T_{g\,in}\ {}^0C$	$T_{g\,out}\ {}^0C$	p	R	F	LMTD
0.01228	22.7	205	480	249	0.40	1.27	0.93	249.8595
0.01616	22.7	194	482	253	0.37	1.34	0.925	258.0758
0.02005	22.7	190	483	255	0.36	1.36	0.92	261.4768
0.02373	22.7	184	483	256	0.35	1.41	0.93	264.7929
0.02864	22.7	171	483	257	0.32	1.52	0.935	271.2981
0.03887	22.7	170	483	260	0.32	1.51	0.935	275.0789

Table 5e

DETERMINED VALUES OF Q AND U.

$\dot{m}=\rho UA$ kg/s	$T_{a\,in}\ {}^0C$	$T_{a\,out}\ {}^0C$	Mean Temp K	C_pkJ/kg K	ΔT	A_{air}	Q kW	U kW/m^2K
0.01228	22.7	205	386.85	1.01534	182.3	0.042	2.2730	0.7966
0.01616	22.7	194	381.35	1.01300	171.3	0.042	2.8042	0.7269
0.02005	22.7	190	379.35	1.01234	167.3	0.042	3.3958	0.7041
0.02373	22.7	184	376.35	1.01114	161.3	0.042	3.8703	0.6623
0.02864	22.7	171	369.85	1.01011	148.3	0.042	4.2903	0.6749
0.03887	22.7	170	369.35	1.01006	147.3	0.042	5.7832	0.5369

Table 5f

DETERMINATION OF EFFECTIVENESS

T_a in ^0C	T_a out ^0C	Mean temp. K	C_p air kJ/Kg.K	T_g in ^0C	T_g out ^0C	Mean temp for gas K	C_p gas kJ/Kg.K	ṁ of air kg/s	ṁC_p of air kW/K	ṁ of gas kg/s	ṁC_p	Effectiveness, ε
22.7	205	386.8	1.01534	480	249	637.5	1.05995	0.01492	0.01515	0.03193	0.033842	0.3969
22.7	194	381.4	1.01300	482	253	640.5	1.06066	0.01964	0.01990	0.03193	0.033864	0.3730
22.7	190	379.3	1.01234	483	255	642.0	1.06101	0.02437	0.02467	0.03193	0.033876	0.3635
22.7	184	376.4	1.01114	483	256	642.5	1.06113	0.02884	0.02912	0.03193	0.033879	0.3504
22.7	171	369.8	1.01011	483	257	643.0	1.06125	0.03481	0.03516	0.03193	0.033883	0.3222
22.7	170	369.3	1.01006	483	260	644.5	1.06160	0.04724	0.04772	0.03193	0.033894	0.3200

Table 5g

DETERMINATION O DWELL TIME AND NORMALIZED TIME

ṁof air kg/s	Mass/passage	velocity	Dwell time θ_d (s)	T_a out ^0C	Time θ	$\theta^*(\theta/\theta_d)$
0.04724	0.002054	0.95	0.316	205	0	0
0.04724	0.002054	0.95	0.316	194	2	379.7468
0.04724	0.002054	0.95	0.316	190	4	759.4937
0.04724	0.002054	0.95	0.316	184	6	1139.2405
0.04724	0.002054	0.95	0.316	171	8	1518.9873
0.04724	0.002054	0.95	0.316	170	10	1898.7342

Table 5h

DETERMINATION OF PERCENTAGE HEAT RECOVERED

ṁ=ρUA kg/s	Q_{air} kW (At exit)	Total$Q_{fluegas}$kW (at inlet)	Fraction of heat recovered	Percentage heat recovered
0.01492	2.7616	25.4830	0.1084	10.84
0.01964	3.4081	25.5673	0.1333	13.33
0.02437	4.1274	25.6103	0.1612	16.12
0.02884	4.7037	25.6125	0.1836	18.36
0.03481	5.2145	25.6155	0.2036	20.36
0.04724	7.0285	25.6239	0.2743	27.43

TABLE 5i

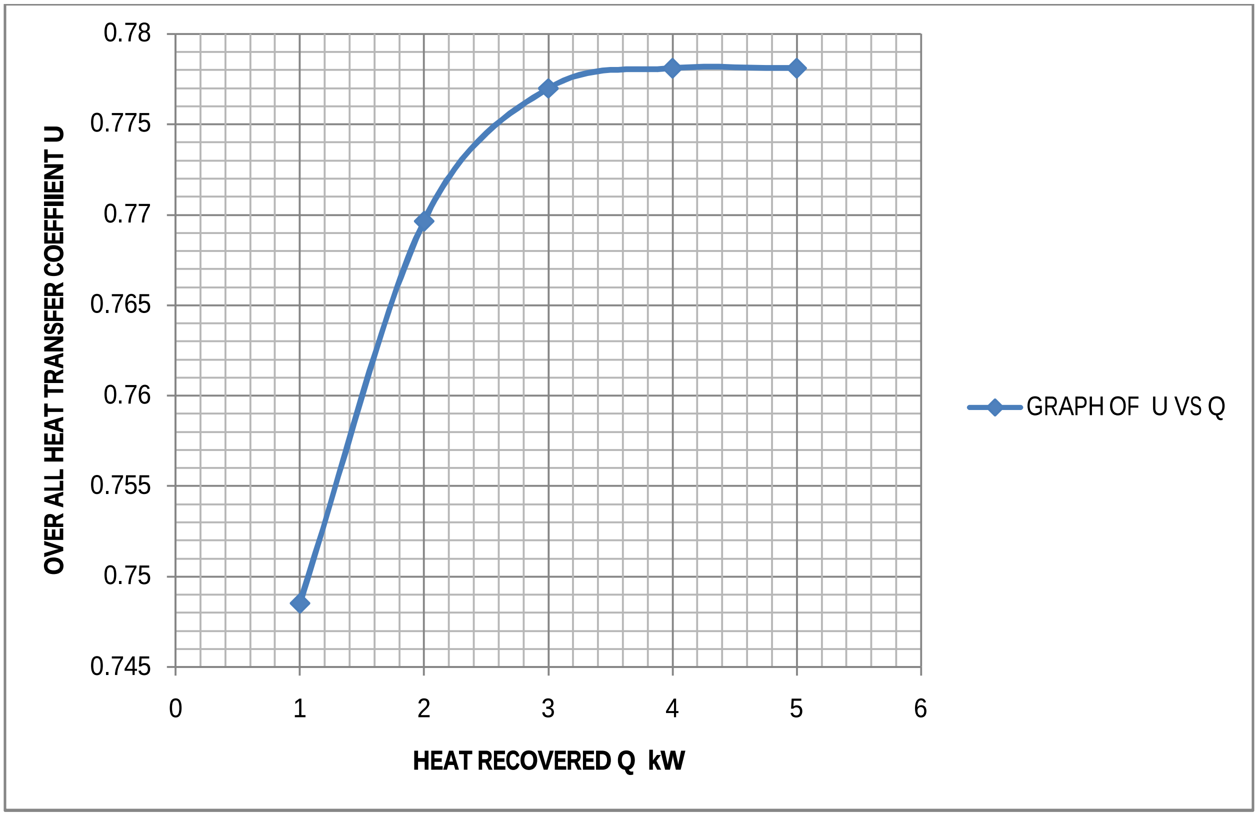
Transient test

Time (s)	$T_{a\ in}$	$T_{a\ out}$	Q kW	U kW/m^{2}K
0	22.7	22.7	0	0
2	22.7	200	8.4780	0.74852
4	22.7	205	8.7171	0.76964
6	22.7	206	8.7919	0.77698
8	22.7	207	8.8128	0.77809
10	22.7	207	8. 8128	0.77809

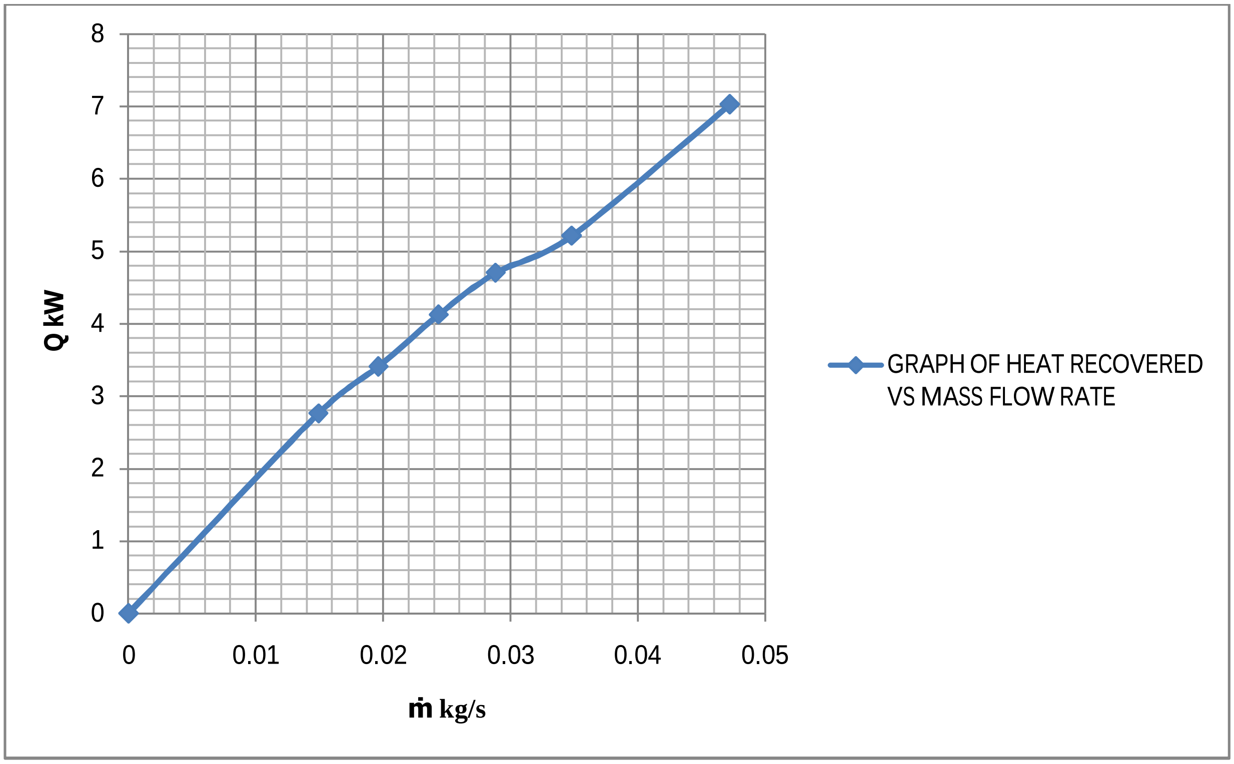
GARPHS.

GRAPH 5.1:

Graph of U vs Q.

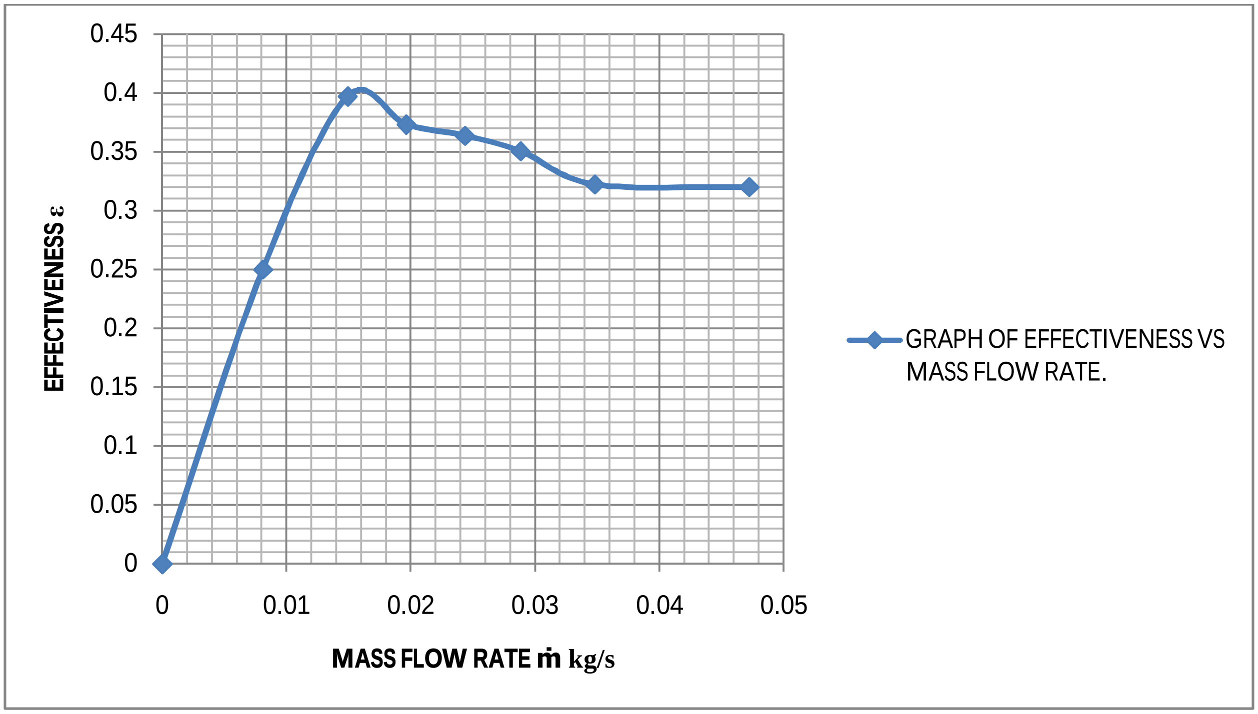


GRAPH 5.2:
Graph of Q vsṁ.



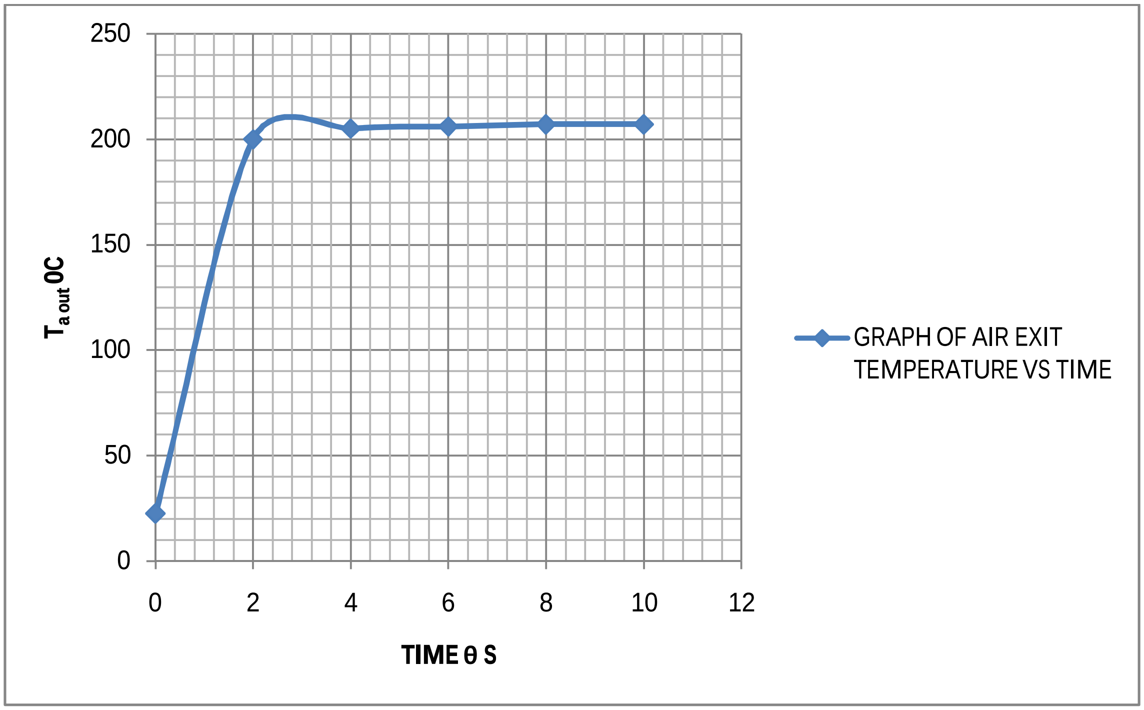
GRAPH 5.3:

Graph of ε vsṁ.



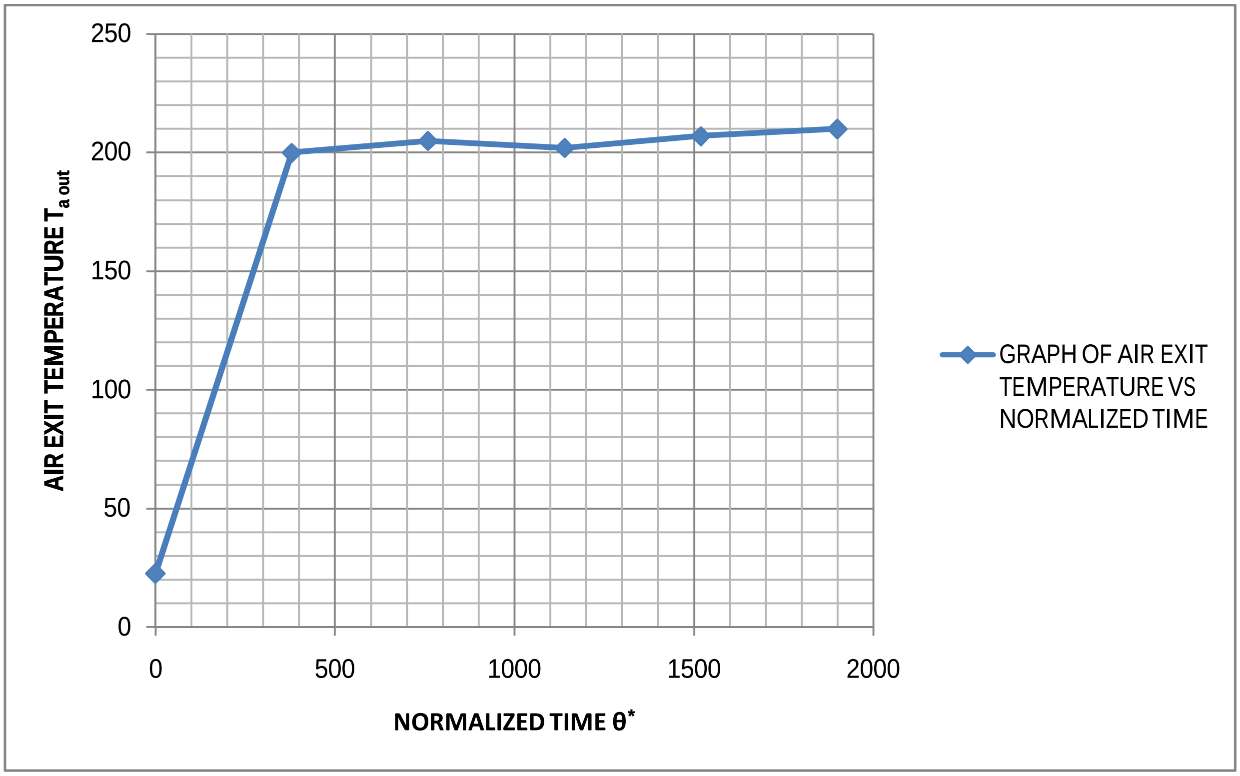
GRAPH 5.4:

Graph of $T_{a\ out}$ vs θ.



GRAPH 5.5:

Graph of $T_{a\ out}$ vs θ^*.



CHAPTER SIX

6.0. BILL OF QUANTITIES

TABLE 7.1

PRODUCTION COST OF THE EXCHANGER MODEL

ITEM	QUANTITY	UNIT PRICE (Ksh)	TOTAL COST (Ksh)
ALUMINIUM SHEET (32 GAUGE) m^2	11.08	600	6648
SILICON SEAL (TUBES)	9	295	2655
GALVANIZED IRON SHEET (30 GAUGE) m^2	1	550	550
MILD STEEL SHEET (16 GAUGE) m^2	2.5	700	1750
BINDING WIRE (28 GAUGE) G1 m	3	30	90
MILD STEEL ANGLE LINE m	2	300	600
BOLTS NAD NUTS	62	10	620
DIESEL (litres)	15	105	1575
HOLTS GUN GUM	1	300	300
GROSS COST			**14 788**
LABOUR			2 220
TOTA PRODUCTION COST			**17 008**

CHAPTER SEVEN

7.0 DISCUSSION

From graph 5.1 the amount of heat recovered Q increased linearly with the overall heal transfer coefficient from values of u and Q of 1.00 kW and 0.748 respectively. At a value of Q= 3.4kW, the value of U reached its maximum and then remained constant for higher values of Q . The optimum value of U was found to be 0.778 from Q=3.4kW and higher values of Q.

Q increased with increasein□.As given in graph 5.2. optimum values □ was never reached due to limited fan capacity.

For the mass flow rate of 0.02kg the corresponding effectiveness is 0.37 or 37% from graph 5.3. The maximum effectiveness possible was taken as 40%.

The temperatures for the above operating conditions were:

Air inlet temperature $T_{a\ in} = 22.7^0C$

Air outlet temperature $T_{air\ out} = 194^0C$

Flue gas inlet temperature $T_{gin} = 482^0C$

Flue gas outlet temperature $T_{gout} = 253^0C$

The optimum operating conditions can therefore be summarized as follows:

Overall heat transfer coefficient U= 0.778kW/m^{2}K

Effectiveness ε = 40%

$T_{air\ in} = 22.7^0C$, $T_{air\ out} = 194^0C$, T_g in = 482^0C, T_g out 253^0C

TRANSIENT TEST

The air exit temperature rose sharply with time within the first two minutes from 22.7^0C to about 200^0C. It then rose to a maximum of 210^0C in the next one minute then dropped to 205^0C at the end of four minutes. This temperature then remained constant.

The maximum time to allow temperatures to reach maximum was 4 minutes.

Normalizing this time with the dwell time the maximum normalized time was found to be 750.

7.1 CONCLUSION

The objective of this project was completion and testing of boiler chimney heat recovery heat exchanger system that could be used to recover heat lost through flue gases. A plate type heat exchanger was used in the design. The systems model was completed and tested under forced convection conditions. From the performance of the model the optimum operating conditions were obtained as:

Overall heat transfer coefficient $U = 0.778 kW/m^2K$

Amount of heat recovered $Q = 3.4$ kW

Effectiveness $\varepsilon = 40\%$

$T_{air\ in} = 22.7^0C$, $T_{air\ out} = 194^0C$, T_{g} in $= 482^0C$, T_{g} out 253^0C

7.2 RECOMMENDATIONS

After completion, testing and analysis of the performance of the exchanger the following recommendations were made:

1. The exchanger core plate spacing should be increased to improve on air flow. This will also increase time required before maintenance as it will slow the rate of blockage due to fouling on the gas side. The negative effect of this is that it will redue the number of plates and hence the amount of heat recovered but it is still a worth change for the lifetime of the plates.
2. An allowance for expansion of the plates should be provided at the ends of the plates. This is to eliminate the slaking that was observed during testing of the exchanger. Slaking of the plates increases resistance to flow.
3. The height, L_2, of the heat exchanger should be increased to increase the time for the flue gases to exchange heat with air.
4. The filters can be fabricated and installed and the model tested for fouling.

7.3 REFERENCES

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5. Anthony F. Mills, *Heat Transfer International,*(1992, Paperback, Student Edition of Text Book)
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7. Ongicha Hillary Buore, Ong'ondiMayaka Eugene, *Design Fabrication and Testing of Boiler Chimney Heat Recovery Unit roport, (*2013)

PICTORIAL REPRESENTATIONS.

Gas side inlet duct.

Exchanger core plates.

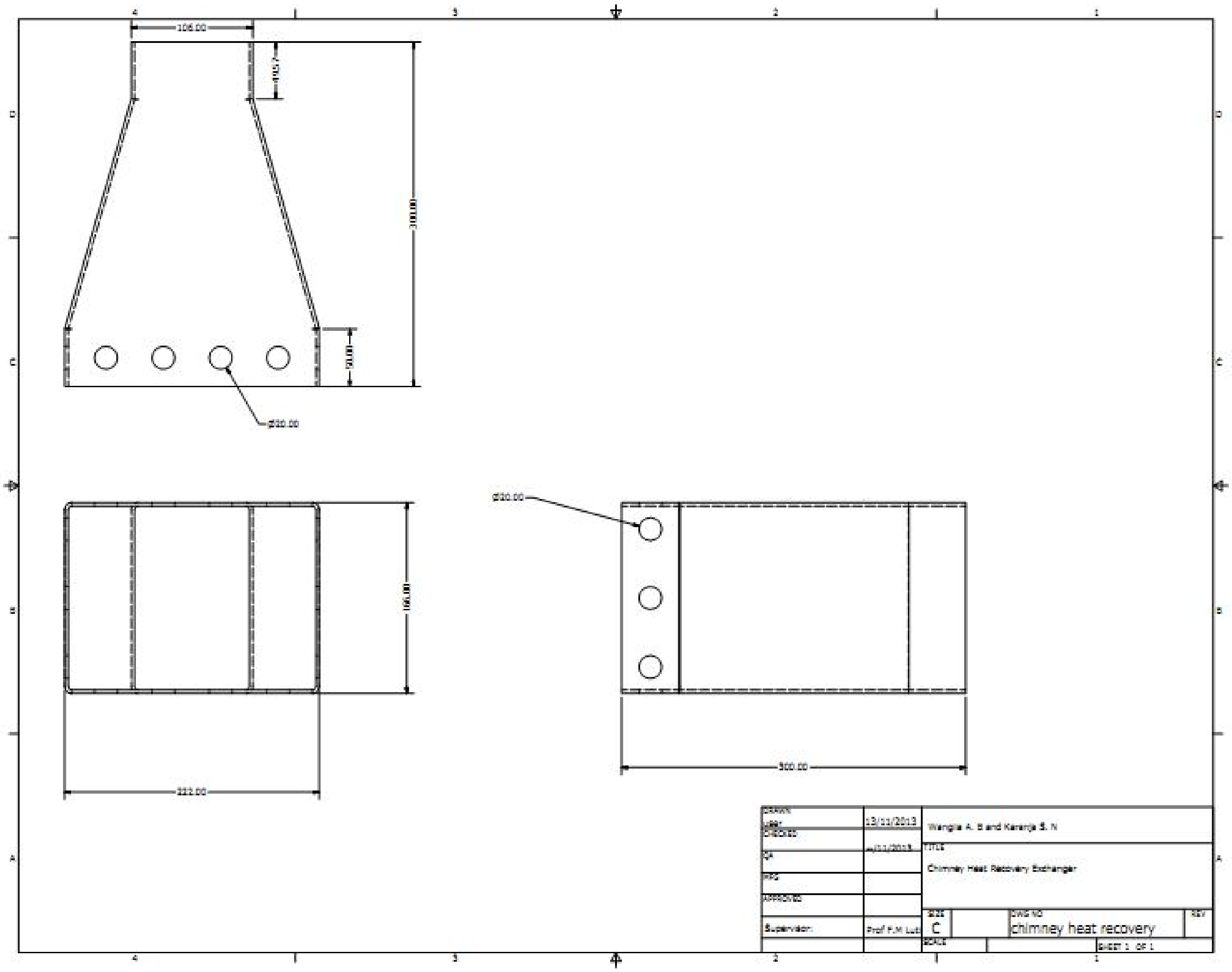
Gas side outlet duct

Exchanger assembly

Experimental set-up

Actual test

DRAWINGS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Supervisor:
Prof F.M Luti
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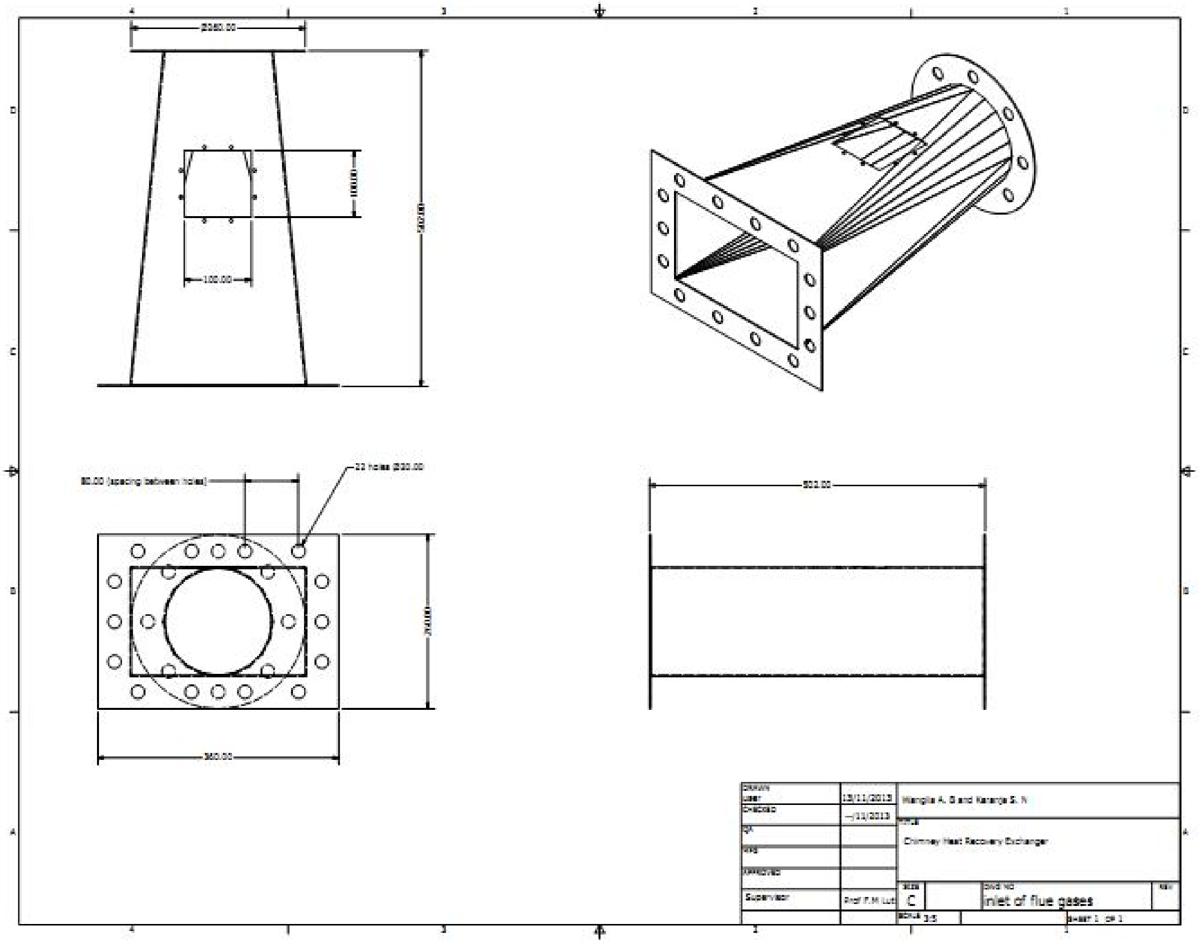

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Chimney Heat Recovery Exchanger
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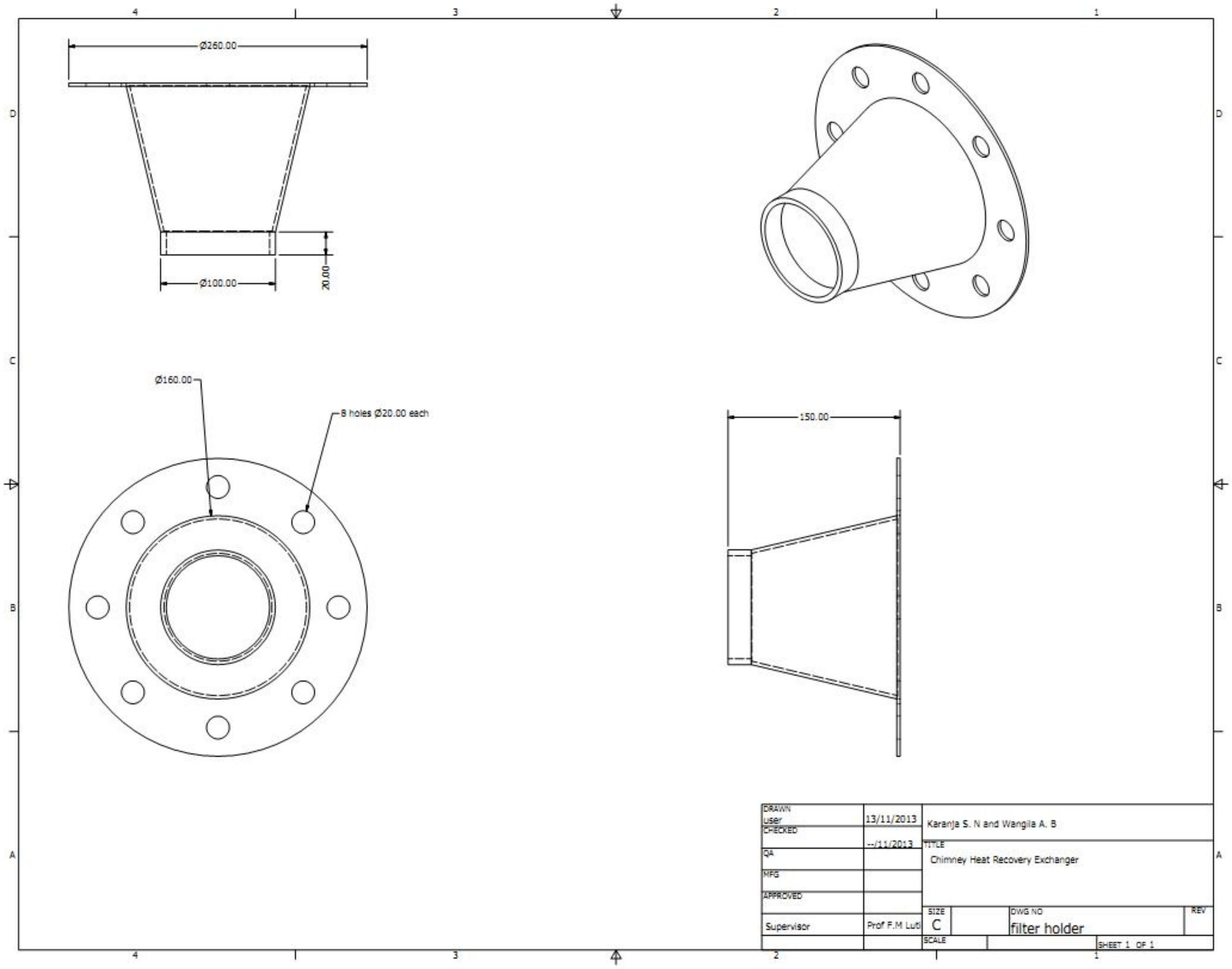
inlet of flue gases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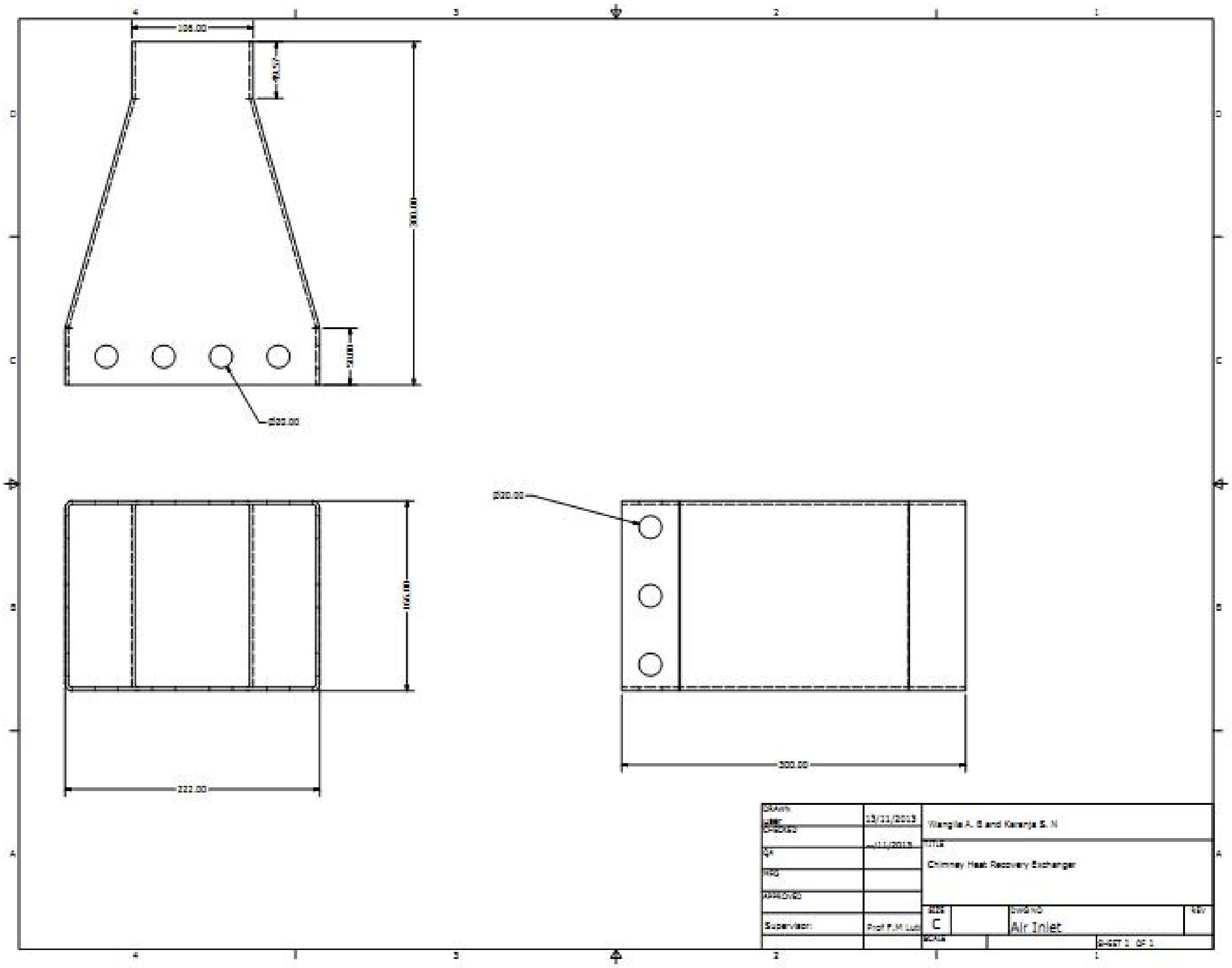

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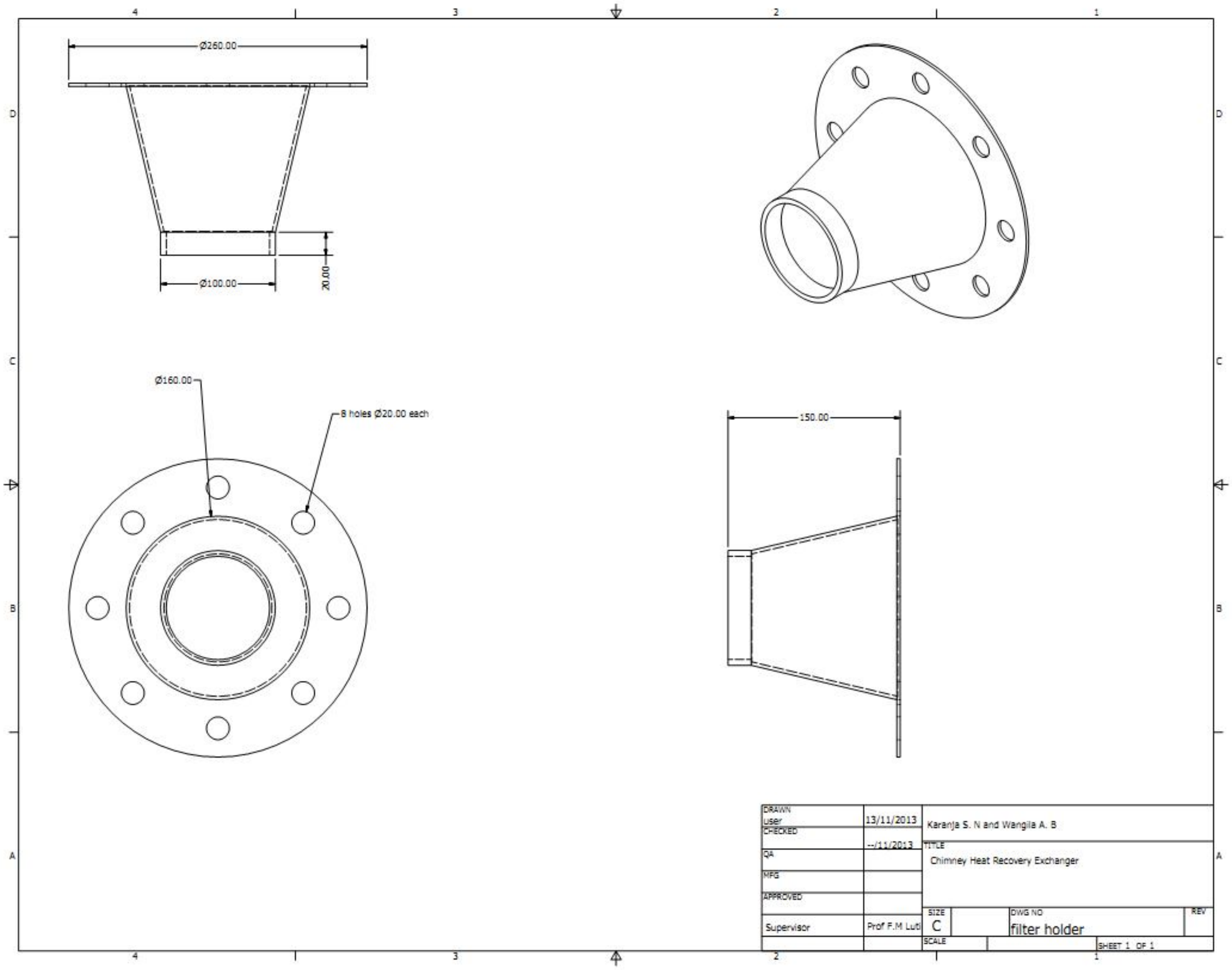
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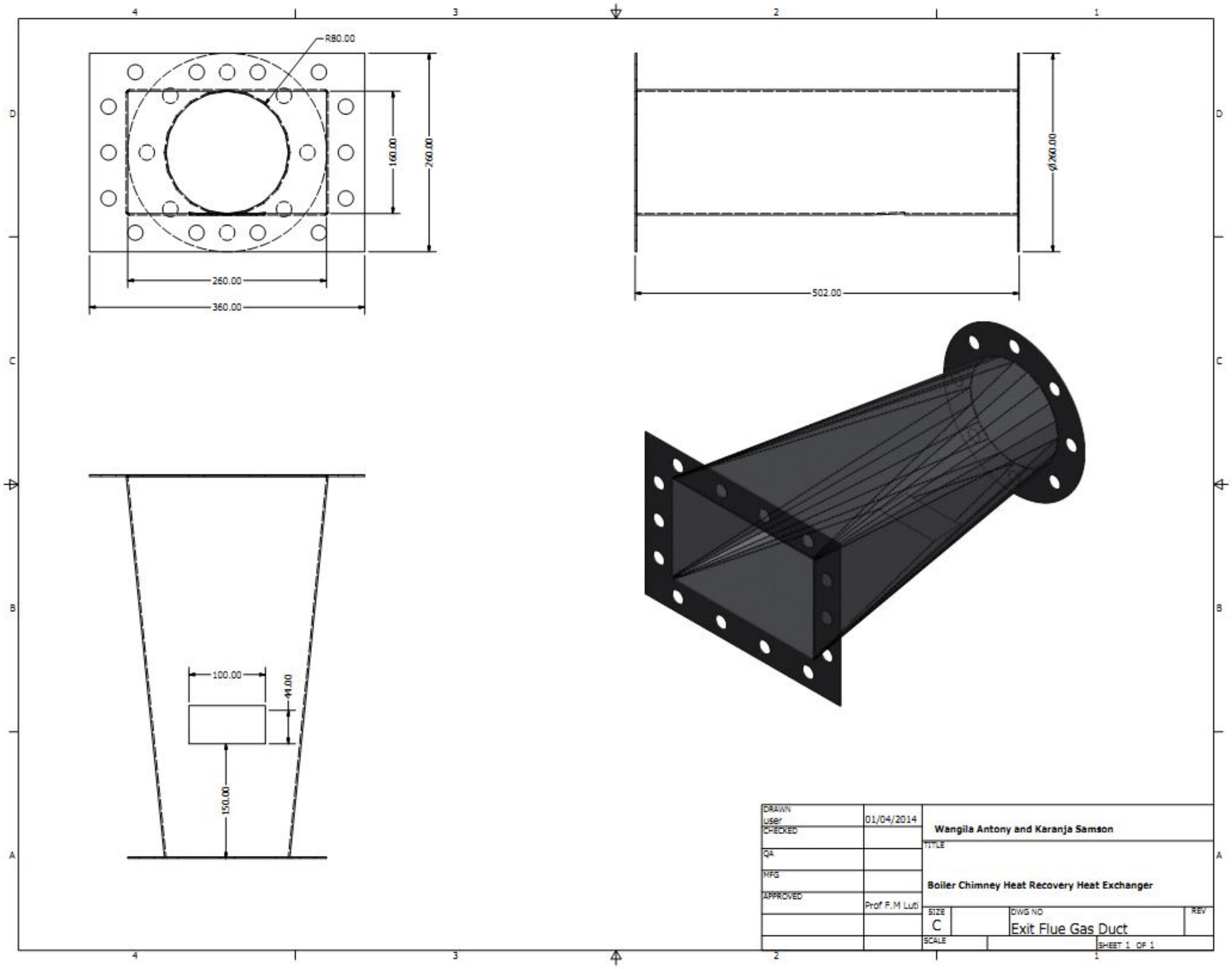
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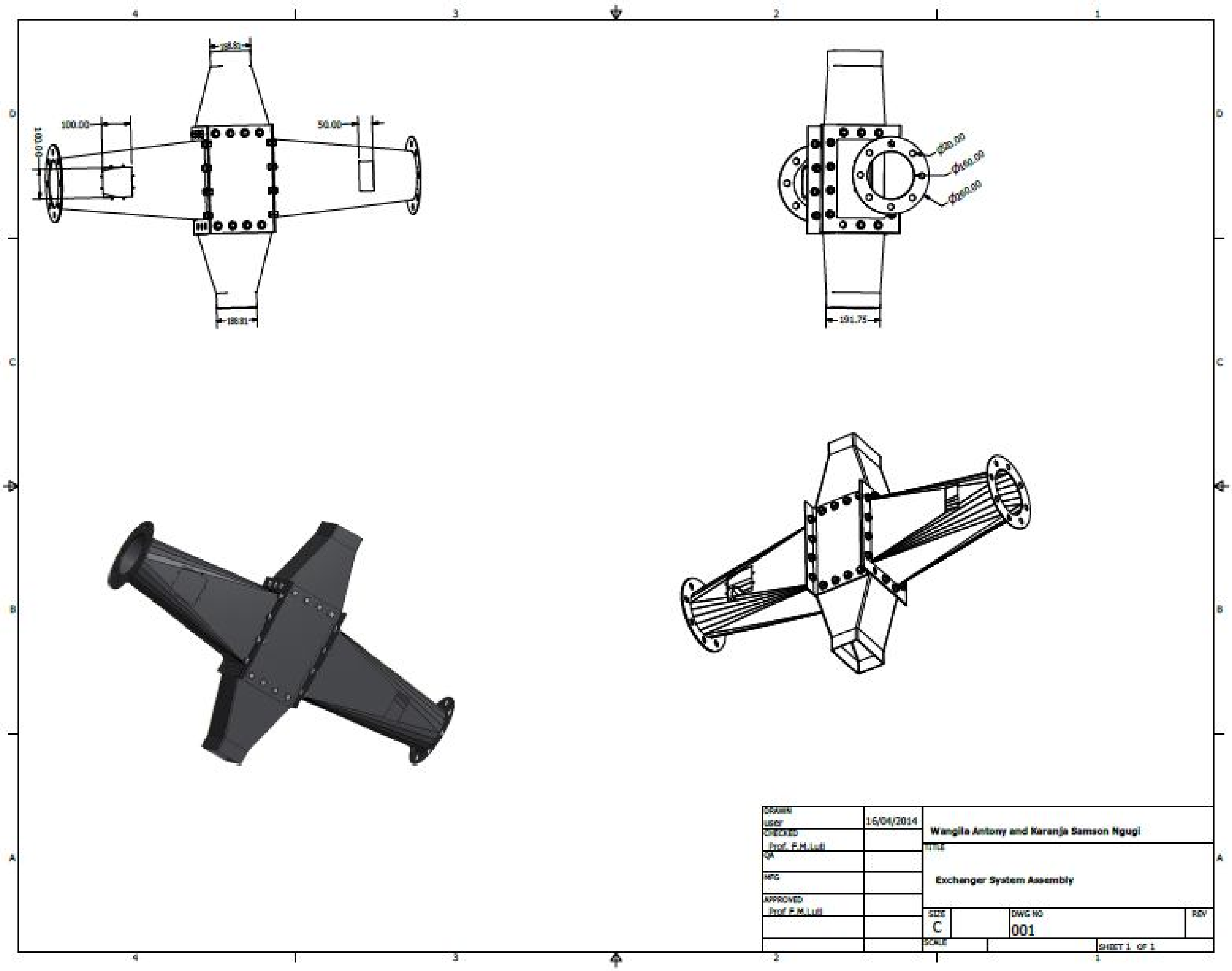
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