UNIVERSITY OF NAIROBI.

SCHOOL OF ENGINEERING

WASTE HEAT RECOVERY FROM BOILER STACK

BY

BILLY MWANGI KING'ORI

F56/12293/2018.

Report Submitted in Partial Fulfillment of the Requirements for the Degree of Master of Science in Energy Management of the University of Nairobi

Department of Mechanical and Manufacturing Engineering

JULY 2022.

DECLARATION.

I **Billy Mwangi King'ori** declare that this report is my original work, and except where acknowledgements and references are made to previous work, the work has not been submitted for examination in any other University.

Signature.....................

Date. 21.11.2022

APPROVAL BY SUPERVISORS.

I confirm that the study was carried out under my supervision and has been submitted for examination with my approval as University supervisor.

Dr. A. Aganda Signature.................. Date.............21/11/2022

Dr. R. Kimilu Signature.............. - Date.............November 21, 2022

DEDICATION.

I dedicate my research project to God, my family and many friends. I am grateful to my loving parents, David and Beatrice, who inculcated a culture of hard work in me for the things I aspire for in life. I sincerely thank my sisters Caroline and Monicah for their encouragement in the course of my project work.

I am grateful to Eric and Nelphat especially for helping me develop my industrial data collection skill. Much gratitude goes to my many friends who were my source of encouragement and support during my challenging times at graduate school.

Finally, I dedicate this research project to fellow researchers who made contributions to research on COVID-19 thus saving lives, you remained dedicated and worked hard during an extremely difficult period.

God bless you all.

ACKNOWLEDGEMENT.

A special thanks to my university supervisors Dr. A. Aganda and Dr. R. Kimilu who were generous with their expertise, immediate feedback and precious time during the research project.

I would also like to thank the University of Nairobi (UoN) mechanical engineering workshop technical team; Mr. Kahiro, Mr. Maina, Mr. Jacktone and Mr. Kibagendi. Their willingness to assist me for this research project made the completion of this research project an enjoyable experience.

I would also like to thank the management teams of different industries for the opportunity to conduct an industrial survey on their boilers.

ABSTRACT

Steam is generated in pressured vessels called boilers. The cost to industry to generate steam is substantial, typically up to 40% of total expenditure. In operating the boilers, efforts are made to have and keep the boiler's efficiency as high as possible and to recover any waste heat. Most of the heat loss from the boilers occurs through stack flue gases; accounting for up to 30% of the total energy input into the boiler.

For large boilers and those that use natural gas, economizers and air – preheaters are installed to recover waste heat from flue gases. However, these heat exchangers are not suited for small boilers whose capacity is less than 6,800 kg/h and also those that use Heavy Fuel Oil (HFO). Most small boilers do not recover waste heat from flue gases.

The small capacity boilers lose a significant amount of energy through the flue gases. To quantify these loses, as part one of this project, a short survey of ten stacks was conducted. The survey found that all the ten stacks had no heat recovery systems. Five were insulated of which in only two was the lagging effective. The other five had no insulation at all. The stacks with well-maintained lagging had the lowest temperature drop (inlet 223°C outlet 210°C). The flue gases retained most of the heat energy and exited the stacks still at high temperature. The flue gas in other un-insulated stacks and with poor lagging most of the energy through the stack surface and exited on average, at 67°C. To recover some of the waste heat the stack can be modified to act as a heat exchanger to recover waste heat from the same flue gases.

In the modification of the stack, the double pipe coil heat exchanger design was adapted. The model stack was a 101.6mm (4 Inches) diameter pipe and 1280mm height. On the external surface of the pipe was brazed a helical 9.53mm copper tube with a 101.6mm pitch. The assembly was then insulated. The flue gases were conveyed in the normal way but lost heat to the water flowing in the copper tube. Tests on the stack were performed representing three scenarios. In the first scenario, hot gases generated by an oil burner were passed through a bare stack pipe. The temperature of

the hot gases at entry to the stack was on average 385°C and exited at 122°C. The second scenario involved passing the hot gases through the stack with the copper tube brazed on the stack outer surface and then covered with a 35.3mm fiberglass insulation. This represented an insulated stack but with no heat recovery. In this scenario, the exit temperature was on average 185°C. The third scenario was similar to the second except that water was passed through the helical tube. This represented the modified stack with heat recovery. The exit gas temperature averaged 136°C. As the water passed through the helical tube, its temperature increased. For example, at a water flow of 0.0114 kg/s, the temperature increased from 22.7 to 48.7°C. The energy absorbed by water was found to be about 30.6% of the recoverable energy in the flue gases.

This project has therefore shown that a significant amount of waste heat from small capacity boilers can be recovered with some limited modifications on the boiler stacks.

Table of Contents

ABBREVIATIONS.

HFO	Heavy Fuel Oil.
PCD	Pitch Circle Diameter.
LMTD	Logarithmic Mean Temperature Difference.
CFD	Computational Fluid Dynamics

NOTATIONS

c_{pfg}**-**	specific heat capacity of flowing flue gas at constant pressure, in kJ/kgK.
m_{fg}**-**	mass flow rate of hot flue gases along the chimney height in kg/s.
c_{pw}**-**	specific heat capacity of flowing water at constant pressure, in kJ/kgK (4.187kJ/KgK) [22].
m_w**-**	mass flow rates of water in the heat exchanger in kg/s.
A-	stack cross sectional area.
V_{fg}**-**	flue gas velocity in m/s.
ρ_{fg}**-**	flue gas density in kg/m3.
c_{fg} **-**	flue gas specific heat capacity in kj/kgK.

LIST OF FIGURES.

LIST OF TABLES.

CHAPTER ONE

INTRODUCTION

1.0 BACKGROUND.

Steam is a popular working fluid in industry as a media for energy transfer in process heating and power generation. This is because liquid water from which steam is generated is cheap, inert and has high specific heat and latent heat. Further, the steam can easily be controlled and distributed.

Steam is generated in a boiler - an enclosed high pressure in which water is converted to steam. This involves energy conversion of the chemical energy in fossil fuels to thermal energy through a combustion process. Generation of steam in boilers consume a lot of energy and accounts for about 30-40% of the total industrial energy and approximately 28% of all energy at commercial facilities [1]. With the growing trend of rising fossil fuel cost and the effect on the climate, there is emphasis on reducing the use of fossil fuel and having boiler efficiency as high as possible.

The thermal efficiency of boilers is a function of the amount of heat loss. Heat loss from a boiler occurs through stack losses, un-burnt carbon, blow down and convection and radiation losses from its surfaces.The highest loss of energy in a boiler is through the flue gases (also called stack loss). It amounts to about 10-30% of the energy supplied [2]. It's therefore important that means are developed to minimize the energy loss through the boiler stack. This can be achieved by for example employing waste heat recovery procedure to the flue gases. Typically waste heat recovery from the flue gases has been done by installing economisers and air preheaters.

The economizer and air preheater are heat exchangers installed within the boiler stack to recover energy from the flue gases and preheat boiler feedwater (economizer) and combustion air (air preaheater). The consequence of preheating boiler feed-water and combustion air is reduction of boiler energy demand and improved boiler efficiency. The economizer is a highly pressurized heat exchanger through which feed water is heated before entry to the boiler. However, the high economizer pressure requirements make it expensive in material and construction and require expertise in the manufacture and installation. The effect of the economizer on the increase of boiler efficiency is such that for every 5℃ increase in feed water temperature leads to a 1% increase of the boiler efficiency [3]. Similarly, there is a 1% increase of boiler efficiency for every 4.5℃ increase in combustion air temperature after the air preheater [4]. However, economisers and air preheaters are not suited for all boilers.

There are two main considerations for the economic installation of economisers and air preheaters in boilers. One is the boiler capacity which must be greater than 15000 Lbs/hr (6,800 kg/hr) [5] and the other is the high cost of installing the economisers. For small capacity boilers it may not be economically viable to install economisers and air preheaters. Further, boilers that use HFO (Heavy Fuel Oil) have a problem of excessive soot fouling on the economizer and air preheater heat transfer surfaces. The effect of soot on heat transfer is demonstrated by the fact that a soot layer 0.8mm thick can reduce the heat transfer by 9.5% while a 4.5mm layer may reduce heat transfer by 69% [14]. Many of the small capacity boilers (mainly used in process heating) do not have waste heat recovery systems and the fuel is usually HFO. However, these boilers lose significant amount of energy through hot flue gases. The exit temperatures are typically between 150-250℃ representing a high energy loss.

The aim of this project was to modify the boiler stack of this small capacity boilers for the purpose of recovering waste heat as it flows through the stack (before it is discharged). In this modification the waste heat recovery system is suggested such that the stack will act as a heat exchanger between the flue gases and water. As the flue gases are conveyed through the stack, waste heat energy is recovered. In this modification there will be no high pressure (water) and no soot fouling. The problem of soot fouling on the surfaces is solved by avoiding direct contact of the heat transfer surfaces with flue gases. The waste heat recovery system is anticipated that it can be installed on any stacks regardless of the boiler size.

This project was conducted in two parts. The first part involved quantifying the heat loss through a boiler stack survey and to determine the status of waste heat recovery. The second part involved the fabrication and testing of a small scale laboratory model. The model stack was designed such that it conveyed the flue gases and simultaneously extracted heat from the gases. The modification avoided both the high pressures experienced in the economizer and the soot fouling. This was by avoiding direct contact of the heat exchanger surfaces and flue gases and introducing water (heat transfer media) in the heat exchanger at normal pressures.

1.2 PROBLEM STATEMENT.

Flue gases exit steam boilers at high temperatures and therefore contain a lot of energy. This energy leaves the boiler and is dumped to the environment. Approximately 10-30% of boiler energy input is lost through the stack (flue gases) accounting for the largest heat loss in boilers. There are heat exchangers which are designed and installed to recover normally some of the waste heat from the flue gases. Economisers are used to recover heat from the flue gases and preheat the boiler feed water while the air preheaters are employed to preheat combustion air. However these waste heat recovery systems don't apply to all stacks. They are not suited for low temperature (150°C - 250°C) flue gases and for small capacity (< 6800 kg/h) boilers. The economizer and air preheater are expensive facilities and their application in small boilers is not economically viable. In boilers that use HFO, the heat exchangers experience poor heat recovery rates due to soot fouling. Hence, there are many boilers especially those used in process heating where there is no heat recovery from the flue gases. All the energy in the flue gases that enter the stack is dumped to the environment. There seem to be no established methods that can be employed to recover heat from flue gases in these types of boiler stacks. Hence, such boiler stacks will continually loose large amounts of energy. In this study, a modification of the stack is suggested such that as the flue gases are conveyed out of the boiler, there is simultaneous recovery of heat from the gases. The modification would also avoid high pressures experienced by the economisers and the heavy soot fouling that occur in some boilers that use HFO.

1.3 OBJECTIVES.

1.3.1 Overall objective:

The main objective of this project was to modify the boiler stack design in order to facilitate waste heat recovery from the flue gases.

1.3.2 Specific objectives.

The specific objectives were to:

(i) Conduct a short survey in ten selected boiler stacks in order to establish the energy in the flue gases and the current status in view of waste heat recovery. In the survey, the following were recorded: the stack dimensions, flue gas temperature in and out of the stack, flue gas flow rates, and lagging condition. Also estimates of heat loss due to the flue gases were made.

(ii) Design, construct and test a laboratory stack model in which the suggested modification were accommodated.

(iii) Estimate the waste heat recovery from the modified stack.

CHAPTER TWO

LITERATURE REVIEW

2.0 INTRODUCTION.

The major loss of waste heat in boilers takes place through the flue gases leaving the boiler at high temperatures. Waste heat recovery from flue gases has been achieved using economizer and air preheater. However, these cannot be used by all boilers especially the small sized boilers and those that use heavy fuel oil (HFO). In this project a modification of the stack is proposed to incorporate a waste heat recovery system that can be applied to any boiler. In this chapter relevant literature concerning waste heat recovery in boiler stacks are reviewed.

2.1 WASTE HEAT RECOVERY IN BOILER STACKS.

Boiler heat loss occurs by a variety of ways. There are three major ways by which the boiler losses heat energy. These are by flue gas losses, traditional surface losses by radiation and convection, and blow down losses [6]. Flue gas constitutes the largest heat loss in boilers, estimated at 30-35 % of the boiler fuel input [6, 2]. By far flue gases loss is the highest heat loss and it is more than all the other losses combined.

The flue gases leave the stack at high temperatures, typically 150-250℃ for small boilers and 232.5-343.5℃ for large boilers [7, 8]. The flue gas temperature without any other consideration is based on the steam temperature at the generating pressure. The flue gas temperature cannot be less than that of steam. The main reason for much higher exit flue gas temperatures than that of steam is due to a number of factors. They include type of fuel, boiler design, heat transfer surface condition, fouling of tube surfaces, excess air, combustion equipment and boiler load [6, 9]. The effect of soot on heat transfer is demonstrated by the fact that a soot layer 0.8mm thick can reduce the heat transfer by 9.5% while a 4.5 mm layer may reduce heat transfer by 69% [14]. However, in this project, in the proposed waste heat recovery system there will be no contact between the flue gases and water tubes. Therefore, fouling is no longer a factor.

Without heat recovery from the gases, all the energy in the flue gases at boiler exit is wasted; whether the stack is insulated or not. For large boilers and those that use natural gas, economisers and air-heaters are installed and waste heat recovery from flue gases is effective. With both heat exchangers, it is possible to recover up to 60 % of the stack energy losses [5].

2.1.1 AIR PREHEATERS.

Air preheaters are heat exchangers that recover heat from the flue gases and transfer the same to the combustion air. They are normally installed after the economizer in the direction of flue gas flow. There are different designs such as regenerative air preheaters, tubular air preheater and others. A typical tubular air preheater is shown in Appendix A [10]. In the use of air preheater, the boiler efficiency improves by 1% for every 4.5°C increase in combustion air and similarly, for every reduction of hot flue gases by 22°C the boiler efficiency improves by 1%, [4, 5]

The air pre-heaters surfaces suffer the problem of condensation especially when the flue gases temperature is low. And also the air preheater effectiveness is affected by fouling from soot where ever HFO is the fuel.

`2.1.2 ECONOMISER.

Economisers are finned heat exchangers used to recover waste heat from flue gases and used preheat the boiler feed-water. This has the effect of improving the boiler efficiency. Appendix B shows a typical finned type boiler economizer installed on the stack. The flue gas is cooled from 260℃ (500℉) to 148.9℃ (300℉) while feed water temperature increases from 104.4℃ (220℉) to 132℃ (277℉) [11].

A number of studies [3, 4, 5, 11, 12, and 13] have been done concerning boiler efficiency improvement through installation of the economizer. According to [3], a 1% improvement of boiler efficiency is realized for every 5°C increase in feedwater temperature. In [4], the 1% improvement was achieved with every 10°C increase in feed water temperature. The same (1%) improvement was found with every 25°C reduction in flue gas exit temperature using an economizer [13]. The boiler efficiency improvement due to the installation of conventional economisers also depends on type of fuel and exit gas temperatures. For a particular gas exit temperature the boiler efficiency improvement is [Appendix C, 5] highest for natural gas followed by fuel oil and least with coal. In general, efficiency improvement for a steam boiler installed with an economizer is 2 – 4% [12].

Economisers are expensive equipment that require specialized material for construction. The feed-water passing through the economizer is highly pressurized to near boiler pressure [8]. Therefore, use of economizers is not for all sizes of boilers and fuels. For small boilers, installation of economisers is not economically viable. The savings accrued will not be sufficient to have a reasonable payback period.

In boilers that use HFO the economizer suffer from heavy fouling due to soot. This can limit the heat transfer between the flue gases and feed water. It should be noted that fouling will not be a factor in the proposed waste heat recovery system as there will be no direct contact between flue gases and the water tubes.

For beneficial use of economisers it is recommended that they be installed where the average stack gas temperature exceeds 232℃, the annual boiler operating time is longer than 2,500 hours, and the stack gas flow rate is greater than 6, 800 kg/h [5].

Many boilers used in process heating are of small size (compact) and nearly all use HFO. Most are therefore not fitted with any waste heat recovery system. The heat energy contained in the flue gas is discharged to the environment. In this project the coil heat exchanger design is adapted in modifying the stack to not only convey the flue gases out of the boiler but also act as a heat exchanger. The stack will allow heat transfer from the flue gases to water at relatively lower temperature and pressure. The suggested modification also avoids problems caused by soot fouling of heat transfer surfaces.

2.2 COIL HEAT EXCHANGERS.

A coil heat exchanger is a device that transfers heat or thermal energy from one fluid to another with the help of coils. A coil heat exchanger consists of a duct whether rectangular or circular on which the tube coil is attached. The coil heat exchanger relies on the coiling tube to transfer heat between two fluids. There are many types of coil heat exchangers. Some of these include stainless steel tube bundles, stainless steel tube immersion coil, copper coil heat exchangers, coil tube–in–shell heat exchanger and others. The coil tube separates the two fluids. One fluid flows inside the tube and another on the outside (ie through the duct). Typically the hot fluid flows through the duct and the cold fluid flows through the tube coil. The heat is transferred from the inner fluid to the wall via convection, it then conducts through a pipe wall to the other side and the outer fluid carries the heat away also through convection.

Shown in figure 2.1 is a coil tube–in–shell heat exchanger where the helical tube is wound on a core. The core only supports the tube coil.

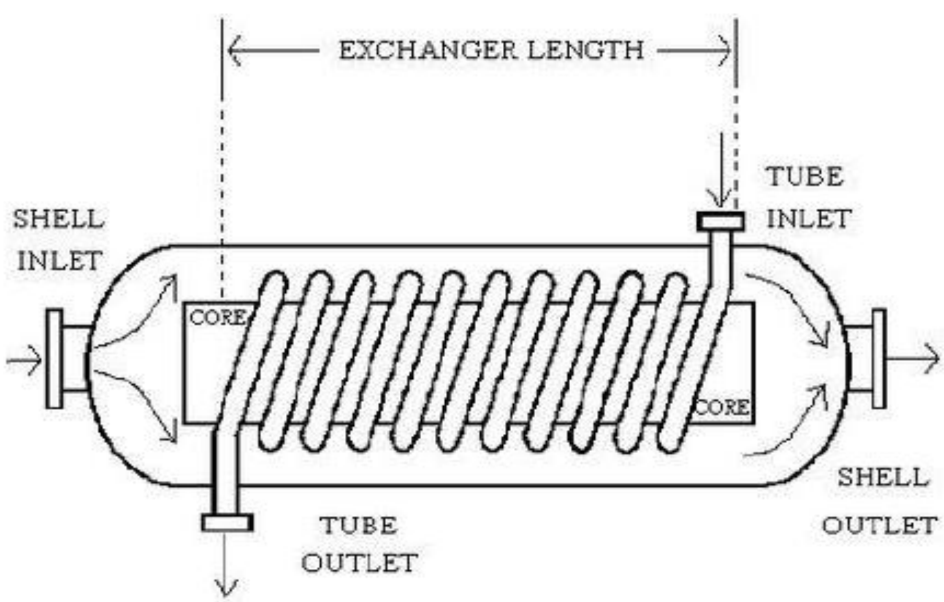


Figure 2.1: Coil heat exchanger wound on a cylindrical shell.

Coil heat exchangers tend to have higher efficiency than other types because of the large number of closely aligned tubes. The design enlarges the heat transfer area which results in a higher overall heat transfer co-efficient.

The coil parameters which influence the coil heat exchanger performance are shown in figure 2.2. The important coil parameters are; the pitch, H, coil tube diameter, 2r, the pitch circle diameter (PCD), $2R_C$ and the curvature ratio r/R_C.

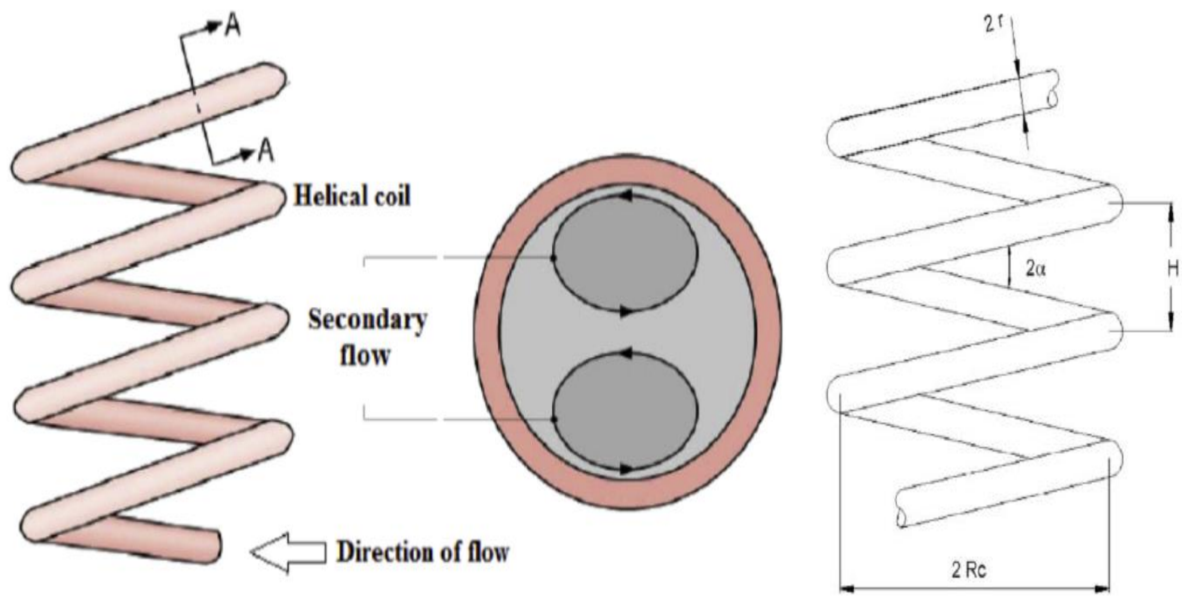


Figure 2.2: Cross sectional representation of the coil heat exchanger.

Some studies [15, 16, 17] have found that the coil pitch has little effect on heat exchanger performance but must be larger than the PCD. Also, the smaller the tube diameter the higher the rate of heat transfer coefficient. Smaller diameter results in more turbulent flow. A decrease in PCD (increase in curvature ratio) leads to an increase in heat transfer coefficient.

Another performance indicator of coil heat exchanger is the ratio of the PCD to coil tube diameter. The Logarithmic Mean Temperature Difference, (LMTD) is highest when the cylindrical pipe to the tube diameter pipe is approximately 10 [17]. This means that if the PCD is known then the small tube diameter can be determined for high heat transfer coefficient.

The high heat transfer rate in coil exchanger is attributed to a complex flow pattern that exists inside a helical pipe. The fluid in the tube experiences high turbulence and hence 'proper mixing' along the coil heat exchanger length [18]. This type of flow is highly responsible for the high heat transfer rate compared to straight heat exchangers. In another study [19], coil heat exchanger offered higher heat transfer rates of up to 10-20 times that of a straight heat exchanger. However, coil heat exchangers experience high pressure drop (2 – 3 times) compared to straight heat exchangers.

Figure 2.3 shows the double pipe coil heat exchanger. The small tube is wound on the larger pipe. The hot water flows through the larger aluminium pipe while the cold water flowed in the smaller copper tube. Experiments on the heat exchanger showed an increase in heat transfer with increased of either mass flow or temperature of the hot water [20].

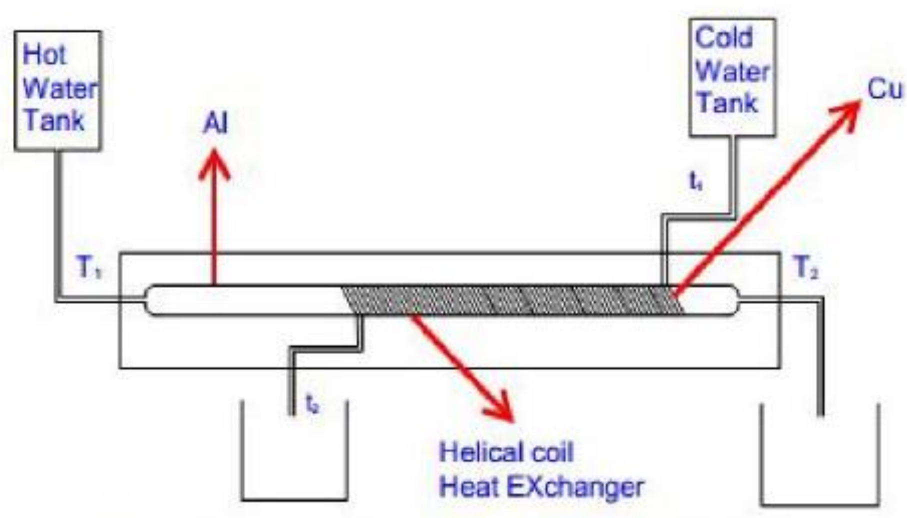


Figure 2.3: Experimental coil heat exchanger [20].

Shown in figure 2.3 is a coil heat exchanger in which heat transfer takes place between the inner bigger tube and the smaller wound coil. The two fluids do not mix and there are no high pressure requirements. This design was adopted in this study in the redesign of the model stack. The bigger tube represents the stack through which flue gases passes as usual and water is passed through the smaller (copper) tube. The heat transfer was from the flue gases (high temperature) to the water (low temperature) in the small tube. To minimize heat loss from the surfaces the arrangement was insulated. The heat in the flue gases would be transferred to the water in the small tube by convection between flue gas with stack surface, conduction through the stack material, conduction through the copper of the small tube and finally to the water by convection. This arrangement was referred to as the modified stack in this study. The details of the design are described in section 3.2.

CHAPTER THREE

METHODOLOGY

3.0 INTRODUCTION.

The aim of this project was to modify boiler stack so as to incorporate a waste heat recovery system. To achieve this objective, the project was carried out in two parts. The first part involved a short survey of ten selected boiler stacks. The purpose of the survey was to estimate the energy in flue gases entering and leaving the stack and establish the status of waste heat recovery. The second part was to design, fabricate and conduct tests on the model laboratory stack that had been modified in the form of double pipe coil heat exchanger. This chapter presents the methods used during the stack survey and in design, construction and testing of the modified stack.

3.1 BOILER STACK SURVEY.

A short survey of ten boiler stacks was conducted in order to determine the energy in the flue gases at boiler exit or stack entry. Additionally, the survey sought the condition of stack lagging and the existence of any waste heat recovery from the flue gases. All the ten stacks were visited and the required data taken.

The ten boiler stacks were randomly selected with close proximity to the engineering workshop, University of Nairobi. The ten boiler stacks were also chosen on the basis of variety in the industry in which they are installed. These industries include; hotel, a referral hospital, an international research facility and a dairy industry. Also the boilers varied in capacity from 1360kg/h to 6800kg/h. During the survey, the chosen boiler stacks were visited. The first visit was to the referral hospital which had two boilers in here known as boiler 1 and 2. The second visit was to the hotel which had three boilers in here known as boiler 3, 4 and 5. The third visit was the dairy industry that had four boilers in here known as boiler 6, 7, 8 and 9. The fourth visit was to the international research facility in here called boiler 10.

Figure 3.1 represents a typical boiler and stack assembly. In all the surveyed stacks, at exit from the boiler and entry into the stack, was a short duct of varying diameter and length - labelled A. At this section, provisions had already been made for flue gas temperature and velocity measurements. Similarly at a height B at what was considered the stack exit (that varied from stack to stack), provisions had been made for similar measurements.

The following data was recorded for each stack:

1. Inlet temperature of the flue gas into the boiler stack. A thermocouple was inserted into a port at A, sensed flue gas temperature which was then displayed and recorded using Lutron YK 2005-TM multi-function thermometer. The gas temperature was recorded at 1 hour interval for a total of 7 to 9 hours.
2. Outlet temperature of the flue gas from the boiler stack. Similarly at port B, a vertical distance that varied from 4.6m to 20m, the gas temperature was sensed by a thermocouple and recorded by the multi-function thermometer. The time intervals and total hours were same as those for inlet.

   As mentioned before, the surveyed stacks had varying heights and diameters. The diameters varied from 0.36m to 0.82 m. The heights varied depending on the surrounding such that in tall structures the stack height is highest. There was no standard stack height. In such a case, the heat transfer per meter height was considered as the appropriate variable that could relate the model stack heat transfer to the actual stack surveyed. Secondly, the parameter important for the relationship between the stack and the model stack is the surface area. For the same per meter height, the heat transfer depends on the surface area and the temperature difference. For the same temperature difference, the heat transfer relation between the model and the actual stack would be in the ratio of the diameters.
3. Volume flow rates of flue gas. At section A, a velocity measurement probe from the KANE 905 gas analyser was inserted to measure flue gas velocity. Using the velocities and the duct size, the flue gas mass rates were estimated. Again, the values were recorded at 1 hour interval for up to 9 hours; and then averaged.

4. The condition of insulation on the stacks. In boiler systems, stacks are normally insulated. The insulation prevents rapid cooling of the flue gas in the stack to avoid condensation. Condensation of water and sulphur dioxide corrodes the stack metal and other downstream equipment surfaces. The insulation therefore keeps the gas temperature above dew point of both water and sulphur dioxide. The interest of lagging in this project is that a heat sink would be created between the stack outer metal surface and the insulation material. The lagging of the stack with the heat sink would limit the heat loss to the environment but increase the heat transfer to the sink thereby enhancing the waste heat recovery.

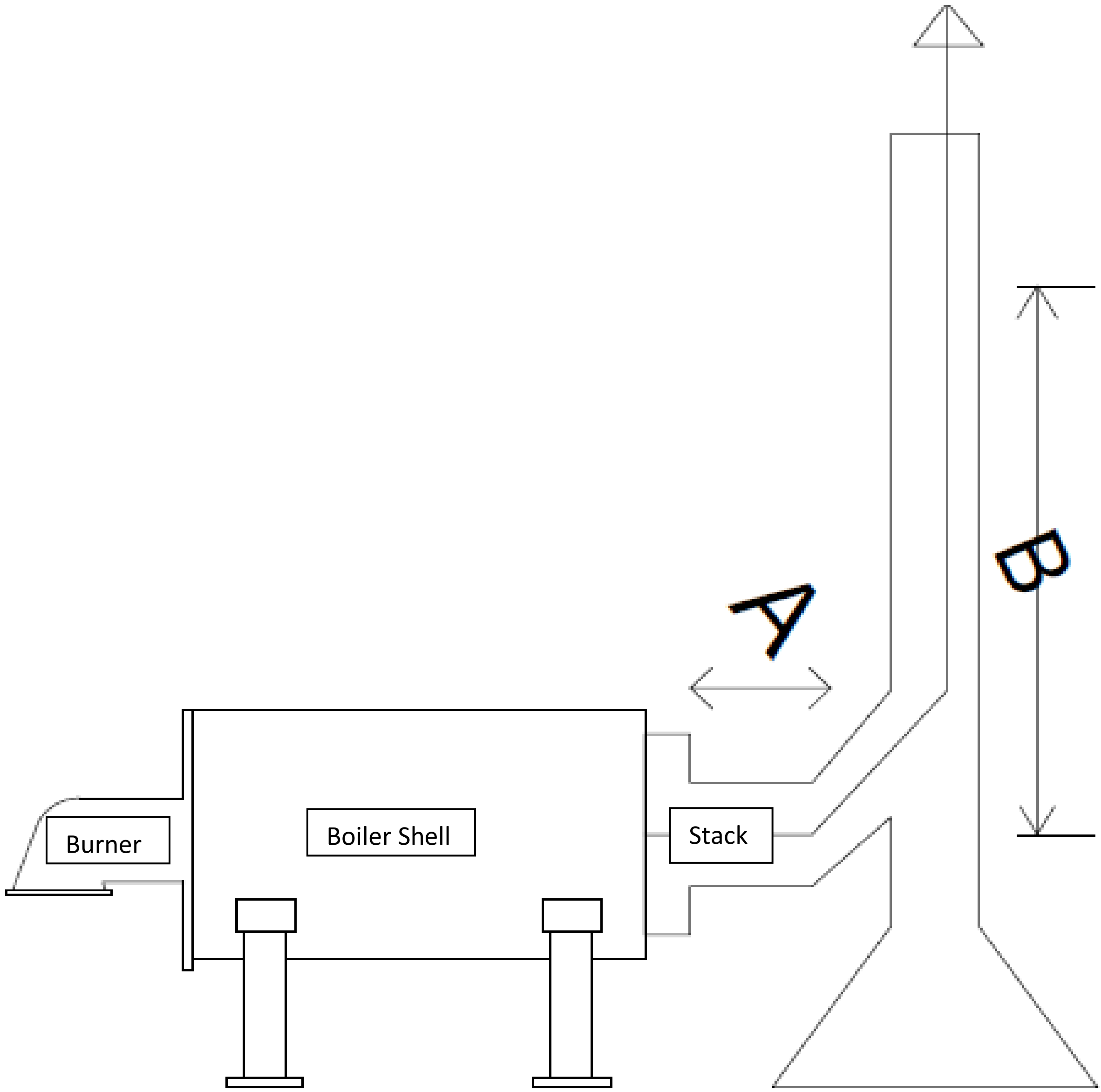


Figure 3.1: Typical illustration of a boiler and stack installation.

3.2 DESIGN AND FABRICATION OF STACK MODEL AND TEST RIG.

In the second part of this study a test rig and modified stack were designed and fabricated, here called the laboratory model. The model stack was fabricated so that detailed data on stack is obtained and also allow the variation of stack conditions. The model stack also enabled the determination of the performance of the proposed waste heat recovery system. On the model, tests concerning the heat recovery effectiveness of the modified stack were preferred on the laboratory model. The rig consisted of a flue gas generator, a conveying duct, and a jiko (a volume space) on which a test stack was fixed. See figure 3.2 (a) and figure 3.2 (b). The flue gas generator was an oil burner and a fan that supplied the gases to the jiko. The jiko was a space that damped out any gas fluctuations before entry to the stack. The laboratory model sits on the jiko which was connected to a burner by a conveying duct. The conveying duct conveyed the flue gases generated from the burner to the jiko.

A 4 inch (101.6mm diameter), 3mm thick black pipe was cut in two equal pieces 1280mm length. One piece was fixed on the jiko with no changes other than aluminium paint on the pipe and *j–type* thermocouple attachments (see figure 3.2 (a)). The *j–type* thermocouples sensed flue gas temperature at the stack inlet (T_1), at the centre height (T_2) and at the exit (T_3). Also measured were surface temperature at entry (T_{10}) and surface temperature at exit (T_9). The ambient temperature (T_4) was measured 2 metres away from the rig. This scenario represented the case of unlagged stack (used as the control).

The second 4 inch pipe was altered to reflect the stack modification proposed in this study. Here, the coil heat exchanger design was adapted. Refer to figure 2.3. A helical groove, 10mm thick and at pitch of 101.6mm (4 inch) was machined on stack pipe (see figure 3.3). A 9.53mm copper tube was brazed on the helical groove and fitted on the stack metal in a helical geometry (see figure 3.4). The assembly was then insulated with a 35.53mm (1 1/4 inch) thick fiberglass insulation (thermal conductivity 0.04 W/mK) (see figures 3.5 (a) and 3.5 (b)). Figure 3.5 (a) and 3.5 (b) shows the cross-section of the modified stack and the relevant stack dimensions. After the insulation, a galvanized 0.5mm sheet metal was used to cover the outside of the insulation by riveting. The choice of copper tube was because the material copper has high thermal conductivity (385 W/m℃) and therefore has low thermal resistance. Also, the highest LMTD is obtained at a D/d ratio of 10 [17]. With stack diameter of 101.6mm and the tube diameter 3/8 inch (9.53mm) the ratio is very close to the requirement. In this assembly the recovery system does not come into contact with flue gases thus avoiding soot fouling. The test set-up for the modified stack is shown in figures 3.2 (b) and 3.6. Additional temperature measurements were; T_7 and T_8 of water in and out of the helical copper tube respectively, T_5 and T_6 the inner surface temperatures of fiberglass lagging at exit and entry respectively.

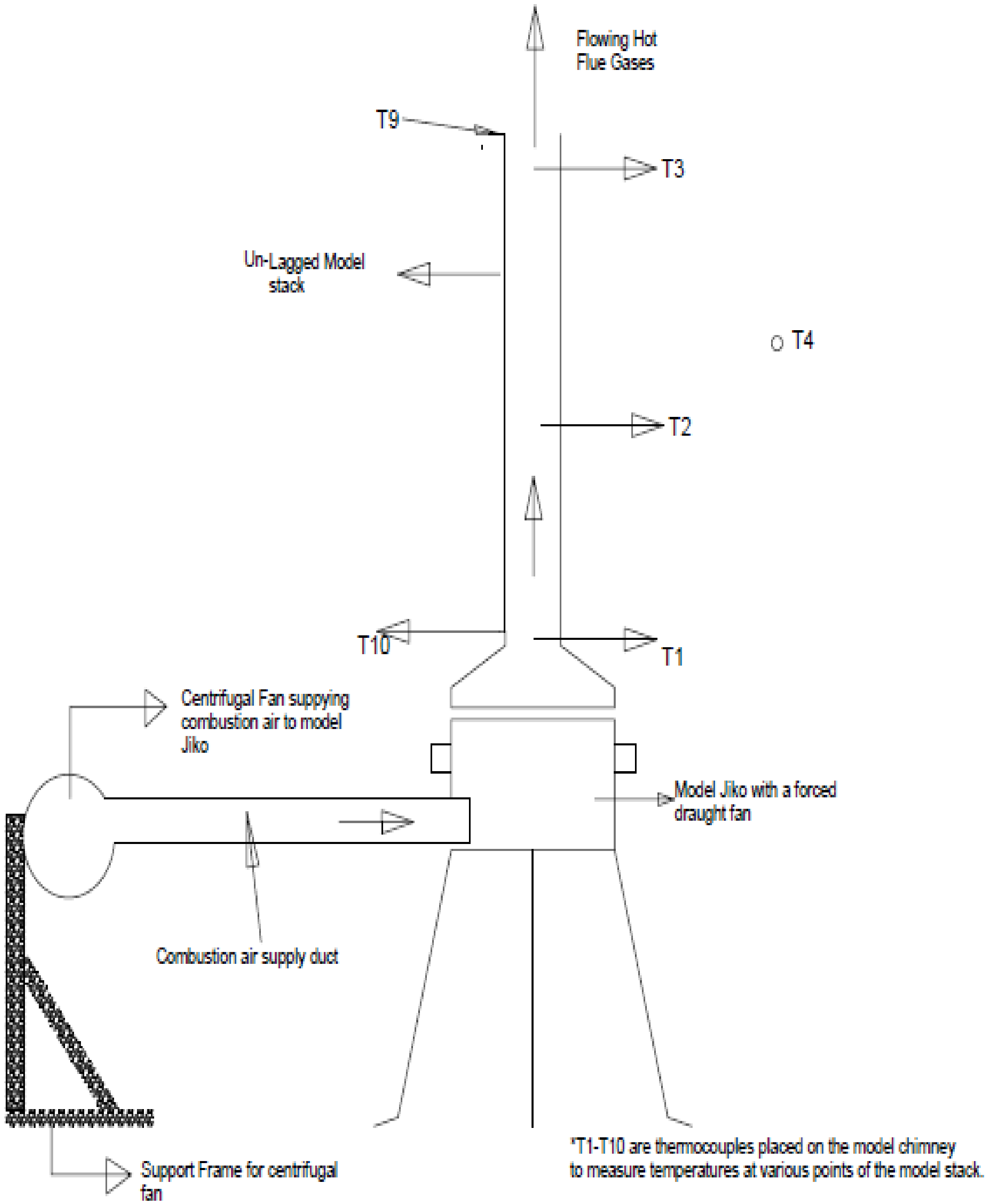


Figure 3.2 (a): Experimental set-up for the Unlagged model chimney stack with fitted j–type thermocouples.

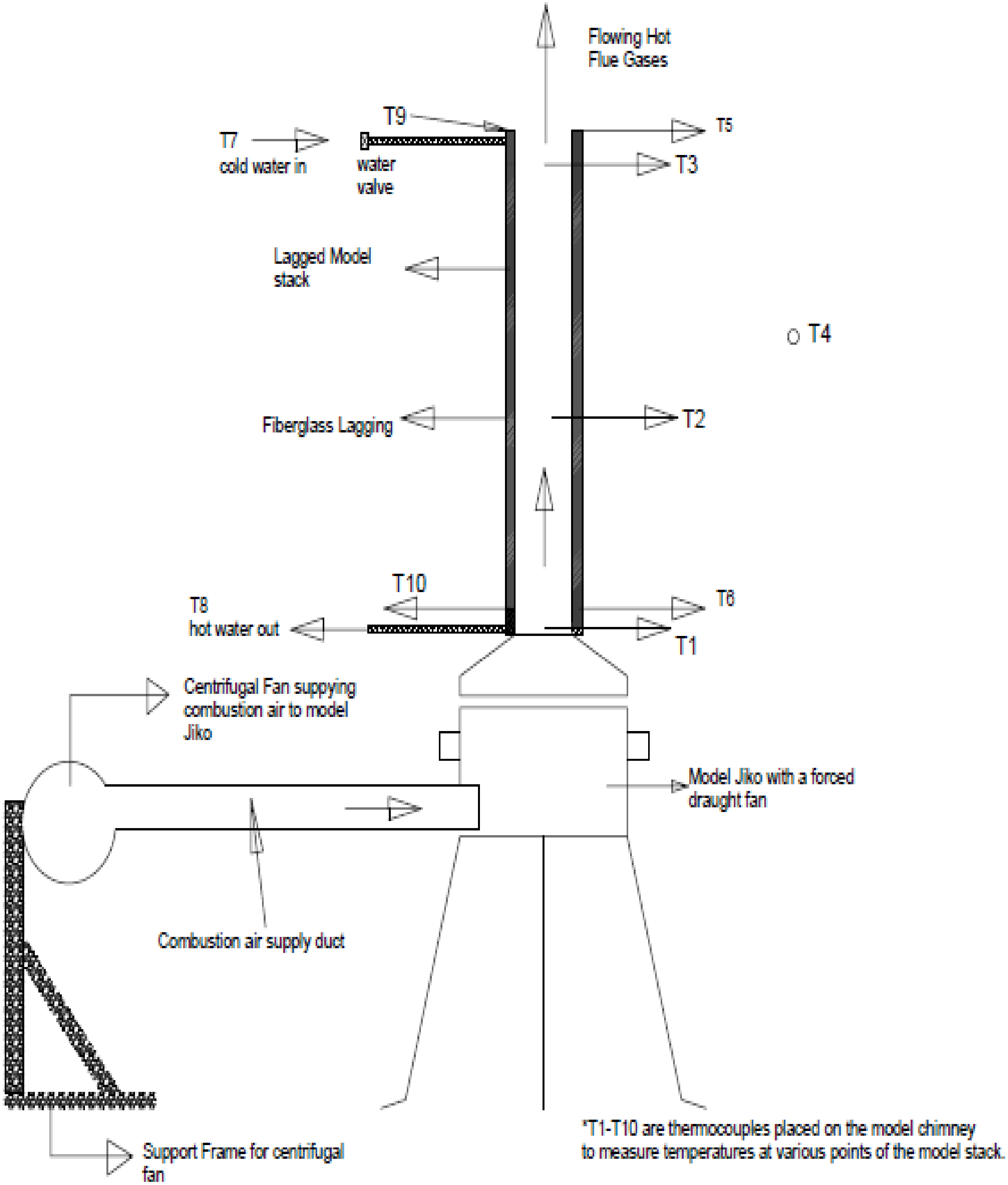


Figure 3.2(b): Experimental set-up for the lagged model chimney stack with fitted j–type thermocouples.

Figure 3.3: Machining the model chimney to create 4inch pitch of 10mm thickness where copper tube was fitted.

Key:

A Lathe Chuck

B Lathe Handwheel

C Lathe Chip Pan

D Lathe Tailstock

E Model stack with 4 inch pitch helical groove machined.

Figure 3.4: 3/8 inch copper tube brazed on the full l height of the model chimney at a pitch of 4 inches.

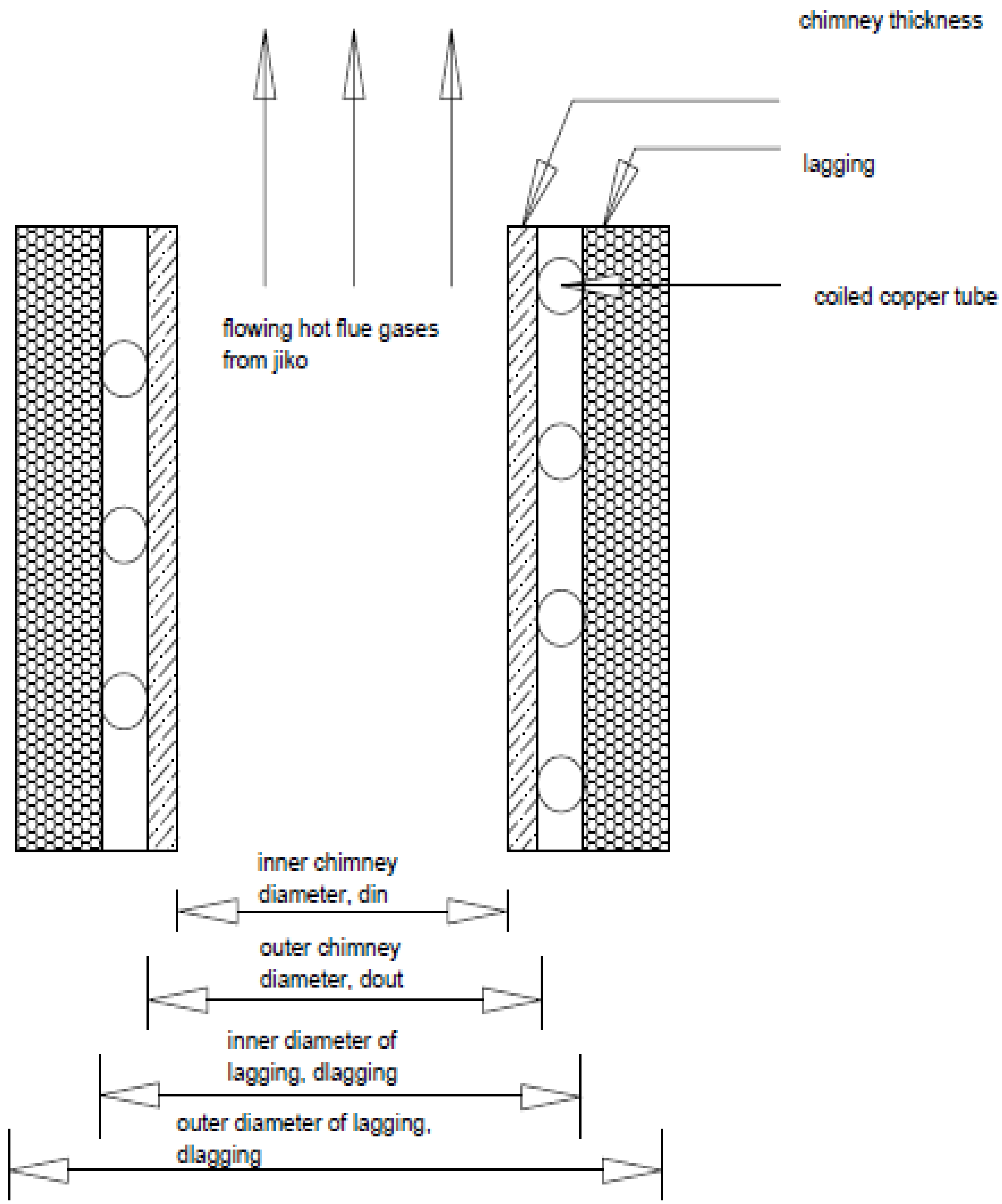


Figure 3.5 (a): Cross section of the modified laboratory model stack.

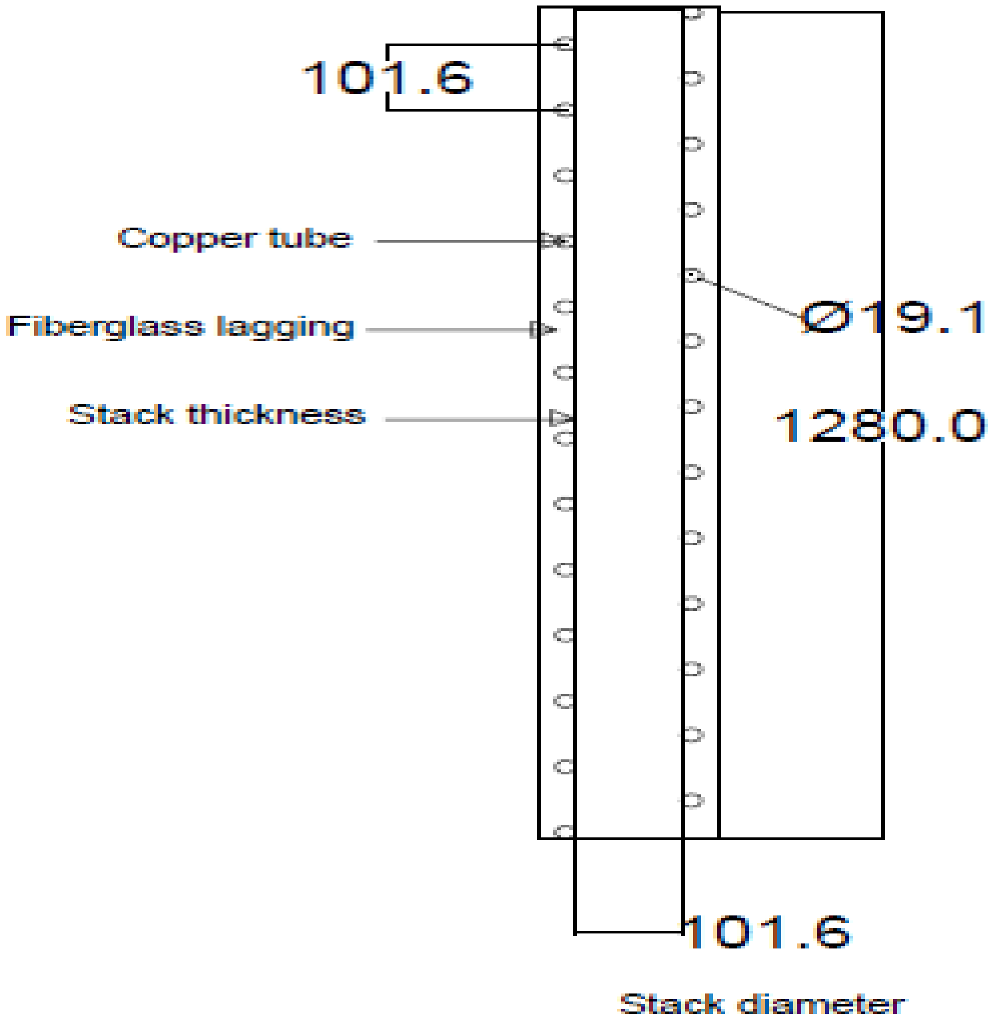


Figure 3.5(b): Lagged laboratory model stack dimensions.

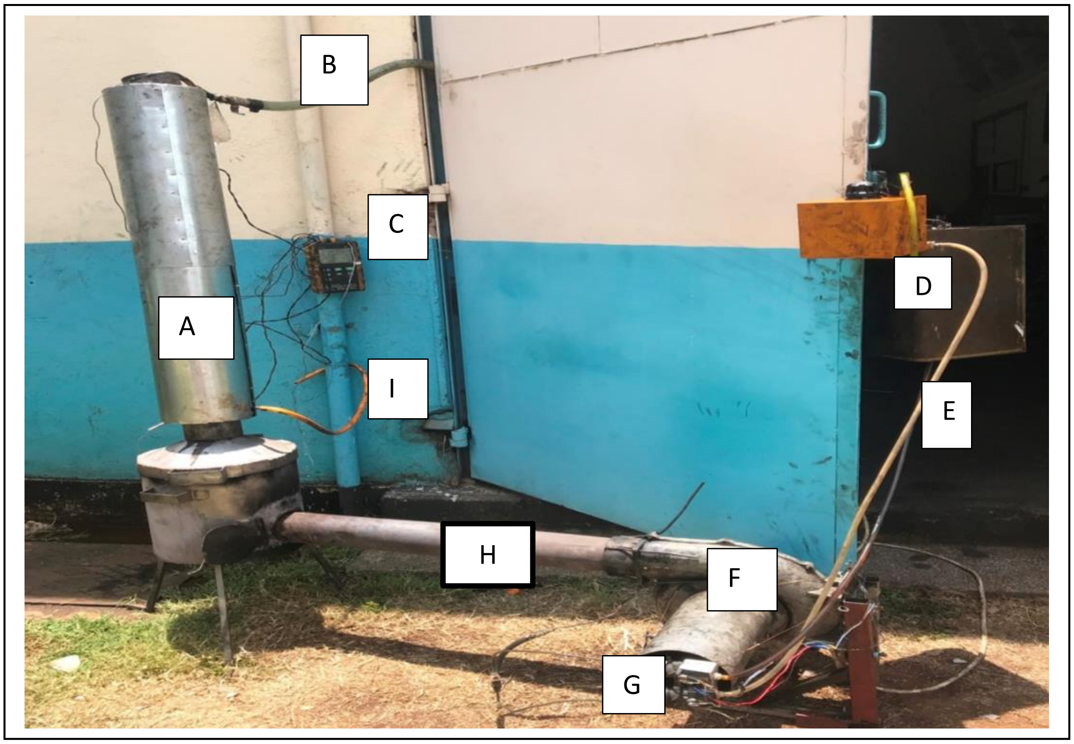


Fig 3.6: Experimental setup waste heat recovery from flue gas.

A Modified stack model under test.

B Water inlet pipe water feed tank with a valve.

C 12 Channel temperature recorder with thermocouples.

D Fuel tanks.

E Fuel piping.

F Fan with damper.

G Fuel pump.

H Horizontal pipe.

I Water outlet pipe.

3.3. EXPERIMENTATION.

Three sets of tests were prepared on the test rig, here called scenario 1, 2 and 3. In the first scenario, the experimental set-up was as in figure 3.2 (a). Flue gases generated by the burner and assisted by a fan were passed through the aluminium painted pipe fixed on the jiko. Under steady conditions the temperatures at various points were recorded. The positions of T_1, T_2 and T_3 were 0.15m, 0.63m and 1.2m from the jiko respectively. Thermocouple T_4 measured ambient temperature and was placed approximately 2meters away from the model stack. Also, positions T_9 and T_{10} were 0.15m and 1.2m vertical height from the jiko respectively. All the thermocouple leads (T_1 – T_{10}) were connected to LUTRON BTM_4298SD 12 channel temperature recorder on which the temperatures were displayed and recorded.

A set of twenty flue gas temperature readings were taken in three hour duration. Also measured was the flue gas velocity at stack exit using PROVA AVM-01 vane anemometer. The flue gas flow was adjusted by a damper fixed on the fan inlet duct. A repeat of the test was done at the new mass flow rate for all scenarios.

In the second scenario, the aluminium painted pipe stack was replaced with the modified one. Aluminium paint is applied for the purposes of preventing corrosion. The other important reason why aluminum paint is used is because the aluminium emissivity is low and this reduces the heat loss from the stack surface by radiation.

Similar tests were undertaken with the modified stack fixed on the jiko. The experimental set-up is shown in Figure 3.2 (b); indicating the various temperature measurement positions. In this experimental setup, additional thermocouples T_5 and T_6 were installed inside the fiberglass lagging at the top and bottom 1.2m and 0.15m from the jiko as shown in figure 3.2 (b). Flue gases were passed through the stack and all temperatures recorded. Also measured was the flue gas velocity at exit. In this second test although the helical copper tube had been installed, water was not introduced in the copper tubes. The set-up was therefore considered as that of an insulated stack. The same set of readings were recorded.

In the third scenario, water was passed through the copper tube as the flue gases flowed through the pipe. The experimental set-up is same as in the second scenario except in addition, temperature measurements T_7 and T_8 representing water inlet and outlet temperatures were recorded. In this scenario, flue gas mass rate was fixed and water flow was varied four times using the installed ball valve (figure 3.2 (b)); between 0.0111 to 0.0118 kg/s. Water flow through the model stack was counter current to the flue gases. The water flow measurement was by a one liter calibrated beaker and a stop watch. This scenario represents a modified stack with a waste heat recovery. The same set of readings was recorded. During the experiment, water continuously passed through the copper tube from a reserve tank whose level was maintained constant. After concluding the experiment, the flue gas flow rate was adjusted and repeat data acquisition taken; for all scenarios.

3.4 COMPUTATIONS:

Heat transfer from the flue gases in the modified stack to water in the copper tubing is by a combination of convection and conduction. Heat from flue gases flows radially across the chimney thickness by conduction. Copper has a thin wall hence small thermal resistance across it. Heat transfer from the flue gases to the water is largely by convection. The heat exchanger assembly cross section is as illustrated in figure 3.5 (a).

A number of values were measured and recorded in section 3.3 including: water mass flow rates, flue gas flow rates, inlet and outlet model stack temperatures, lagging temperature, water temperature into and out of the stack and stack surface temperatures. A number of quantities were calculated using the measured temperatures and velocities in section 3.3. These include;

i) Thermal energy lost by flue gases between stack inlet and outlet was estimated using equation 3.1.

$$Q_L = m_{fg} c_{fg} (T_1 - T_3) \qquad (3.1)$$

ii) The thermal energy in the flue gases was computed assuming that the max energy is recovered when the gas temperature is reduced to ambient - 30°C. That is,

$$Q_{max} = m_{fg} c_{fg} (T_1 - 30°C) \qquad (3.2)$$

iii) The energy gained by water through the modified stack copper tube was estimated by equation 3.3,

$$Q_w = m_w c_{pw}(T_7 - T_8) \quad (3.3).$$

Mass flow rate of water was estimated by measuring the average time taken to fill a one liter calibrated beaker when water flowed through the copper tube. (Volume flow rate in liters/s equals mass flow rate in kg/s as density of water is I kg/m^3).

iv) The fraction of energy recovered to that in the gases was estimated by eq. 3.4. $\% = \frac{Q_w}{Q_L} \times 100$ (3.4)

v) Mass flow rate of flue gases (kg/s) was estimated by eq. 3.5

$$m_{fg} = A * v_{fg} * \rho_{fg} \quad (3.5)$$

v_{fg} is the average measured flue gas velocity through the stack.

vi) The proportion of waste energy absorbed by water from flue gases is obtained as a percentage of the ratio of the energy absorbed by water to the energy contained in the flue gases as expressed in equation 3.6.

% waste heat absorbed by water=$\frac{Q_w}{Q_{max}} * 100\%$ (3.6)

vii) The proportion of energy loss by flue gases through the boiler stack to energy contained by the flue gases was estimated using eq 3.7.

$$\text{Proportion of energy lost through stack} = \frac{Q_L}{Q_{Max}} \quad (3.7)$$

The flue gas properties i.e specific heat capacity and density are functions of temperature. These variation with temperature is given in, appendix , figure D. The values used in this study were interpolated from the same figure [21].

CHAPTER FOUR

RESULTS AND DISCUSSIONS.

4.1 INTRODUCTION.

The purpose of this work was to modify the existing boiler stack in order to facilitate the waste heat recovery from the stack. The project was executed in two parts. The first was a short industrial survey of ten selected boiler stacks. The second part was the construction of a small scale modified laboratory model stack on which experiments were conducted. In this chapter the findings are presented and the corresponding discussions are made.

4.2 BOILER STACK SURVEY.

In part one of this project, a short survey involving ten selected boiler stacks was undertaken. The purpose of the survey was to estimate the energy content of flue gases as they flowed through the stack and determine their status in as far as the waste heat recovery is concerned. At the onset, it was noted that the boilers' capacities were low and varied from 1360 kg/h to 6800kg/h and none had a waste heat recovery system. At these low capacities, the boilers are usually not equipped with either an economizer or an air preheater.

Out of the ten boilers surveyed, two boiler stacks were well lagged (1 – 2); three boiler stacks (3 -5) had ineffective and poorly maintained lagging while the other five (6 – 10) had no lagging at all. Lagging of boiler stacks minimizes temperature drop as flue gases flow through the stack and therefore avoiding condensation. The flue gases retained most of the energy until they are eventually discharged to the environment.

Table 4.1 (a) shows the measured quantities obtained through the boiler stack survey stack's for (Boilers 1 – 5). The temperature drop for boiler 1 and 2 were maximum at 13.3℃. This was because the boiler stacks were well lagged. In these boilers, there was little energy loss through stack surface and the flue energy retained most of the energy until discharge to environment. The flue gas exit temperature on average 210°C is high enough to allow heat recovery from them.

Stacks 3 - 5 inlet temperatures were lower by about 20°C compared to those of 1 – 2 stacks. It was found that the stacks' insulation was poorly maintained and therefore not effective. The outlet temperatures were even much lower, on average 66.6°C. On average the temperature drop between inlet and outlet were about 129 °C. This means that in stacks 3 – 5, there was a significant loss of energy through the stack surface. It is this stack (surface) energy loss that means are sort to recover and hence the basis of this work.

Shown in Table 4.1 (b), are the computed results for boiler stacks 1 – 5. For well insulated stacks (1 – 2) the flue gases lost only 6 – 7% of its energy. However, for poorly insulated stacks (3 – 5), about 78% of energy in flue gases was lost through the stack surface.

Listed in Table 4.2 (a) are the measured temperatures for stacks 6 – 10. These stacks were not lagged at all other than the aluminium paint on their surfaces. The stack flue gas temperature drop was on average 153°C for boiler stacks 6 – 8 and 86°C for 9 – 10 stacks. Most of the energy loss occurred through the unlagged stack surfaces.

Table 4.2 (b) shows the computed results for stacks 6 – 10. In this case, the proportion of the energy lost through the stack surfaces were on average 76.2%. Some of this energy could be recovered if the stack design was modified for this purpose instead of letting it diffuse to the environment. In this project, modification on the stacks is suggested that could facilitate the heat recovery.

The results from the tests are presented and discussed in the sections 4.3. & 4.4

Table 4.1(a): Boiler 1-5 boiler stack survey (lagged).

Description	**Boiler 1**	**Boiler 2**	**Boiler 3**	**Boiler 4**	**Boiler 5**
Boiler capacity (kg/hr)	6800	6800	1500	1500	1500
Ambient boiler room dry bulb temperature (℃)	25.7℃	26.3℃	28.7℃	28.5℃	29.4℃
Average flue gas temperature at chimney entry, ℃ (T_a)	218.5℃	227.0℃	192.2℃	194.9℃	198.0℃
Average flue gas temperature at chimney exit, ℃ (T_b)	205.2℃	215.1℃	66.3℃	65.8℃	67.6℃
Velocity of flue gases (m/s)-v_{fg}	12.2 m/s	12.4 m/s	5.8 m/s	6.1 m/s	5.6 m/s
Stack circumference (m)	2.14m	2.14m	1.06m	1.06m	1.06m

Table 4.1 (b): Computed results for boilers 1-5.

Description	**Boiler 1**	**Boiler 2**	**Boiler 3**	**Boiler 4**	**Boiler 5**
Stack cross-sectional area(m^2)- **A**	$0.36m^2$	$0.36m^2$	$0.1m^2$	$0.1m^2$	$0.1m^2$
Velocity of flue gases (m/s)-v_{fg}	12.2 m/s	12.4 m/s	5.8 m/s	6.1 m/s	5.6 m/s
Density of flue gases at flue gas exit temperature (kg/m^3)- $\boldsymbol{\rho_{fg}}$	$0.7228kg/m^3$	$0.7126kg/m^3$	$0.7637kg/m^3$	$0.7583kg/m^3$	$0.752kg/m^3$
Mass flow rate of flue gases (kg/s)-$\boldsymbol{m_{fg}=A^* v_{fg} * \rho_{fg}}$	**3.175 kg/s**	**3.181 kg/s**	**0.443 kg/s**	**0.463 kg/s**	**0.421 kg/s**
Specific heat capacity of flue gases (kj/kgK)-c_{fg}	1.102 kj/kgK	1.104 kj/kgK	1.095 kj/kgK	1.096 kj/kgK	1.096 kj/kgK
Temperature drop across boiler stack, (T_a-T_b) ℃	13.3℃	11.9℃	125.9℃	129.1℃	130.4℃
$\boldsymbol{Q_L}$- $\boldsymbol{m_{fg}c_{fg}(T_a-T_b)}$ [kJ]	**46.53kJ**	**41.79kJ**	**61.07kJ**	**65.46 kJ**	**60.17kJ**
Max temperature drop by flue, (T_a-30℃)℃	188.5℃	197.0℃	162.2℃	164.9℃	168.0℃
$\boldsymbol{Q_{max}}$, = $\boldsymbol{m_{fg}c_{fg}(T_a-30)}$ [kJ]	**659.53kJ**	**691.83kJ**	**78.68kJ**	**83.68kJ**	**77.52kJ**
$\frac{Q_L}{Q_{max}} \times 100$	**7.06%**	**6.04%**	**77.62%**	**78.22%**	**77.62%**

Table 4.2(a): Boiler 6-10 boiler stack survey (not lagged).

Description	**Boiler 6**	**Boiler 7**	**Boiler 8**	**Boiler 9**	**Boiler10**
Boiler capacity (kg/hr)	2000	1500	2820	4,000	1360
Ambient boiler room dry bulb temperature (◦C)	29.6℃	32.4◦C	31.4℃	23℃	32.5℃
Average flue gas temperature at chimney entry, ℃ (T_a)	223.6℃	221.0℃	217.2℃	147.4℃	153.0℃
Average flue gas temperature at chimney exit, ℃ (T_b)	81.6℃	64.7℃	61.8℃	56.6℃	71.8℃
Velocity of flue gases (m/s)-v_{fg}	3.7 m/s	4.1 m/s	4.7 m/s	4.8 m/s	3.8 m/s
Stack circumference (m)	1.26m	1.95m	2.58m	2.26m	0.28m

Table 4.2 (b): Computed results for boilers 6-10.

Description	**Boiler 6**	**Boiler 7**	**Boiler 8**	**Boiler 9**	**Boiler10**
Stack cross sectional area(m^2)-**A**	$0.24m^2$	$0.30m^2$	$0.53m^2$	$0.41m^2$	$0.28m^2$
Velocity of flue gases (m/s)-v_{fg}	3.7 m/s	4.1 m/s	4.7 m/s	4.8 m/s	3.8 m/s
Density of flue gases at flue gas exit temperature (kg/m^3)- $\boldsymbol{\rho_{fg}}$	0.717 kg/m^3	0.7205 kg/m^3	0.7255 kg/m^3	0.8543 kg/m^3	0.8429 kg/m^3
Mass flow rate of flue gases (kg/s)-$m_{fg}=A * v_{fg} * \rho_{fg}$	**0.637 kg/s**	**0.886 kg/s**	**1.807 kg/s**	**1.681 kg/s**	**0.897 kg/s**
Specific heat capacity of flue gases (kj/kgK)-c_{fg}	1.103 kj/kgK	1.102 kj/kgK	1.101 kj/kgK	1.082 kj/kgK	1.083 kj/kgK
Max temperature drop across boiler stack, (T_a-T_b) ℃	142.0℃	156.3℃	155.4℃	90.8℃	81.2℃
Q_L- $m_{fg}c_{fg}(T_a-T_b)$ [kJ]	**99.77kJ**	**152.61kJ**	**309.17 kJ**	**165.15 kJ**	**78.88kJ**
Temperature drop by flue, (T_a-30℃)℃	193.6℃	191.0℃	187.2℃	117.4℃	123.0℃
Q_{max}, = $m_{fg}c_{fg}(T_a-30)$ [kJ]	**136.0kJ**	**186.49kJ**	**372.44 kJ**	**213.53 kJ**	**119.49 kJ**
$\frac{Q_L}{Q_{max}} \times 100$	**73.4%**	**81.83%**	**83.01%**	**77.52%**	**66.01%**

4.3. STACK MODEL.

The second part of this project was to construct a stack model in which the flue gases are conveyed out of the boiler and also recover waste heat from the stack surfaces. A number of tests were preferred on the model in order to estimate the waste heat recovery.

4.4 EXPERIMENTATION.

Tests or experiments on the model stack were performed in three scenarios. In the first scenario, flue gases were conveyed through the stack with no waste heat recovery and the model stack was not lagged. The average values are listed in table 4.3 (Scenario 1). There was an average temperature drop of 262.3 ℃ over the height of the model stack. Because of the high temperatures, the heat loss through the surface was also high. This scenario presents a high potential for waste heat recovery from the stack surfaces.

It should be noted that in table 4.3, experiment 1(a) means a test in scenario 1(bare pipe) with flue gas rate of (a) 0.0166 m 3 /s and similarly, experiment 1(b) was a bare pipe but with flue gas rate of (b) 0.0235 m 3 /s. The increase of the flue gas flow was achieved by adjusting the damper at fan inlet. The temperatures listed in table 4.3 were the average of the twenty sets of readings recorded in a duration of 3 hours.

Table 4.3: Bare stack experiment results- Scenario 1.

	Average temperature measured, ℃									
	T1	***T2***	***T3***	***T4***	***T5***	***T6***	***T7***	***T8***	***T9***	***T10***
Experiment 1a	384.7	195.0	122.4	27.0	-	-	-	-	81.9	100.9
Experiment 1b	371.2	223.0	152.3	27.5	-	-	-	-	98.8	118.8

In scenario 2 and 3, the bare pipe stack was replaced by a modified one. The results for scenario 2 are shown in table 4.4 (Scenario 2). In this scenario, water was not passed through the copper tube helical coil. The stack in this case was hence considered insulated with no heat recovery. The temperature drop was on average 205.5℃. The stack surface temperatures were 71.5°C at stack inlet and 54.8°C at exit. Although the stack was insulated, the high surface temperatures indicate still a high rate of heat loss through the surfaces. It seems the lagging on the pipe was not sufficient. To minimize the heat loss and therefore lower surface temperatures, more insulation is required. The data represents two flue gas flow rates 0.0166m3/s denoted experiment 2 (a) and 0.0235 m3/s denoted experiment 2 (b).

Table 4.4: Modified model stack experiment results- Scenario 2.

	Average temperature measured, °C									
	T1	***T2***	***T3***	***T4***	***T5***	***T6***	***T7***	***T8***	***T9***	***T10***
Experiment 2a	390.6	248.4	185.1	27.0	109.5	125.0	-	-	54.8	71.5
Experiment 2b	323.8	224.7	166.1	27.1	104.9	113.8	-	-	49.2	64.0

In scenario 3, water was passed through the copper tube at different rates i.e 0.0111kg/s, 0.0114kg/s, 0.0115kg/s and 0.0118 kg/s, counter to the flue gases. This was repeated for the two flue gas flow rates. The outlet water temperature T_7 decreased with increased water flow rates. The surface temperature did not show any trend with variation of the water flow.

Table 4.5 (a): Modified model stack experiment results (Flue gas flow rate 0.0166m^3/s)-Scenario 3.

	Average temperature measured, °C									
	T1	***T2***	***T3***	***T4***	***T5***	***T6***	***T7***	***T8***	***T9***	***T10***
Experiment 3a	366.4	207.2	149.1	27.7	70.0	84.6	51.5	22.9	54.7	67.3
Experiment 4a	327.2	200.8	135.7	27.7	78.3	91.4	48.7	22.7	63.9	71.3
Experiment 5a	345.5	203.6	146.5	27.8	72.2	89.1	47.1	22.7	56.2	66.6
Experiment 6a	333.4	185.6	134.3	28.0	68.5	76.5	46.5	22.6	54.0	67.8

Table 4.5 (b): Modified model stack experiment results (Flue gas flow rate 0.0235m^3/s)- Scenario 3.

	Average temperature measured, °C									
	T1	***T2***	***T3***	***T4***	***T5***	***T6***	***T7***	***T8***	***T9***	***T10***
Experiment 3b	339.7	182.5	140.8	27.4	63.5	75.9	49.8	22.5	51.1	63.0
Experiment 4b	315.9	198.5	130.1	28.0	70.5	79.2	48.5	22.5	55.2	66.6
Experiment 5b	339.3	206.2	145.1	28.0	64.7	76.1	46.4	22.6	57.4	70.9
Experiment 6b	311.5	193.9	128.4	27.6	65.7	74.1	45.7	22.8	54.8	65.2

From table 4.3 (Scenario 1), the bare stack gas outlet temperature was 122.4°C. This is lower than that for the insulated stack and for the modified one. The bare stack loses much more energy from its surface. When insulated, less heat was lost resulting to higher exit temperature – 185°C (table 4.4-Scenario 2). When water is passed through the modified stack, it absorbs energy from the flue gases, resulting in lower gas temperature at exit - 149°C (table 4.5-Scenario 3). In this experiment, the water temperature increased by 29°C after flowing through the stack. This is a measure of energy recovery. Further, the stack surface temperature was 54 – 67 °C, the lowest of the three scenarios.

4.5 WASTE HEAT RECOVERY

From the measured temperatures, volume of flue gas and the water mass flow on the test rig, a number of parameters were computed. The parameters enabled the determination of waste heat recovery from flue gases.

Table 4.6 shows the computed results of the three scenarios. The flue gas temperature drop between inlet and outlet (T_1 – T_3) was highest with the bare pipe (scenario 1) at 262°C, followed by the insulated stack 205°C and least for modified stack at 191.5°C. The energy in flue gases that was lost through the stack surface as a fraction of the total energy in flue gases were 0.739, 0.570, and 0.644 (eq 3.7) for scenarios 1, 2 and 3 respectively. The difference between scenario 2 and 3 was largely due to the water flowing through the modified stack. The percentage energy loss for scenario 1 mirrors that of unlagged stacks surveyed in part one of this project.

Shown in table 4.6 are computed quantities associated with flue gas as it passed through the stack. The bare stack (experiment 1(a) and 1(b)) loses more energy from the stack surface in view of the higher flue gas temperature drop compared to the modified stack (experiment 3(a) and 3(b)). At 0.0166m3/s flue gas volume flow rate, the temperature drop across the model stack is higher than when the volume flow rate of flue gas is 0.0235 m 3 /s.

Table 4.6: Computed results for laboratory experiments.

	m_{fg} (Kg/s)	C_{fg} (KJ/KgK)	$(T_1\text{-}T_3)$ (℃)	$(T_1\text{-}30)$ (℃)	Q_L (KW)	Q_{max} (KW)	$\frac{Q_L}{Q_{max}} \times 100$
Experiment 1a	*0.0134*	*1.115*	262.3	*354.7*	*3.919*	*5.300*	*73.9%*
Experiment 1b	*0.0172*	*1.119*	218.9	*341.2*	*4.213*	*6.567*	*64.1%*
Experiment 2a	*0.0106*	*1.126*	*205.5*	*360.6*	*2.453*	*4.304*	*57.0%*
Experiment 2b	*0.0159*	*1.116*	*157.7*	*293.8*	*2.798*	*5.213*	*53.7%*
Experiment 3a	*0.0117*	*1.117*	*217.3*	*336.4*	*2.840*	*4.396*	*64.6%*
Experiment 3b	*0.0170*	*1.121*	*198.9*	*309.7*	*3.790*	*5.902*	*64.2%*
Experiment 4a	*0.0122*	*1.111*	*191.5*	*297.2*	*2.596*	*4.028*	*64.4%*
Experiment 4b	*0.0174*	*1.120*	*185.8*	*285.9*	*3.621*	*5.572*	*65.0%*
Experiment 5a	*0.0117*	*1.114*	*199.0*	*315.5*	*2.594*	*4.112*	*63.1%*
Experiment 5b	*0.0172*	*1.114*	*194.2*	*309.3*	*3.721*	*5.926*	*62.8%*
Experiment 6a	*0.0121*	*1.120*	*199.1*	*303.4*	*2.698*	*4.112*	*65.6%*
Experiment 6b	*0.0182*	*1.118*	*183.1*	*281.5*	*3.726*	*5.728*	*65.0%*

Table 4.7 estimates the amount of energy contained by flue gases between inlet and outlet, energy recovered by water and the proportion of energy recovered by water. (equation 3.1, 3.3 and 3.6 respectively). The change of water temperature as it flowed through the stack also indicates or is a measure of the energy recovered from the flue gas. Figure 4.1 shows the increase of water temperature at varying mass flow rate. As expected the increase in temperature change is high at low water flow rates. For example, at 0.0114 kg/s (experiment 4(a), table 4.6) the flue gas temperature drop was 191.5°C. The water temperature rise through the stack tube was by 26.0°C (48.7 – 22.7,

table 4.7). The recovered energy due to the rise was estimated to be 1.241 kW (table 4.7). The total (max) energy contained in flue gas was 4.051 kW (table 4.7). Therefore the energy absorbed (recovered) from the flue gases was 30.6% of the total gas energy (See table 4.7).

Figure 4.2 shows the proportion of waste heat energy recovered as a fraction of the energy contained in the flue gases at various water rates. The % energy recovered does not change much with water flow rate but is a strong function of the volume rates of the flue gas. The % energy recovered is much higher at 0.0166 m^3 /s compared to that at 0.0235 m^3 /s. The total energy in flue gases cannot be recovered fully as this would require a very tall (large surface area) stack to enable the outlet flue gas temperature approach the ambient. The energy loss by the stack surface is given by the change in temperature (T_1 – T_3) between inlet and outlet. For experiment 4(a) energy lost by flue gases was 2.596 kW. Hence the fraction of this amount recovered was 47.8%.

Table 4.7: Computed results of energy for energy recovered by laboratory model.

	M_w (Kg/s)	$(T_7\text{-}T_8)$ (℃)	Q_w (KW)	Q_L	Q_{max} (KW)	$\frac{Q_W}{Q_L} * 100$	$\frac{Q_w}{Q_{max}} * 100$
Experiment 1a			-	*3.919*	5.300	-	-
Experiment 1b			-	*4.213*	6.567	-	-
Experiment 2a			-	*2.453*	4.304	-	-
Experiment 2b			-	*2.798*	5.213	-	-
Experiment 3a	0.0111	28.6	1.329	*2.840*	4.396	*46.8%*	*30.2%*
Experiment 3b	0.0111	27.3	1.269	*3.790*	5.902	*33.5%*	*21.5%*
Experiment 4a	0.0114	26.0	1.241	*2.596*	4.051	*47.8%*	*30.6%*
Experiment 4b	0.0114	26.0	1.241	*3.621*	5.572	*34.3%*	*22.3%*
Experiment 5a	0.0115	24.4	1.175	*2.594*	4.112	*45.3%*	*28.6%*
Experiment 5b	0.0115	23.8	1.146	*3.721*	5.926	*30.8%*	*19.3%*
Experiment 6a	0.0118	23.9	1.181	*2.698*	4.112	*43.8%*	*28.7%*
Experiment 6b	0.0118	22.9	1.131	*3.726*	5.728	*30.4%*	*19.8%*

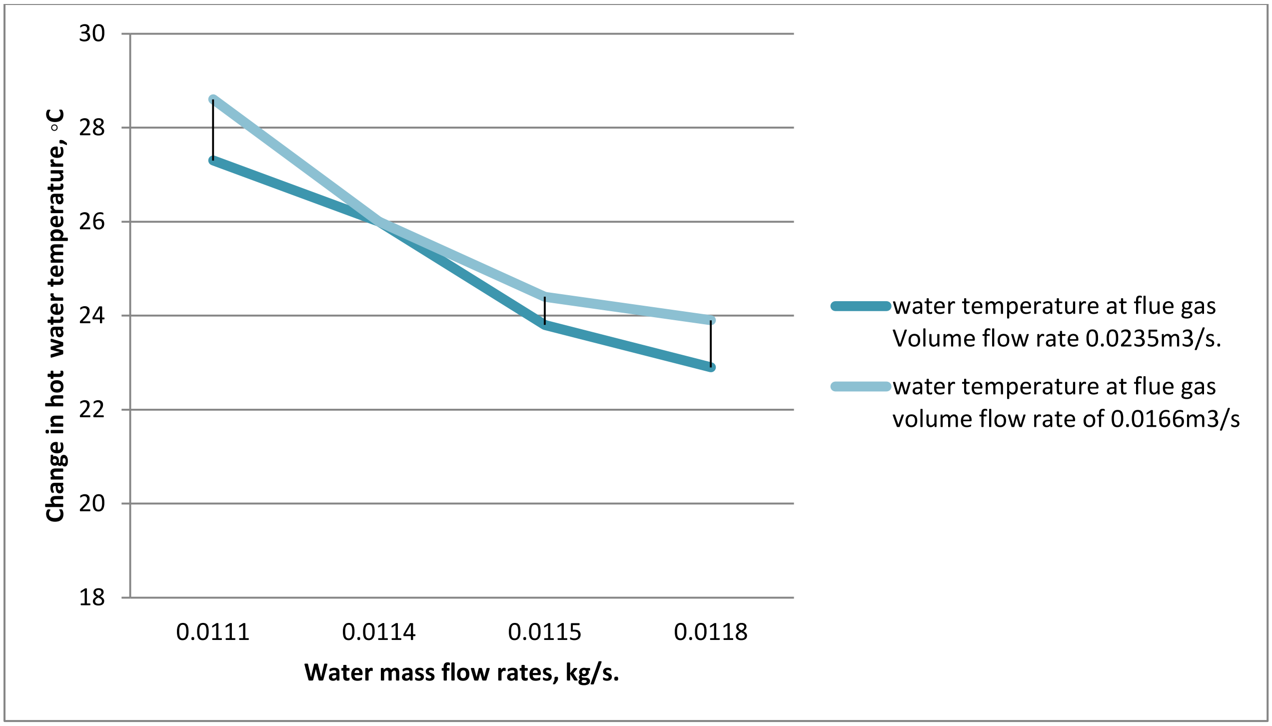


Figure 4.1: Change in hot water temperature with water flow rates, kg/s.

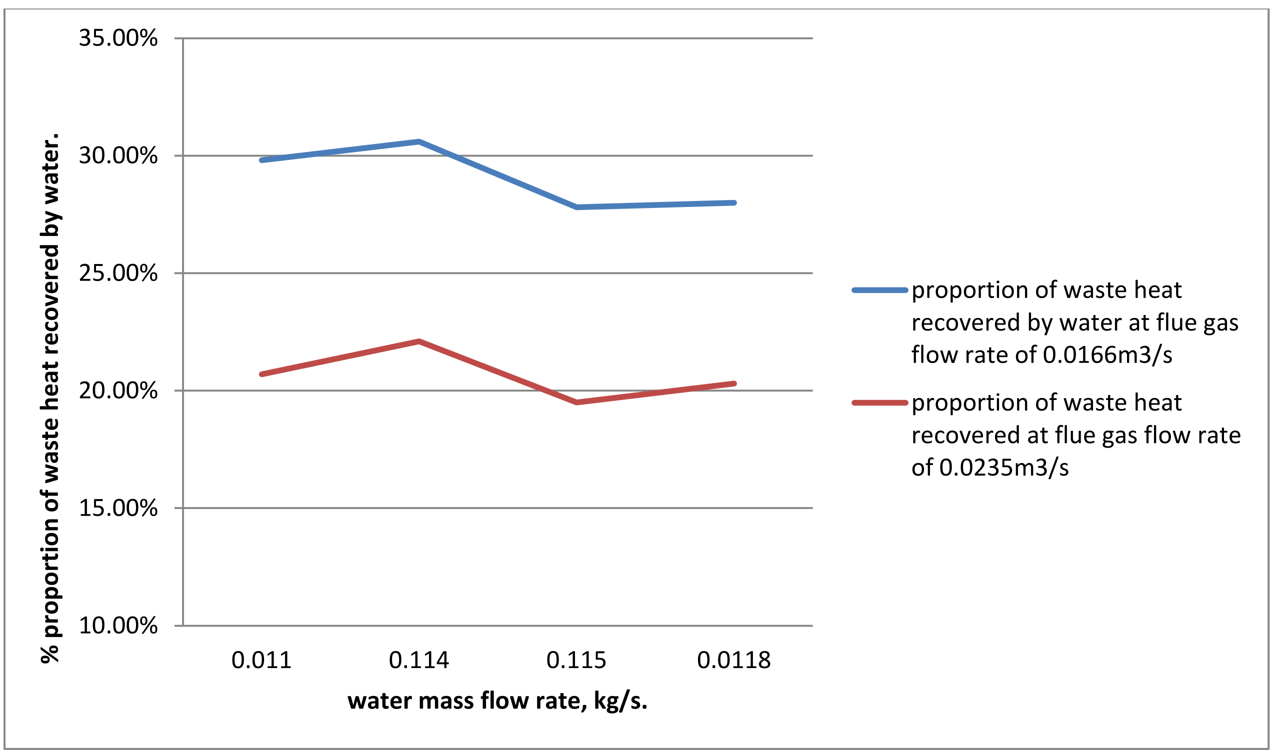


Figure 4.2: Proportion of waste energy recovered with water mass flow rates (kg/s).

CHAPTER FIVE

CONCLUSIONS AND RECOMMENDATIONS.

5.1 INTRODUCTION.

This project was an attempt to modify boiler stack in order to recover waste heat from the exhaust gases. A short industrial survey of existing boiler stacks was done to estimate the energy content of flue gases at entry to the stack and as it flows through it. Also noted was the condition of the stack insulation and therefore the heat loss through the surface. A stack test rig and a modified laboratory scale stack were constructed. A number of tests were carried out to quantify the energy loss from the surface and the waste heat recovery possible. The following are the project findings.

5.2 STACK SURVEY.

Ten boilers of relatively low capacity (< 6800 kg/h) had their stacks studied. Two stacks (1 – 2) were well insulated and kept at good condition. The temperature of gases at inlet and exit were on average 223°C and 210°C respectively. Flue gas maintained the heat energy up to when they were discharged to the environment. The high exit temperatures present an opportunity for heat recovery. The potential temperature drop was 143°C (210-67°C).

Three of the stacks (3 – 5) were insulated. However, the insulation was poorly maintained that rendered it less effective. The inlet gas temperature was on average 195°C and 66.5°C at exit. Even at slightly lower inlet gas temperature, the temperature drop in the stacks was on average 128°C. Most of the energy in the flue gas was lost through the stack surface.

Half of the surveyed stacks (6 – 10) were not insulated at all except for the aluminium paint. Their temperatures were on average 192°C at stack entry and 67°C at exit. The temperature drop was therefore 125°C between inlet and outlet. Again, most of the energy in the flue gases was lost through the stacks' surface.

5.3 THE STACK MODEL.

5.3.1 Stack Fabrication.

A small scale modified model stack was fabricated upon which tests were performed. In modifying the stack, the coil heat exchanger design was adapted The modified stack consisted of a 4 inch (101.6mm) diameter mild steel cylindrical pipe on which a helical 3/8 inches (9.53mm) copper tube was wound on full length (1280mm) of the cylindrical pipe. The assembly was then insulated with fiberglass. Combustion gases from a diesel burner were passed through the stack model and water was passed through the helical copper tube. This arrangement (coil heat exchanger design) allowed heat transfer from the flue gases to the water flowing through the tube.

5.3.2 Experimentation.

A number of tests or experiments were performed on the model stack; in three scenarios. The first scenario involved a bare pipe fixed on a test rig. There was no insulation and no heat recovery system. The flue gas generated by a burner and assisted by a fan, was passed through the stack. As the gas flowed through the stack, temperatures at various positions were recorded. In a typical test, flue gas stack inlet temperature was 384.7°C while the outlet (no lagging and no heat recovery) was 122.4◦C. This was equivalent to 204.9◦C temperature drop per meter, indicating a high heat loss through the stack surface.

In the second scenario, hot gases were directed into the modified model stack but the water was not allowed in the copper tube. This arrangement was considered equivalent to an insulated stack. The temperature difference between inlet and outlet was 205.5◦C, i.e. a temperature drop of 160.5℃ per meter. The increase of outlet temperature as compared to bare pipe was due to stack insulation.

The third scenario involved passing water through the copper tube as flue gases pass through the stack – i.e. modified stack with waste heat recovery. The first set of tests were performed when the flue gas was fixed at 0.0166 m^3 /s. Temperature at various points such as inlet, outlet, surface temperature, water temperature inlet and outlet as well as the water flow rate in the copper tube were recorded. The second set tests were performed after flue gas flow was adjusted to 0.0235 m^3 /s and the readings repeated. At either of the flue gas volume rate, the water flow was varied four times from 0.0111 kg/s to 0.0118 kg/s. From the various readings, a number of parameters were computed. These included –max energy content of the flue gas that could be recovered, energy lost by flue gases in the stack, energy absorbed (recovered) by water from the stack, proportion of waste heat in the stack that was recovered and proportion of energy recovered to that lost through the stack surface. For example, when water was passed at 0.0114kg/s, flue gas (at 0.0166 m 3 /s) temperature drop was 191.5 ℃ and the water temperature rose from 22.7℃ to 48.7℃. This represented 30.6% of the energy in the flue gases. The energy recovered was about 48% of the energy lost through the stack surfaces.

5.4 RECOMMENDATIONS.

The accuracy of the laboratory experimental results could improve by use of temperature recorder with data logging and recording capabilities assisted with CFD simulation software. The use of a calibrated digital flow-meter to measure cold water flow rate could improve experimental data accuracy. Computer simulation tools could be accurately used to estimate waste energy recovered for the model stack and the results used to estimate waste energy recoverable in industrial boiler stacks. The temperatures were manually taken and recorded; data acquisition systems can store more data accurately. Further investigation can be done with a taller and larger diameter stack. The surface temperature for all scenarios was rather high. The fiberglass insulation was not adequate and more could be added or changed. Further investigation on the coil heat exchanger using a double helical coil for waste heat recovery could be conducted.

REFERENCES

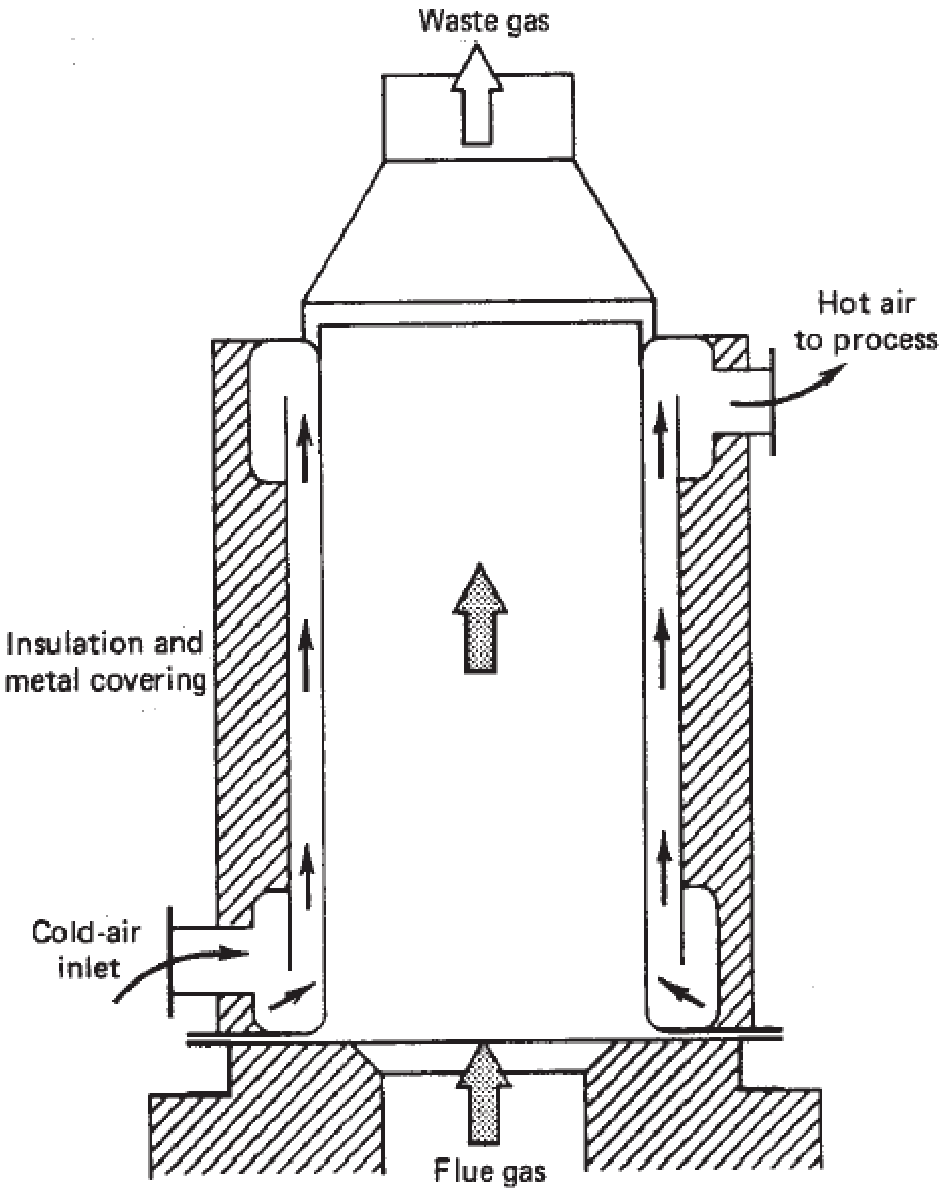
1. Energy technology network, energy technology systems analysis programme', IEA ETSAP-technology brief 101-May 2010.
2. Saidur R, Ahamed JU, Masjuki HH. Energy, 'energy and economic analysis of industrial boilers'. Energy Policy 2010; 38(5):2188–97.
3. Spirax Sarco,An explanation of specialist boiler types and other specialist feature,Spirax Sarco,Cheltenham 2011.
4. A. Bhatia ,'Improving Energy Efficiency of Boiler Systems.'
5. P.C. Lu T.T. Fu,S.C. Garg and G. Nowakowski; 'Boiler stack gas heat recovery', Sponsored by Naval Engineering Command, September 1987, N-1776.
6. Rahul Dev Gupta, Sudhir Ghai, Ajai Jain. "Energy Efficiency Improvement Strategies for Industrial Boilers: A Case Study". Journal of Engineering and Technology. Vol 1. Issue 1. Jan-June 2011.
7. R. Saidur n, J.U.Ahamed,H.H.Masjuki, Energy, exergy and economic analysis of industrial boilers, (2003) Elsevier Ltd. doi:10.1016/j.enpol.2009.11.087.
8. P.Ravindra Kumar, V.R.Raju, N.Ravi Kumar, Ch.V.Krishna, Laki Reddy, Bali Reddy, 'Investigation of Improvement in Boiler Efficiency through Incorporation of Additional Bank of Tubes in the Economiser for Supercritical Steam Power Cycles,' International Journal of Engineering Research and Development eISSN : 2278-067X, pISSN :2278-800X, www.ijerd.com Volume 4, Issue 8 (November 2012), PP. 94-100
9. Energy management services, 'Boiler tune-up guide for natural gas and light fuel oil operation,' GREG HARRELL, PH.D., P.E.
10. Wayne C. Turner, Steve Doty, 'Energy Management Handbook,' Sixth Edition. The Fairmont Press Inc. 2007.
11. U.S Department of Energy, 'Waste heat recovery: Technology and opportunities in U.S. Industry,' industries technology program. BCS, Incorporated, March 2008.
12. C.R.Wilson, 1982, Modern boiler economizers –Developments & applications, Volume (Issue) 2(2).

13. Einstein, Dan, Worrell, Ernst, Khrushch, Marta, Steam systems in industry: Energy use and energy efficiency improvement potentials, 2001, Lawrence Berkeley National Laboratory.
14. Conservation CIPfE, Dockrill P, Ontario. Boilers and Heaters: Improving Energy Efficiency. CIPEC; 2001.
15. 'Helically Coiled Heat Exchangers' by J. S. Jayakumar.
16. Rithy Kong, Thoranis Deethayat, Attakorn Asanakham, Tanongkiat Kiatsiriroat, 'Heat transfer phenomena on waste heat recovery of combustion stack gas with deionized water in helical coiled heat exchange,' Case Studies in Thermal Engineering 12 (2018).
17. Gopal Kumar Deshmukh, Kuvadiya Manish N, Rankit A. Patel, Ramesh H. Bhoi, " Parametric Analysis of Tube in Tube Helical Coil Heat Exchanger at Constant Wall Temperature, IJSTE-International Journal of Science Technology & Engineering, Volume 1, Issue-10,April 2015.
18. Bibin Prasad, Sujith V, Mohammed Shaban K, Saju Haneef, Sandeep N, Vishnu Raj,' Comparison of Heat Transfer between a Helical and Straight Tube Heat Exchanger,' International Journal of Engineering Research and Technology. ISSN 0974-3154 Volume 6, Number 1 (2013).
19. Devendra Borse, Jayesh V. Bute,' A Review on Helical Coil Heat Exchanger,' International Journal for Research in Applied Science & Engineering Technology (IJRASET) ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 6.887, Volume 6 Issue II, February 2018.
20. M Balachandran, "Experimental and CFD study of a helically coiled heat exchanger using water as a fluid ", International Journal of Mechanical and Production Engineering, ISSN: 2320- 2092, Vol 3, Issue-5, May-2015.
21. https://www.pipeflowcalculations.com/tables/flue-gas.xhtml.
22. R.K. Rajput, "Engineering Thermodynamics", third edition-2007, Laxmi Publications (P) Ltd.
23. https://keison.co.uk/products/kanemay/905.pdf.
24. https://dam-assets.fluke.com/s3fs-public/6008893_0000_ENG_A_W.PDF.
25. http://lutron.co.kr/database/pdf/YK-2005TM.pdf.

26. https://micronix.eu/data/eu/att/002/4032-1988.pdf.
27. https://www.prova.com.tw/img/download/AVM01&03-DATA%20SHEET-2017.pdf.

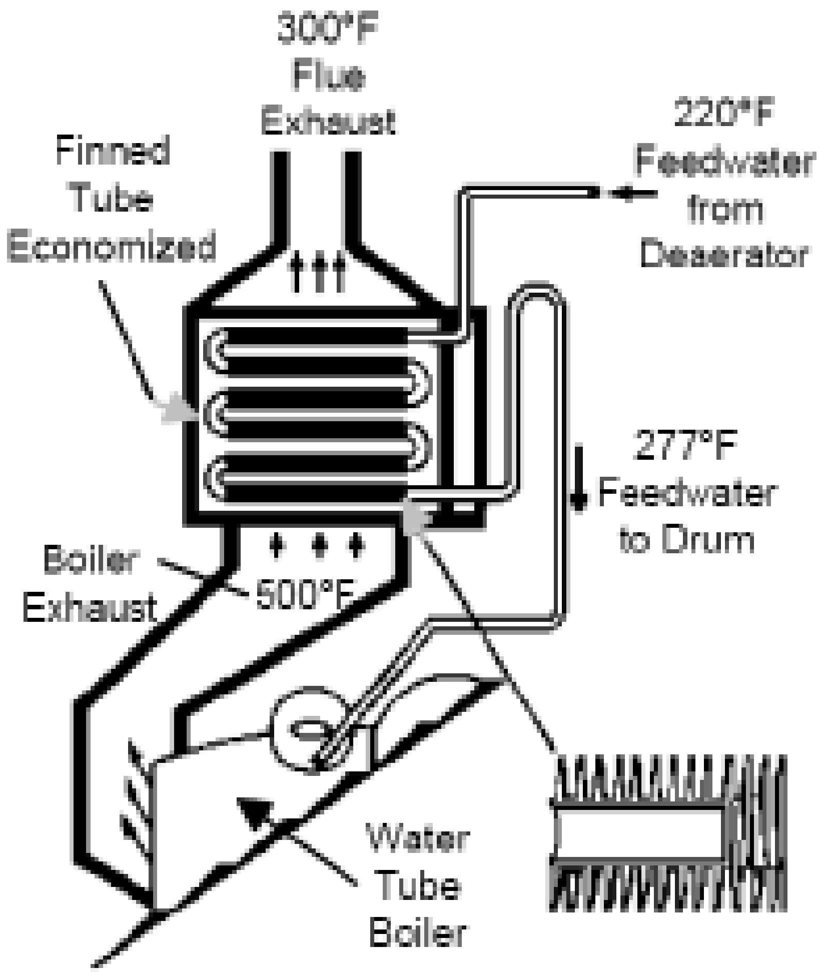
APPENDICES

APPENDIX A



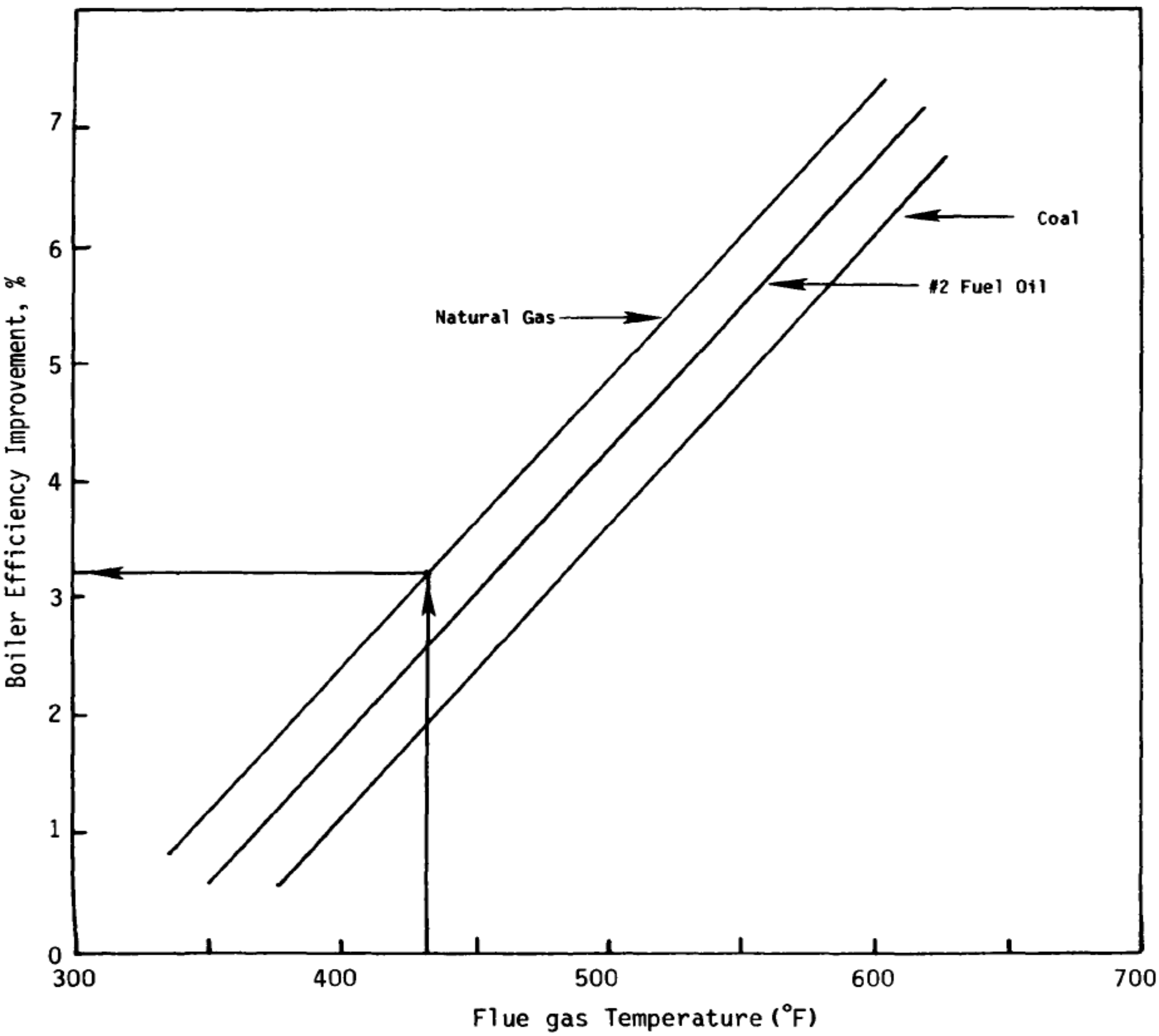
Air preheater showing air movement [10].

APPENDIX B



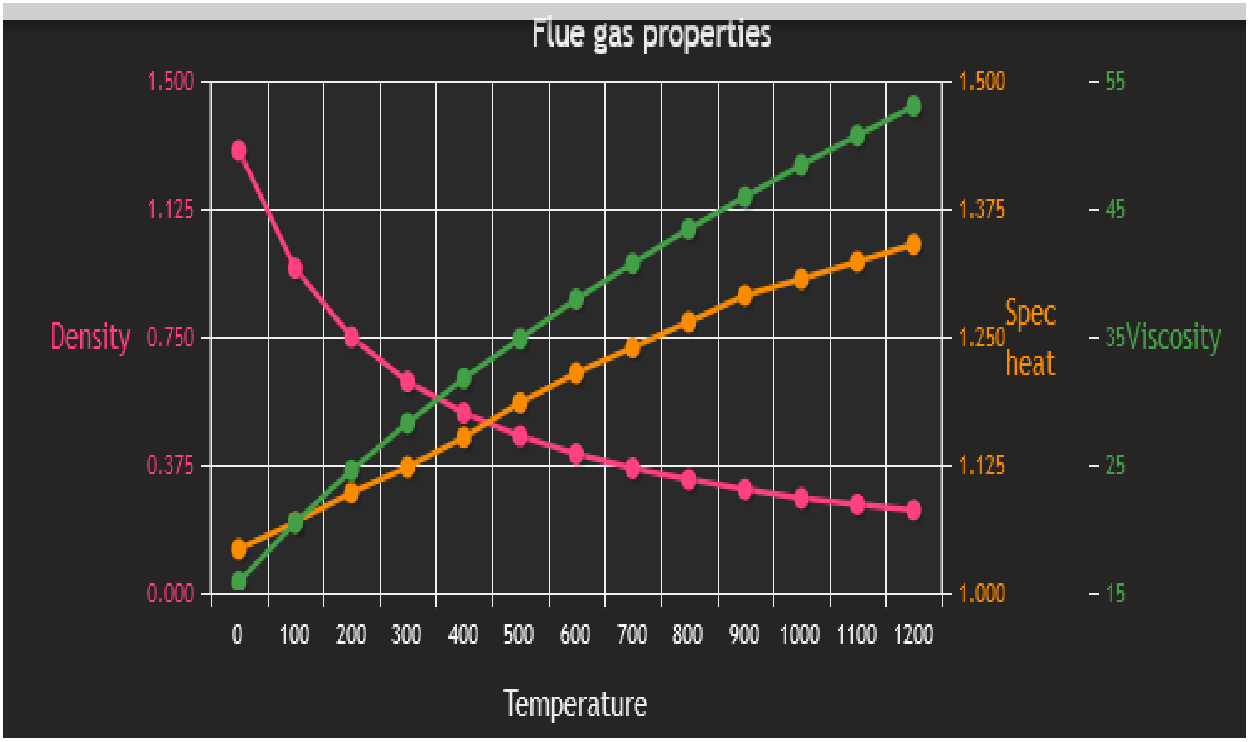
A typical finned type boiler economizer [11].

APPENDIX C



The boiler efficiency improvement for conventional economisers [5].

APPENDIX D



Changes in specific heat capacity, density and viscosity with flue gas temperatures [21].

Flue gas Property	Boiler 1	Boiler 2	Boiler 3	Boiler 4	Boiler 5
Density of flue gases at flue gas exit temperature (kg/m^3)- ρ_{fg}	0.7228kg/m^3	0.7126kg/m^3	0.7637kg/m^3	0.7583kg/m^3	0.752kg/m^3
Specific heat capacity of flue gases (kj/kgK)-c_{fg}	1.102 kj/kgK	1.104 kj/kgK	1.095 kj/kgK	1.096 kj/kgK	1.096 kj/kgK

Flue gas Property	Boiler 6	Boiler 7	Boiler 8	Boiler 9	Boiler10
Density of flue gases at flue gas exit temperature (kg/m^3)- ρ_{fg}	0.717 kg/m^3	0.7205 kg/m^3	0.7255 kg/m^3	0.8543 kg/m^3	0.8429 kg/m^3
Specific heat capacity of flue gases (kj/kgK)-c_{fg}	1.103 kj/kgK	1.102 kj/kgK	1.101 kj/kgK	1.082 kj/kgK	1.083 kj/kgK

APPENDIX E

	Flue gas density at temperature,			Specific heat capacity density at temperature,			Average specific heat capacity	Average Flue gas density
	T1	T2	T3	T1	T2	T3		
Experiment 1a	0.539	0.758	0.905	1.147	1.096	1.103	1.115	0.734
Experiment 1b	0.551	0.718	0.844	1.143	1.103	1.112	1.119	0.704
Experiment 2a	0.534	0.645	0.778	1.148	1.109	1.122	1.126	0.652
Experiment 2b	0.595	0.716	0.816	1.129	1.103	1.116	1.116	0.709
Experiment 3a	0.56	0.739	0.851	1.141	1.099	1.111	1.117	0.715
Experiment 3b	0.58	0.783	0.868	1.134	1.121	1.109	1.121	0.744
Experiment 4a	0.59	0.747	0.877	1.13	1.097	1.107	1.111	0.739
Experiment 4b	0.6	0.751	0.889	1.127	1.126	1.106	1.120	0.747
Experiment 5a	0.58	0.743	0.856	1.135	1.098	1.11	1.114	0.725
Experiment 5b	0.58	0.74	0.859	1.133	1.099	1.11	1.114	0.727
Experiment 6a	0.59	0.777	0.881	1.132	1.122	1.107	1.120	0.682
Experiment 6b	0.61	0.76	0.893	1.125	1.124	1.105	1.118	0.683

Modified stack flue gas specific heat capacity and density at varying flue gas temperatures.

Equipment used during the industrial survey,

(i) KANE 905 flue gas analyser [23] - flue gas temperature accuracy ±2.0℃ ±0.3% reading, excess air ±1.0%, Efficiency 1.0%.

(ii) 62 mini-max infrared thermometer [24]- Accuracy 2.0℃ at 0-10℃ ,3.0℃ at -30- -10℃ , 1℃ or 1% of 1.0℃ of reading , whichever is greater.

(iii) Lutron YK-2005TM [25] – Accuracy ±0.2%±0.5℃.

(iv) Lutron BTM-4298SD 12 Channel temperature recorder [26] - Accuracy ±0.4%±0.5℃.

(v) Prova AVM-01 anemometer [27] - Accuracy ±3%±0.3.

(vi) 1 liter calibrated beaker.