The investigation of the effect of atmospheric conditions on the temperature drop across heat treatment systems

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Declaration

I, Velaphi Absolom Phoswa, declare that:

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As the student's supervisor, I agree with the submission of this thesis.

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Abstract

The purpose of the present study is to investigate the effect of atmospheric conditions on the temperature drop across heat treatment systems (piping and vessels). Due to the economy and power energy consumption, it has become necessary for Isegen SA (Pty) Ltd to conduct a numerical analysis on heat transfer since they are unable to predict and quantify heat energy losses in the steam reticulation systems. These losses occur due to the fact that there is no proper methodology to effectively predict heat energy losses in heating processes. When a steam pipeline is at a higher temperature than the air surrounding it, heat will pass through the wall of the pipeline from the steam to the surrounding air. This heat loss may cause the temperature of the steam to fall and the boiler efficiency decreases because the boiler requires more gas to maintain the plant steam required.

Steam savings are very vital for Isegen SA (Pty) Ltd as most of the heating systems that they utilize are a direct steam injection. This means a large quantity of the sensible heat energy of the steam distributed into the factory is not recovered. Therefore, a numerical analysis was developed in order to investigate heat energy losses and steam consumption, while flowing through the insulated steam pipeline depending on the ambient and operational temperatures. The study proved that there is a heat energy loss as well as a temperature loss during the steam transportation through the insulated steam pipeline after conducting the numerical analysis. The thickness conduction resistance was obtained for thermal insulation complying with the required standards. The methodology involved the use of application of Excel spreadsheet to develop a theoretical model of the problem. The obtained model allowed us to get the required solution of the problem and calculate the heat loss for different thicknesses of the pipe's insulation. The application of Excel spreadsheet steady state thermal method was used for obtaining the value of heat loss of steam at a critical thickness of insulation.

After conducting the cost analysis, it has been found that Isegen SA (Pty) Ltd could save between one to two million Rands per year in fuel. The cost analysis conducted is based on the steam cost and heat energy losses. This study showed that steam waste is costly in both an environmental and financial sense. Therefore, this required prompt attention in order to ensure that the steam system is working at its optimum efficiency with minimal impact on the environment. This study was acknowledged by the Isegen SA (Pty) Ltd management, and a new technological process is to be implemented (see Appendix R for the relevant documents).

Preface

This dissertation contains eight chapters and ten appendixes. Chapter one starts with the introduction continues further to briefly discuss the history of Isegen SA (Pty) Ltd and the aim and objectives of the study. The study focused on the interaction between the various energy systems and the surrounding environment to predict the rate of heat loss and to develop strategies to minimize the heat losses. The problem statement and broad solution of the study were also included and discussed in this chapter.

Chapter two reviews the available works of literature on heat transfer and temperature drop correlations for the insulated steam pipeline. In this chapter, eight paragraphs are presented of the literature survey covering the convection heat transfer; theory background; heat transfer experimental methods; radiation modelling; heat energy transfer in the pipeline; optimum thermal design of steam pipelines; heat transfer engineering and summary of heat transfer theory. In chapter three, the significant background was given on the description of steam supply systems, boiler descriptions, the distribution and end-users, condensate return and how system performance was determined. The chapter continued further to discuss the importance of steam quality and best practice guidelines. This chapter provides an understanding of the steam distribution and its uses in the manufacturing process. In order to achieve the main objectives of the study, one must understand the manufacturing process well and the required operational characteristics of the steam.

To be able to use the simulations with confidence, verification and validation are necessary. Thus, is the subject of chapter four. This chapter investigates heat energy losses and steam consumption while flowing through the insulated long-distance pipeline. This study aims to present a basic application of heat energy required and heat losses during the steam supply to the process plant. The numerical analysis was conducted to investigate the heat energy losses and steam consumption while flowing through the insulated pipeline from the ambient temperature to the operational temperature. The results of these developments and user aspects regarding simulation results recovery, and exemplifies the need for the tools presented in this thesis by showing their application both in modelling orientated context and in theoretical calculations engineering context. The calculated heat transfer coefficient and heat loss are compared to the existing models. The company needed to use appropriate methods to reduce the heat loss, taking into account all the parameters that impact the energy cost.

Chapter five is about the experimental study work that was conducted during the research. The experimentation purpose was to discover new unknown phenomenal or for validation purpose, and simulation to understand interactions of the known components of a system. In the current context, modelling and simulation are thus used for predictions to assist in solving the real-world problem by taking into account all the parameters that impact the energy cost fuel, condensation, power generation. It will eventually reduce the fuel costs, the emission surcharges as well as maximize the process efficiency. A number of experiments have been conducted under similar conditions to obtain the average and reliable results. This elaborated further in this chapter, also identified those parts of this existing platform that needed further developments because of the subject of the present study. The results of these developments are described in chapter five which concerns the technical aspects of condensate removal, along with suggestions on how to improve the current work practice at Isegen SA. Insulating the condensate lines reduces heat loss from the water returning to the boiler and savings in fuel consumption. Further, it reduces radiation energy transfer and thus contributes to a drop in the surrounding temperature in the workplace.

In chapter six cost analysis of heat energy loss and production lost were presented. The cost analysis was conducted based on the production lost due to inefficient operation of the steam trace lines which use the steam supplied by the steam system and heat energy losses. The study proved that it is necessary for Isegen to implement this study to save fuel consumption and production lost and also steam usage.

Finally, chapter seven described conclusions that may be drawn from the present work and indicated possible directions for future research.

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NOMENCLATURE

$\dot{Q}$	Heat transfer rate	[kW]
h	Convection Heat transfer coefficient	[kW/m^2K]
A	Surface area	[m^2]
t_1	Surface temperature	[°C]
t_2	Surroundings temperature	[°C]
$\dot{m}_{sn}$	Flow rate of in-coming steam to end user No	[t/h]
h_{sn}	Enthalpy of in-coming steam	[kJ/kg]
$\dot{m}_{cn}$	Flow rate of out-coming condensation from end-user No	[t/h]
t_{cn}	Temperature of out-coming condensation	[°C]
$\dot{m}_s$	Mass flow rate of steam	[t/h]
h_s	Steam enthalpy	[kJ/kg]
$\dot{m}_{CR}$	Mass flow rate of condensation	[kg/s]
t_{CR}	Temperature of condensation	[°C]
$\dot{m}_{MU}$	Mass flow rate of make-up water	[kg/s]
GCV	Gross Calorific Value of fuel used for running the boiler	[kJ/kg]
$\dot{M}_F$	Flow rate of fuel	[t/h]
λ	Thermal conductivity	[W/mK]
δ_A	Thickness of the stagnant fluid A film	[mm]
δ_B	Thickness of the stagnant fluid B film	[mm]
U	Heat transfer coefficient	[W/m^2K]
V	Potential difference	[V]

I	Current	[A]
R	Thermal resistance	[K/W]
r_M	Mean radius	[mm]
A_M	Mean area	[m^2]
ε	Emissivity	[−]
x	Thickness	[mm]
σ	Stephan-Boltzmann constant	[$W/m^2\ K^4$]
η_b	Boiler efficiency	[%]
C_p	Specific heat capacity	[J/kg K]
G	Specific Gibbs free energy	[J/kg]
E	Energy	[J]
ρ	Density	[kg/m^3]
η	Boiler efficiency	[−]
NDE	Non-Destructive Examination	[−]
CTL	Coal-to-Liquid	[−]
LNG	Liquefied Natural Gas	[−]
Re	Reynolds number	[−]
$\dot{m}$	Boiler steam mass flow rate	[kg/s]
d_i	Pipe dimeter	[m]
μ_{steam}	Dynamic viscosity of steam	[kg/ms]
h_i	Inside convection heat transfer coefficient	[W/m^2 K]

C	Constant	$[-]$
n	Constant	$[-]$
λ_{steam}	Thermal conductivity	[W/mK]
d_i	Pipe internal diameter	$[m]$
P_r	Prandtl number	$[-]$
ν	Kinematic viscosity	$[m^2/s]$
ρ	Density	[kg/m^3]
D	Pipe outside diameter	[m]

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Chapter One

1. Introduction

In this chapter, the theory of steam plant technologies and their operating principles, as well as a description of the heat and heat transfer in their steam pipelines, is introduced. Steam has come a long way from its traditional associations with locomotives and the industrial revolution. Today, Steam is an integral part of modern technology. Steam has a huge role in our food, textile, chemical, medical, power, heating, and transport industries. They could not exist or perform as they do without it. Steam provides a means of transporting controllable amounts of energy from a central, often automated, boiler house, where it can be efficiently and economically generated and transported to the point of use. Therefore, as steam moves around a plant, it can equally be considered to be the transport and provision of energy. For many reasons, steam is one of the most widely used substances for conveying heat energy. Its use is popular throughout the industry for a broad range of tasks from mechanical power production to space heating and process applications.

The use of thermal energy typically required as a heating source for process fluid heat exchangers, reboilers, reactors, combustion air preheaters, and other types of heat transfer equipment. Saturated steam is often used as a source of thermal energy. The pipeline plays a role in conveying steam from the boiler to the process continuously. Since pipeline systems are complex and the ambient environment is highly variable, quickly determining the heat loss of a pipeline system is a difficult problem for engineers and pipe network designers. Yang (2009) stated that, firstly, the pipeline structure and the mass flow distribution influence significantly on the convection heat transfer inside the pipeline. Secondly, the pipe material, the cladding material, and the insulation have great effects on the heat conduction inside the pipe wall. Thirdly, the variable environmental parameters are challenges to the analysis of convection heat transfer outside the pipeline. Therefore, the heat transfer through the pipeline system is an interesting and also challenging situation.

Water can exist in the form of a solid, liquid, or gas. The focus will be largely on the liquid and gas phases and changes that occur during the transition between these two. Steam is the vaporized state of water which contains heat energy intended for transfer into a variety of processes such as air heating and vaporizing liquids in refining processes.

In the early days reciprocating steam engines and most locomotives were designed to operate on an open cycle where the exhaust steam was discharged to the atmosphere. This necessitated an adequate supply of fresh water. Impurities in the water accumulated in the boiler during the steam generating process and some form of water treatment in the boiler or prior to the water entering the boiler were required. If, however, the exhaust steam was condensed and reused, a substantial benefit was obtained from the expansion of the steam down to vacuum conditions in a condenser leading to improved efficiency. This cycle of boiling, expansion, condensation, and return to the boiler is commonly known as the Rankine Cycle (Hung *et al*. 1997).

Steam generators, typically boilers, generally operate within such a thermodynamic cycle, with work produced by the expansion of steam from a high pressure to low pressure. After condensation, some work is required to pump the water back into the boiler. The work required in pumping the liquid against the pressure difference is considerably less than the work produced by the steam in expanding across the same pressure difference. The prime energy input to the cycle is that required to generate steam from water. Heat energy is added to the boiler and heat is rejected when the steam is condensed to water.

Isegen SA (Pty) Ltd is the sole manufacturer of Food Acidulants, Phthalic and Maleic Anhydrides (PA and MA), and Plasticisers in South Africa. Isegen SA (Pty) Ltd are unable to predict and quantify heat energy losses in the steam reticulation systems. It is a necessity to develop a heat loss evaluation system that will predict and quantify heat energy losses for various atmospheric weather conditions and insulation inefficiencies. The methodology that will be conducted uses the numerical heat transfer analysis (force-free convections, enhanced convective cooling and heat flow resistance via heat transfer mechanisms).

1.1 Aim and Objective

In particular, research is focused on the interaction between the various energy systems and the surrounding environment, in order to predict the rate of heat loss and to develop strategies to minimize them.

The main objectives of the thesis are summarized below:

- To develop a heat transfer evaluation system that will predict and quantify heat losses for various conditions (weather change and insulation inefficiency).
- Develop techniques to determine heat losses in a steam pipeline in order to predict and minimize heat losses.

1.2 Company History

Isegen SA (Pty) Ltd is the sole manufacturer of Food Acidulants, Phthalic and Maleic Anhydrides (PA and MA, respectively), as well as Plasticisers in South Africa. Isegen SA's operations extend over three manufacturing sites – two of which are located in Durban (more specifically: Isipingo and Umgeni) and the third in Germiston near Johannesburg. The Head Office is on location at the Isipingo site which is conveniently situated next to the South Durban petrochemical complex. It is close to the Durban export harbour, which is the largest in Africa. Isegen SA (Pty) Ltd has a long history in the manufacture and marketing of Food Acids dating back to 1974. The Food Acid plant uses in-house MA which is derived from butane supplied by the refinery situated next door to produce Malic Acid and Fumaric Acid and is now the most integrated Food Acids manufacturer anywhere in the world. This enables the company to benefit from many synergies by being on the same site as the MA plant; such as the use of steam generated from the exothermic oxidation reaction of MA to drive the Food Acidulants plant. In addition, steam is also supplied by the PA plant. A range of Food Acidulants are manufactured, most of which have unique characteristics or properties. Isegen SA (Pty) Ltd is the only company known worldwide to have perfected the granulation of these acidulants. Some of these products have been patented and trademarked around the world. The manufacture of Plasticisers in South Africa dates back to the 1950's. Esters produced from PA and Adipic acid, are supplied to the South African PVC Industry. Recent initiatives have been undertaken to ensure that the plants are world-class with respect to quality. The manufacture of PA in South Africa, dates back to the 1950's, and the manufacture of MA was started in the 1980's. PA and MA are supplied to the Alkyd Resin and Unsaturated Polyester Resin Markets in South Africa. Recent initiatives have been undertaken to expand the capacity of the plants and ensure that the plants are world-class, with respect to quality and raw material conversion.

1.2.1 General

The Company is accredited with both ISO 14000 and ISO 9001:2008. Furthermore, the Company subscribes to the Black Empowerment and the Employment Equity initiatives of the South African government. The company has been fully accepted as a member of the Proudly South African campaign. ISO 22000:2005 has been implemented in the Food Acids plant. The food acids comply with the FCC6, National Formulary, and BP specifications and comply with Kosher and with Halaal certification requirements. World-class manufacturing procedures are employed and stringent Environment Health and Safety protocols are adhered to at Isegen SA (Pty) Ltd, where people, customers, suppliers and the environment are treated as paramount. Isegen SA (Pty) Ltd has already achieved a number of world firsts, such as the only commercial production of granular Malic Acid globally.

1.3 Problem Statement

Due to the economy and power energy consumption, it has become necessary for Isegen SA (Pty) Ltd to conduct a numerical analysis on heat transfer since they are unable to predict and quantify heat energy losses in the steam reticulation systems. These losses occur due to the fact that there is no proper methodology to effectively predict heat energy losses in heating processes. Therefore, when a steam pipeline is at a higher temperature than the air surrounding it, heat will pass through the wall of the pipeline from the steam to the surrounding air, this heat loss may cause the temperature of the steam to fall. According to Isegen SA (Pty) Ltd, it is estimated that approximately 20% of heat energy is lost in every heat transfer system. This is very energy inefficient and costly to the business due to the lost production of steam as there is no proven specific methodology to quantify the efficiency.

The aim of this study is to investigate the effect of atmospheric conditions on the temperature drop across heat treatment systems (piping and vessels). Isegen SA (Pty) Ltd is a chemical industry that manufactures different types of acids. The factory uses water-tube boilers to generate steam that is utilized to warm up various equipment and to produce products such as Malic acid. However, the current system is not energy efficient because of the severe heat energy losses that occur across the steam pipeline. These heat losses occur due to the inefficient insulation and varying atmospheric conditions. Furthermore, the pipelines that supply the product experience blockages, since the steam tracing system is not effective. Isegen SA (Pty) Ltd is currently unable to predict and quantify the heat loss across the steam reticulation system. Therefore, the boiler input (energy) does not account for the heat losses that occur across the steam reticulation system.

1.4 Broad Solution

The objectives of this study are:

1.4.1 To develop a heat transfer evaluation system that will predict and quantify heat losses for various conditions (weather change and insulation inefficiency).

The numerical analysis was carried out to investigate heat energy losses and the steam consumption while flowing through the insulated long-distance pipeline from the ambient temperature to the operational temperature. Accurately determining steam costs is important for monitoring and managing energy use in a plant, for evaluating proposed design changes to the generation/distribution process itself, and for continuing to identify competitive advantages through plant efficiency improvements. Steam costs are highly dependent on the path that steam follows in the generation and distribution system. Based on the numerical predictions, a significant difference was found between the analysed required steam consumption and heat energy losses while flowing through the insulated long-distance pipeline.

The effect of thermal insulation conditions on the heat losses taking place at Isegen SA (Pty) Ltd steam pipeline network was studied. This was done by quantifying the magnitude of such losses and their comparison with the overall energy losses taking place during the steam transport from the boiler house to the process area. The appropriate analysis for the experimental measurements led to obtaining the condensation heat transfer coefficient and heat losses in a thermal system's different operating parameters. The study was based on the results of a field inventory of the conditions of the pipelines thermal insulation.

According to the field inventory, only $18 - 20\%$ of the pipeline network's length has proper insulation that is of good quality (from regular to very bad condition); they are, however, responsible for nearly half of the heat losses. The improvement of the efficiency of the steam transportation system (i.e. the reduction of heat losses) is an area that needs attention.

1.4.2 Develop techniques to determine heat losses in a steam pipeline in order to predict and minimize heat losses.

Based on heat transfer theory, a theoretical model was established using the application of Excel spreadsheet. The model was developed for assessment of the heat loss of pipeline systems. The software can obtain results with acceptable accuracy as the comparison between software and numerical evaluation shows reasonable agreement. It also has a user-friendly interface, which ensures the easy use for both engineers and other competent personnel who do not have a thermodynamics background. In summary, the model not only provides an easy way for fast assessment of the heat loss, but also offers a useful tool for the optimal design and energy saving analysis of pipeline systems.

Chapter Two

2. Literature Review

This chapter provides a critical review of the current knowledge published including substantive findings through summary, classification, and comparison of prior research studies and theoretical articles. The main purpose of this literature review is to develop an understanding of the topic at hand through previous work done on the heating treatment of metals. The scope of this chapter includes convection heat transfer, heat treatment and experimental methods on metals, radiation model as well as the heat energy transfer in the pipeline.

2.1 Convection Heat Transfer

The numerical solution of heat transfers and other related processes were made possible when the laws governing these processes had been expressed in a mathematical form. This is generally in terms of differential equations. In 1701 a review of the background of "Newton's Law of Cooling" led to the conclusion that it is appropriate to credit Newton with the concept of the convective heat transfer coefficient (HTC). Furthermore, because his early experiments with heat were eluded to in the Principia, the 300^{th} anniversary of this volume should honour him as a pioneer in heat transfer as well as in solid and fluid mechanics. Isaac Newton is widely credited with the convective heat transfer coefficient equation (Cheng and Schulenberg, 2001).

$$\dot{Q} = hA(t_2 - t_1) \tag{1.1}$$

2.2 Theory background

The process of heat exchange between two fluids that are at different temperatures and separated by a solid wall occurs in many engineering applications. The device used to implement this exchange is called a heat exchanger, and specific applications may be found in the sounding area of heating and air conditioning, power production, waste heat recovery, and chemical processing. According to Kraus (2011), the flow of heat from the fluid through a solid wall to another fluid is also often encountered in chemical engineering practice. The heat transfer may be latent heat accompanying the phase changes such as condensation or vaporization, or it may be sensible heat coming from increasing or decreasing the temperature of the fluid without phase change. Heat transfer is the movement of energy due to a temperature difference. There are three physical mechanisms of heat transfer; conduction, convection, radiation. The three heat transfer mechanisms were described in chapter two. All three modes may occur simultaneously in problems of practical importance.

2.3 Heat Transfer Experimental Methods

According to Kulkarni and Radhakrishna (2005), the students from the Department of Mechanics of B.M.S. College of Engineering used two methods to measure the HTC at the metal-mold interface during the casting process. The first method measured the size of the gap formed between the metal and the mold during the casting process and estimated the value of the HTC based on the gap size. The second method measured the temperature of the metal and the mold at certain surfaces, and the reverse method was used to derive the HTC at the gap. A procedure was also developed to use the temperature measurement data in order to obtain the HTC as a function of the casting temperature near the interference.

Garcia (2005) attempted to quantify the transient interfacial metal mold heat transfer coefficient by emphasizing different factors affecting the heat flow across such an interface during a solidification stage. These are the thermophysical properties of the contacting materials: the casting geometry, the orientation of the casting-mold interface with respect to the gravity (contact pressure), the mold temperature, the pouring temperature, the roughness of mold contacting surface and the mold coatings. It was proven that heat flow is controlled by the thermal resistance at the curing mold interface during rubber curing and casting in the metallic molds. Thus, the heat transfer coefficient at the metal mold interface has a predominant effect on the rate of heat transfer. The effect of pressure on molten metal will only affect the heat transfer rate at least at the initial steps of solidification in some processes such as low pressure and die casting. According to Guerlac and Henry (1980), there are various theories on the nature of heat that were developed by Scottish chemist Joseph Black in 1761. These include discovering that ice can absorb heat without changing its temperature when melting. This led Black into concluding that the heat must have combined with the ice particles and become latent, henceforth the theory of latent heat energy was formulated. According to Rumford (1804), a cannon manufacturer, Sir Benjamin Thompson, demonstrated through the use of friction that it was possible to convert work to heat. He designed a specially shaped cannon barrel that was thoroughly insulated against heat loss. Furthermore, he used a dull drill bit to replace the sharp boring tool and immersed the front part of the gun in a water vessel. Thereafter he heated up the water in the vessel from ambient to a boiling point within 2.5 hours without the use of fire.

Eastop (1993) stated that using a hot burner that heats energy can be conduction transferred from one molecule to another molecule within a substance touching a hot burner by conduction. He also stated that heat energy can be conventionally transferred by the mass movement of liquid or gas by convection, and that heat energy can be transferred by waves through radiation.

When heat is transferred the thermal energy always moves from warmer to cooler objects. The heat never flows from a cooler object to a warmer object. The warmer object loses the thermal energy and becomes cooler as the cooler object gains the thermal energy and becomes warmer. The heat energy conversion, in general, and specific heat capacity, in particular, has been the main topic in energy saving research for the last 30 years. Energy enhancement techniques studies were successfully conducted in the past. The thermal energy is dependent on the total amount of molecules in motion. Therefore, the thermal energy absorbed increases as the volume of the component under warm-up increases. The thermal heat energy is directly proportional to the specific heat capacity of a substance that is required to raise the temperature of a given mass of the substance by some amount. The heat capacity varies from substance to substance. According to Zalba and Marin (2003), it was proven that the heat is thermal energy that is transferred from one object to another when the objects are at different temperatures. The amount of heat that is transferred when two objects are brought into contact depends on the difference in the temperature between the objects. The heat transfer continues until both objects are at the same temperature. Heat transfer research and studies were done successfully in the past through experiments and research work done by various researchers. Cheng and Schulenberg (2001) proved that heat transfer at high pressures is mainly characterized by the thermal physical properties which very strongly near the pseudo-critical line. The steam heating efficiency of rapid heat cycle molding (RHCM) can be effectively improved by increasing the thermal conductivity of the cavity or core material. This situation is diametrically opposite for an electric heating RHCM (Wang and Zhao, 2010). The thermal response of a new electric heating mold was also simulated. The simulation results showed that the cooling plate can significantly improve the cooling and heating efficiency.

2.4 Radiation Modelling

According to Gokhale *et al*. (2003), a simulation of a ceramic furnace model with pottery kiln-walls, the pottery ware and the gases in the kiln were all modelled as one-dimensional entities and the heat transfer through the kiln-walls was analysed using advanced computational fluid dynamics (CFD) software. It was proven that heat transfer through honeycomb panels is non-isotropic and difficult to predict using CFD if the effect of the cover faces is taken aside, and convection and radiation within the honeycomb cells can be neglected in comparison with conduction along the ribbons. Martinez (1995) stated a numerical-mathematical model detailing the radiant exchange between surfaces. A generalized multiple-surface shape model was constructed, developed, and tested to a satisfactory degree of complexity, at which point it was incorporated into CFD. According to Reynolds (2005), the predictions by the model were compared to selected results from pilot-scale DC arc furnace tests run at Mintek to evaluate the significance of radiation heat transfer on the overall energy loss from the freeboard regions of such furnaces.

2.5 Heat Energy Transfer in Pipeline

Pettrukhov (1970) stated that heat energy transfer in pipelines has been evaluated for almost 60 years. Nusselt's paper, published in 1910, was the first paper to analyse at a scientific level. Later, during subsequent years, different investigators and scientists had studied flow through pipes for various fluids. As a result of this, the relation of Nusselt number with Reynold's number was formulated.

Diessler and Eian (1952) considered the comparative approximations as similar to Graetz' and gave the analytical solution for the problem faced in fully developed pipe flow in which fluid properties vary along the radius. Adekunle *et al*. (2014) showed that heat transfer through the pipe is dependent upon the wall thickness of the boundaries. This was shown by maintaining the integrity of the specifications. Wan Kai and Wang Ping (2013) stated that the computational fluid dynamics CFD simulation of small and medium gauge $90°$ circulars bend. They used a standard $k - \varepsilon$ model with FLUENT software on a large diameter CFD numerical simulation of air flow in a $90°$ twisted tube, with three-dimensional stress field and velocity field in the pipe. Exploring the non-change law of fully developed pipe flow through a CFD numerical simulation on large-diameter flue gas lays the foundation for analysis and numerical simulation of the non-circular cross-section.

2.6 Optimum Thermal Design of Steam Pipelines

Abdallah and Krameldin, 1999 stated that the majority portion of electric power generated by conventional and nuclear is produced by steam all over the world. Also, steam is used extensively in many chemical petroleum's, food, water desalination, and many other industries. For the last fifty years, little improvements have been made in the thermal efficiency of steam boilers. The major developments have been carried out in the direction of maintaining the efficiency of low-grade fuel and reducing labour and maintenance charges. The annual cost of fuel nuclear and non-nuclear was often greater than the combined cost of other expenses of steam power plants. Hence, a great amount of money could be saved by designing steam pipelines in such a way that the total annual cost of pipes, cost of insulation material, cost of burned fuel and cost of maintenance was a minimum value. However, the insulation was used for one or more of the following reasons: to conserve heat, maintain desired temperature conditions, retard changes in temperature, prevent inside steam condensation, provide fire protection, and provide safety from burn hazards. It was therefore essential that the system performs as designed. Thus, the performance properties, and especially the factors that affect them, were an important consideration to economic installation, lifetime, and thermal efficiency and environmental pollution by-products of combustion gases.

The variety of insulation types and forms have properties that vary over significance ranges, and thus the uses of a material can also vary widely. The selection of insulation is determined by considering several properties-related factors. The cost of insulation material constitutes a large portion of the total cost especially for large-diameter high-temperature steam pipelines, where the insulation material price increases with its thickness. Selecting a suitable type of insulation was also an important problem in energy conservation. The final selection of the most suitable insulation was based on the available information concerning the properties relevant to the requirements. Generally, the major parameters that determine the choice of the insulation are material cost, thermal properties, mechanical properties, installation and cost, and lifetime.

Sloane (992) stated that the problem of thermal performance and cost evaluation of insulation has been studied extensively by many investigators for purpose of optimizing the insulation material thickness from hot surfaces as one of the methods adopted for energy conservation. The problem was vital when the working fluids have high temperatures and the fluid transmitting pipelines have a long span and large bores where the heat loss was too much. The study an economic balance was made between the capital charges on pipeline insulation material which increases with the insulation thickness and between the monetary loss of fuel in the form of heat loss which proportional inversely to the insulation thickness and directly its thermal conductivity.

2.7 Heat Transfer Engineering

Campo (2010) his research interests are in heat transfer, fluid dynamics, and energy conversion. Campo previously conducted the study based on the quick algebraic estimate of the thickness of insulation for the design of process pipelines with allowable heat losses to ambient air. The underlying analysis in this article on thermal design rests on the powerful formulation of a 1-D extended lumped model with detailed physical insight for the three-participating media: the internal fluid, the insulation annulus, and the external air. The superiority of the 1-D extended lumped model for the 2-D differential model is self-evident because it admits analytic treatment continually from the beginning to the end. Consequently, with the 1-D extended lumped model, it is possible to obtain analytic expressions for the mean bulk temperature and the total heat transfer for bare pipes and insulated pipes under any cooling conditions. A step-by-step calculation procedure has been developed and tested for the analytic determination of the thickness of the insulation of pipes carrying high-temperature, viscous Newtonian fluids with either laminar or turbulent motion. The total heat release from the internal fluids to the external air at atmospheric temperature may take place by either forced or natural convection.

The outcome of the lumped modelling linked to an ordinary differential equation and an efficient fixed-point iteration technique leaves no doubt about their credentials as a top contender procedure for estimating the thickness of insulation in process pipes not only analytically, but more importantly, accurately and rapidly. In sum, it was expected that the proposed computational approach, united to the practical set of results, would provide useful tools for thermal design engineers in the future.

2.8 Summary of Heat Transfer Theory

Heat always flows from the source of higher energy to one of lesser energy. The greater the temperature difference the more rapid the energy flow. Temperature is a relative measurement of thermal pressure but it is not a unit of energy. Heat lost by one medium is always equal to the amount of heat gained by the other medium, minus losses in the transfer. Every material has its own unique properties that influence its ability to move heat energy.

Hence, it is a necessity to develop a heat loss evaluation system that will predict and quantify heat energy losses for various atmospheric weather conditions and insulation inefficiencies.

Chapter Three

3. Steam Distribution and Use to Manufacturing Process

This chapter provides an understanding of steam distribution and its uses in the manufacturing process. If an analysis showed that a particular device or program could increase the efficiency of a central station boiler by five percent, in most cases it would be carefully considered and exhaustively researched. An observed industrial phenomenon takes place when the same scenario occurs outside the powerhouse walls. The same efficiency improvement potential may not receive the same consideration, analysis or study. This is, of course, a generality but in many cases the statement is accurate. What makes up quality steam? Steam that is conditioned to maximize energy transfer. In order to conduct this experimental work, one must understand the manufacturing process well and the required operational characteristics of the steam. The following discusses some of the factors contributing to quality steam.

3.1 System Description

According to Zoran *et al*. (2008), steam systems generate and distribute thermal energy in the form of steam, which is then used for a variety of process applications. Deacon (2008) stated that, on average, $40\ \%$ of all the fuel burned in the industry is consumed in order to raise steam. Steam is used to heat raw materials and treat semi-finished products. It is also a source of power for equipment, as well as for heat and electricity generation. Many companies can improve steam system performance through the installation of more efficient system equipment and processes. The entire steam system must be considered in order to optimize energy use and achieve cost savings. A typical steam system, as seen in Figure 2.1, includes the following subsystems

- Boiler.
- Steam distribution system (including control valves, steam traps, insulation, etc.
- Steam end-users (including control systems, steam traps, insulation, etc.
- Condensation return system (including piping, storage tanks, insulation, pump, etc.
- Metering, monitoring and control systems.

The characteristics of each of the above-mentioned subsystems and their energy performances will be analyzed separately. For each subsystem, and the energy balances that define its energy efficiency need to be established. The measurement points for steam system monitoring and control are indicated in Figure 3.1. These are the minimum measurement requirements for practical operational performance monitoring and analysis of the steam system as a whole. All of these measuring points often do not exist within companies, but even if they do exist, it is advisable to check the calibration records and assure oneself about the accuracy of the instrumentation.

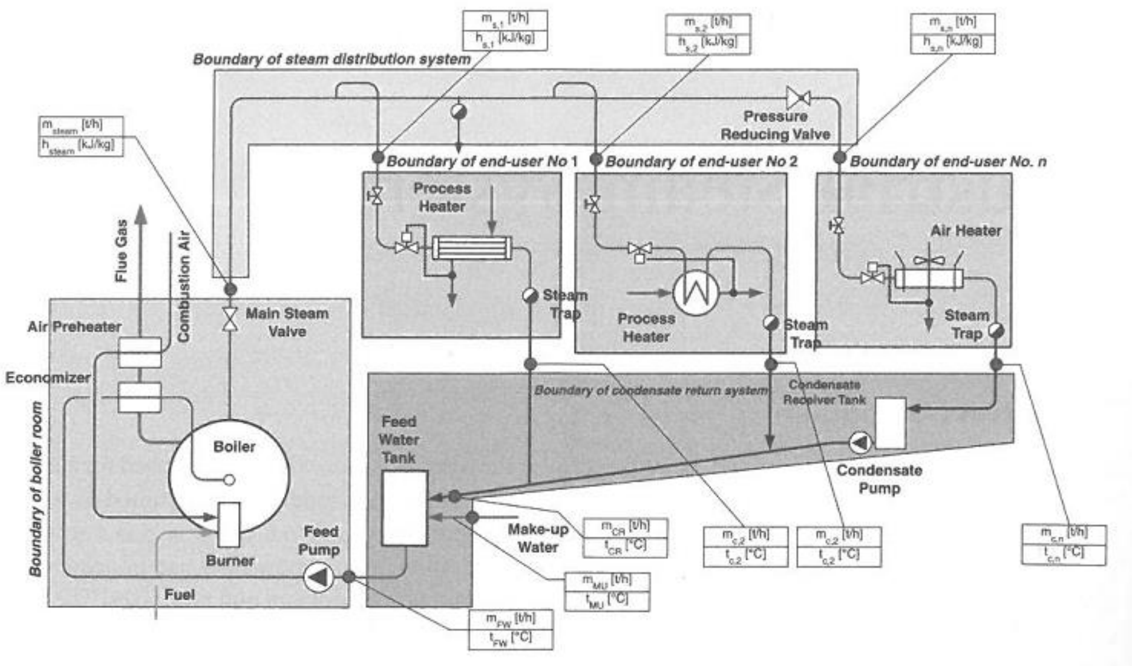


Figure 3.1: Overall Steam System Definition (Morvay and Gvozdenc, 1990).

3.1.1 Boiler Description

The boiler is a device that converts the chemical energy of fuel into useful heat output. Typical heat outputs are steam (saturated or superheated), hot water or thermal fluids like mineral oil. There are many different types of boilers but all of them can be classified into two basic groups:

- Water tube boilers where water is contained in tubes and flames as well as hot combustion gases pass around them.
- Fire-tube or shell boilers where combustion gases pass through a furnace tube and after that enter tube bundles immersed in water within the shell.

Isegen SA (Pty) Ltd. has a water tube boiler with an integrated super-heater that has been manufactured by John Thompson (Pty) Ltd for specific high-pressure steam process applications for both South Africa and export markets. It provides 8 t/h of saturated steam at 14.5 bar(g), a temperature of 198.3℃ and is a dual-burner gas-fired boiler manufactured in South Africa. It can also be oil-fired or dual-fuel fired. Also known as the THOMPSON MULTIPAC SH boiler, it is a high-efficiency, three-pass, wetback package boiler fitted with an integrated, low maintenance super-heater. The boiler is equipped with a fully automatic, modulating burner, complete with a forced draught fan and gas train. The burner fitted to the boiler in the front has been selected to fire coal, oven gas or diesel and has the facility for quick changeover from one fuel to another.

A large steam space and a mesh demister ensure a dryness fraction exceeding $99\ \%$ at the inlet to the super-heater while the boiler is operating at full load. According to John Thompson (Pty) Ltd. steam leaves the super-heater at $250°C \pm 5°C$ at full load. The boiler incorporates a convective super-heater in a counter flow arrangement located in the front smoke box between the exit of the second pass tubes and the entry to the third pass tubes. Steam from the boiler passes through a super-heater tube bank which raises the steam temperature to $55°C$ above that of the boiler, typically for use in turbines where condensation is undesirable. Final steam temperature can be adjusted with the aid of a bypass damper incorporated in the flue gas pass. A safety valve on the outlet relieves excess pressure. A vent valve is fitted to protect the super-heater against overheating during start-up. Due to the convective nature of the super-heater, the steam temperature will drop off as the load reduces. This effect can be counterbalanced by manually closing the bypass damper at lower loads. The bypass damper can also, if required, be equipped with an actuator and control loop to automatically control steam temperature to $\pm 5°C$ over a wide load range. The super-heater smoke box is fitted with easily removable doors to facilitate tube cleaning and access to the super-heater elements. The water level in the boiler is controlled by a modulating feed water level controller. This system incorporates audible alarms to warn the operator when the water in the boiler falls to a low level or rises to a high level. In addition to this, an entirely independent overriding float-operated switch is used to stop the burner if the water level in the boiler falls to a dangerously low level. A control panel housing the Burner Management System, control switches and warning lamps etc. are mounted on one side of the boiler. The scheme of a case study boiler house is presented in Figure 3.2 below

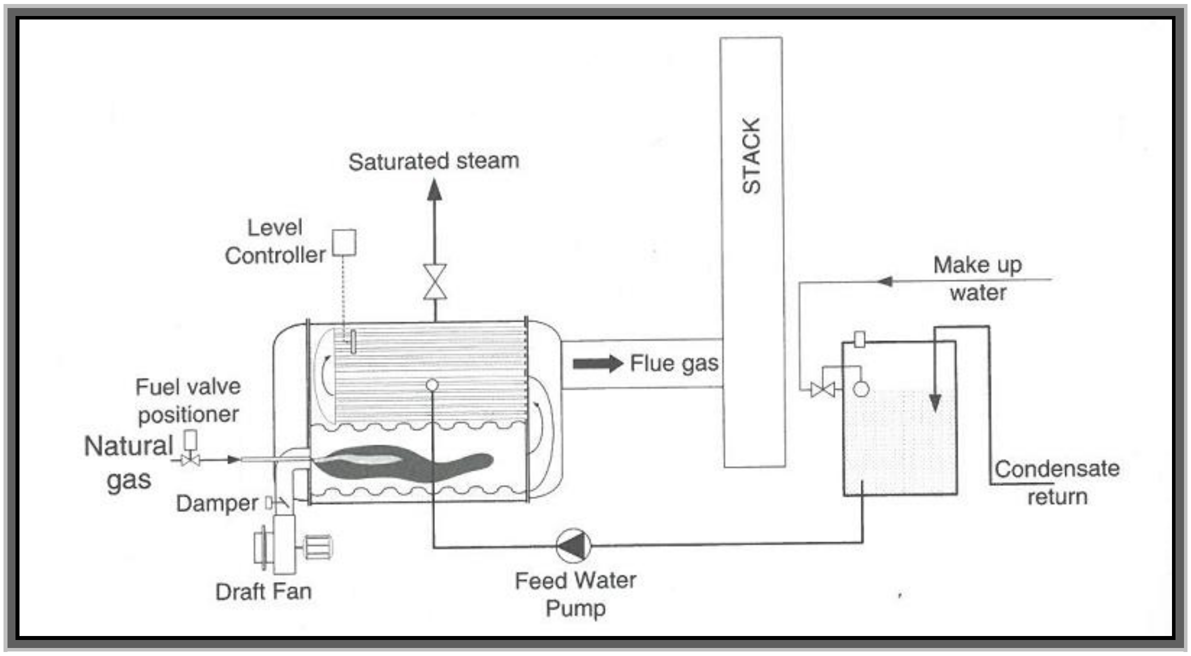


Figure 3.2: The scheme of a case study boiler (Morvay and Gvozdenc, 1990).

3.1.2 Steam Distribution System

This system delivers steam for end-users. Steam flow depends on the needs of end-users. Steam condensation occurs due to the inevitable cooling of steam in pipelines and it is necessary to remove the condensation from the pipeline at certain points. For that purpose, steam traps are used, not only in the steam distribution system but also at the end-users. The type and characteristics of steam traps can be found in the available literature or directly from the manufacturers. The insulation of the steam distribution system is very important for good performance of this subsystem and for the high efficiency of the steam system as a whole.

3.1.3 Steam end-users

There are a variety of steam end-users in the industry and each of them should be the subject of energy management in respect of monitoring and estimating their individual effectiveness and energy efficiency. We are not going to enter into a detailed analysis of various end-users. However, for the proper operation of any end-user, it is necessary to deliver the required steam flow at the corresponding pressure. This parameter is established by the manufacturer's instructions for operation, to use it at the designed capacity, and to set up and implement a proper maintenance plan.

Table 3.1: Boiler efficiency according to boiler type (based on GCV). Reproduced from: Good practice guide No. 30 (1993) energy efficiency operation of industrial boiler plant.

Boiler Type	Efficiency (%)
Condensing	88-92
High-Efficiency Modular	80-82
Shell Boiler – Hot Water	78-80
Shell Boiler – Steam	75-77
Reverse Flame	72-75
Cast Iron Sectional	68-71
Steam Generator	75-78
Water Tube with Economizer	75-78

3.1.4 Condensation Return System

One of the oldest measures for increasing energy efficiency in industrial steam systems is condensation recovery. In the past, this measure did not draw much attention as energy was relatively cheap and condensation recovery systems were comparatively expensive. In some process, steam is used directly in the process and it is then not possible to consider condensation recovery. It is the same when steam may be polluted by hazardous substances from the process and when, for safety reasons, condensation is discarded. However, even in such cases, it is possible and even desirable to utilize the condensation energy value by means of heat exchangers.

3.1.5 System Performance

Bailly and Haler (1990) mentioned that it is possible to define the energy efficiency of the subsystem shown in Figure 3.1. In this way, the following efficiencies (performance indicators) are obtained

For Boiler

$$\eta_B = \frac{m_{steam}\, X\, (h_{steam} - 4.21\, .\, t_{FW})}{M_{FUEL}.\, GCV} \tag{3.1}$$

For Steam Distribution System

$$\eta_{SD} = \frac{\sum_{n=1}^{N} m_{S,n}\, .\, h_{S,n}}{m_{steam}\, .\, h_{steam}} \tag{3.2}$$

For end-users System

$$E_{EU} = \sum_{n=1}^{N} \left(m_{s,n}\, .\, h_{s,n} - m_{c,n}\, .\, 4.21.\, t_{c,n}\right)(n = 1,\ 2,\ 3 \ldots \ldots N) \tag{3.3}$$

For Condensation Return System

$$\eta_{CR} = \frac{m_{CR}\, .\, 4.21.\, t_{CR}}{m_{MU}\, .\, 4.21.\, t_{MU} + \sum_{n=1}^{N} m_{c,n}\, .\, 4.21.\, t_{c,n}} \tag{3.4}$$

The overall system energy efficiency can be defined as the ratio of the energy delivered to and used by all end-users, and the energy supplied to the boiler.

$$\eta_{SS} = \frac{\text{Energy Used by End} - \text{Users}}{\text{Energy Supplied to System by Fuel}} = \frac{\mathrm{E_{EU}}}{\mathrm{E_{FUEL}}} \tag{3.5}$$

Finally, we can rewrite the equation for the calculation of steam system energy efficiency (Eq. 3.5) as follows

$$\eta_{ss} = \frac{\sum_{n=1}^{N} (m_{s,n}\, .\, h_{s,n} - m_{c,n}\, .\, 4.21\, .\, t_{FW})}{M_{FUEL}\, .\, GCV} \tag{3.6}$$

Where:

$m_{s,n}$ = flow rate of incoming steam to end user No. n, (t/h)

$h_{s,n}$= enthalpy of in-coming steam, (kJ/kg)

$m_{c,n}$ = flow rate of out-coming condensation from end-user No. n, (t/h)

$t_{c,n}$ = temperature of out-coming condensation, (^{0}C)

4.21 = isobaric specific heat of water, (kJ/kg.K)

m_s = mass flow rate of steam, (t/h)

h_s = steam enthalpy, (kJ/kg)

m_{CR} = mass flow rate of condensation, (t/h)

t_{CR} = temperature of condensation, (^{0}C)

m_{MU} = mass flow rate of make-up water, (t/h)

GCV = Gross Calorific Value of fuel used for running the boiler

M_{FUEL} = flow rate of fuel

3.1.6 Steam Pressure

Deacon (1991) stated that the steam pressure plays an important role when trying to maximize the energy transfer of steam. The higher the steam pressure, the higher the temperature and total energy content of a unit of steam. Latent heat or heat of vaporization is the energy released by steam and changes as pressure changes. Latent heat quantity increases as pressure decreases. To maximize latent heat transfer, steam should be used at as low a pressure as possible. Steam pressure also controls the saturated steam temperature. As pressure increases, so does temperature. Since temperature difference governs heat energy transfer, the higher the temperature, the easier it is to transfer heat. The ease of energy transfer can yield a smaller heat exchanger due to the improved heat transfer. The trade-off for the smaller heat exchanger could be that much more steam is consumed, due to the decrease in latent heat of the higher pressured steam. When steam is distributed through the piping system, steam pressure drops. Steam mains and steam branch lines are sized to distribute the steam without excessive pressure drops. When steam pressure drops, the total energy content of the steam also drops. To avoid this energy loss, steam mains and branches must be sized very carefully to avoid making the energy loss even higher than normal. In the system when pressure drops, so does temperature. This could slow heat transfer, creating a demand for more steam, which increases the pressure drop, wasting more total energy. In addition, velocity will increase, contributing to more erosion and noise within the piping.

3.1.7 Clean Steam

Steam that is clean goes a long way toward improving energy transfer and reducing system maintenance. From an energy aspect, the biggest impact will be seen in instrumentation and control devices. Clogged control parts can lead to poor pressure and temperature control. In the previous section, it was shown how pressure can impact energy transfer. In control devices, particularly pressure reducing stations and control valves, clean steam is far less likely to cause leaks at the valve seating surfaces. Valve leaks are likely to cause steam loss, poor pressure control and result in energy loss. Steam flow measurement, pressure measurement, and temperature measurement will most likely produce accurate data on clean steam. When not accurate, poor control decisions can result in resultant energy loss.

3.1.8 Heat Transfer Potential

Pressure has been discussed as having an impact on heat transfer potential through its impact on steam flow, steam temperature, and heat energy release. Accumulations of air and non-condensable gases in the steam system can also limit steam flow, steam temperature, and heat energy release. Air is present in the system on start-up and is also introduced through vacuum breakers on temperature-controlled processes. Non-condensable gases are liberated in the boiler. Carbon dioxide and oxygen are dissolved in the boiler feed-water as carbonates and bicarbonates. These non-condensable gases, when released, flow with the steam and can create energy problems. The gases cause a temperature reduction by contributing to total system pressure. Dalton’s Law of Partial Pressure states that the pressure of the mixture of steam and other gases is equal to the sum of the partial pressures. This effectively reduces steam pressure and the resultant temperature. As steam temperature drops, so does the energy transfer. When steam condenses, the other gases are forced to the heat exchanger surface, forming a stagnant film. As we all know, air is a good insulator. Seen in Figure 3.3 as a comparison, a film of air thick has the same resistance to heat transfer as a film of water thick, or steel thick, or copper thick. So, besides reducing temperature, the gases can act as an effective insulating barrier to energy transfer.

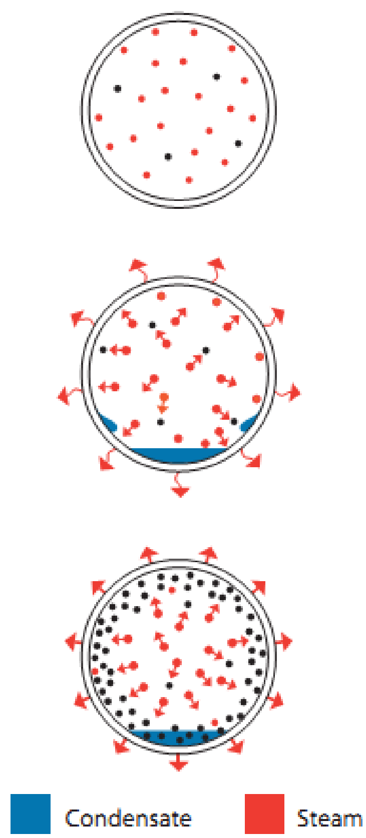


Figure 3.3: Plating out Non-Condensable cause insulating effect, Deacon (1991).

The gases also take up volume and don't condense into liquid (hence the term non-condensable). If allowed to accumulate for long periods, they take up enough volume to effectively block steam flow and all energy transfer. When condensation flow is also blocked, dangerous water hammer can also occur. When allowed to cool in the presence of condensation, carbon dioxide can combine with the water to form carbonic acid. Since gas accumulation causes a temperature drop, this acid energy formation is highly probable. The corrosion of iron forms a soluble bicarbonate which leaves no protective coating on the metal. If oxygen is also present, rust forms and CO_2 is released, which is now free to cause more corrosion. Once the gases become dissolved, they could be drained and removed, but they corrode on the way. This corrosion is free to cause steam leaks which are an energy loss and thus the corrosion is a maintenance and safety problem. In summary, steam energy transfer can be affected by many factors. The traditional definition of steam quality covers one such factor; the amount of water entrained in the steam. Other factors include steam pressure, both in the distribution system and at delivery, steam cleanliness, and the amount of air or non-condensable gases present in the heat transfer equipment.

3.1.9 Measuring Steam Quality

According to American Society of Mechanical Engineers Standard (ASME) the measurement of traditional steam quality is possible with standardized test instrumentation. Measurements are essential to the safe, accurate and reliable operation of the steam plant. Quality must be measured to qualify flow-meter readings for steam billing purposes, to check boiler treatment practices, and as a troubleshooting tool. Trends in boiler and steam generating equipment design have increased heat transfer surface areas in more compact packages. Since more steam is coming from smaller units, there is a greater likelihood that the equipment will carry over (throw some water out with the steam). As steam travels through the distribution system, the piping radiates heat and some condensation forms. Boiler treatment with chemicals can cause upset conditions and priming (boiler overflow). This introduces water into the boiler header and distribution piping. Chemical treatment can also affect the specific boiling action within the boiler and also affects carryover. Steam quality must be measured to assure and control the safe, reliable operation of the boiler or steam generating equipment. Field testing methods and procedures for steam quality measurements are described in the ASME Performance Test Code 19.11 Steam and Water Purity in the Power Cycle. One of the methods, the throttling calorimeter, is capable of measuring steam quality directly (see Figure 3.4). The other methods include

- Ion exchange.
- Conductivity (electrical).
- Sodium tracer flame photometry.
- Specific ion electrodes.

These methods determine the solids content of steam, including the solids carried over by water droplets. The throttling calorimeter is the only direct way to determine steam quality. This method is most accurate below $4136\ \mathrm{kPa}$ and where moisture content is above $0.5\ \%$. The calorimeter is a simple, easy to use instrument. When properly insulated and installed, it is very accurate. The calorimeter works on the principle that when steam expands without doing work, the heat content does not change. If the steam is dry and is discharged to the atmosphere, it will become superheated due to the pressure drop. If there is any moisture present, the temperature of the atmospheric discharge will be reduced. Measuring the temperature and comparing to the maximum possible temperature will indicate the steam.

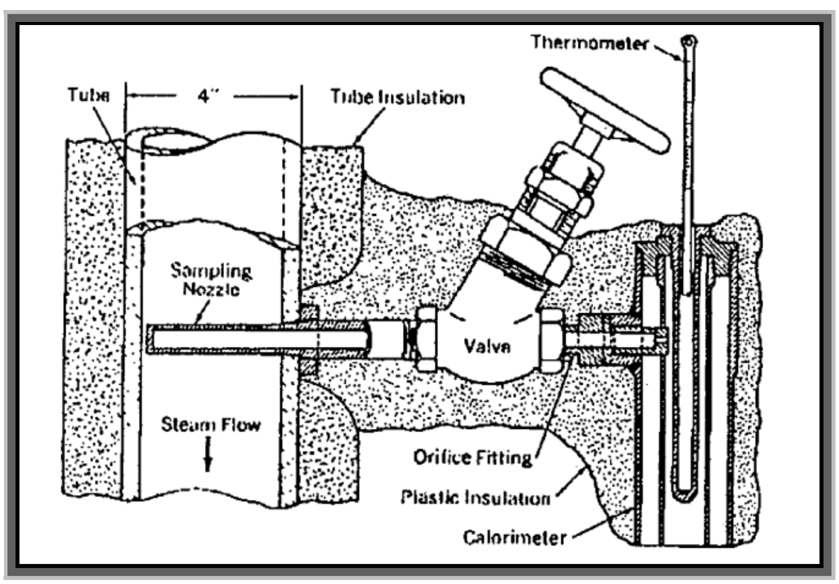


Figure 3.4: Throttling calorimeter shown installed in a steam pipe, Deacon (1991).

The other methods, by measuring solids, give an indirect measure of steam quality. The solids in the steam are sampled and counted and then compared to the solids content of the water in the boiler. This relationship has been found to be highly accurate. It has limited application at locations away from the boiler, where the calorimeter can still be used.

3.1.10 Low-Quality Problem

When low steam quality is detected, it usually indicates a problem with the water level control, feed-water treatment, blow-down cycle or even boiler sizing. Equipment such as separators, baffles, and mist eliminators may have failed. Boiler or steam generator operating procedures may need to be changed. A boiler or steam generator that is making low-quality steam tends to overload the distribution system. Moisture increases the mass flow through the steam lines, increasing the pressure drop. Noise, velocity and generally poor operation can result. Steam quality is, therefore, an important function to control.

3.1.11 Redefine Steam Quality

When considering the definition of maximizing energy transfer, other characteristics of the system must be considered. If steam is to serve its function of heat energy transfer, the steam system should be designed, installed, and maintained to allow maximum heat energy transfer to occur. To do this, the steam must be

- Delivered at the highest pressure possible.
- Used at the lowest pressure possible.
- Be kept as clean as possible, especially at control devices.
- Be free from high concentrations of air and non-condensable, especially at condensing surfaces.
- Be free from excessive moisture (the traditional definition of steam quality).

By setting objectives to accomplish these steam quality goals, an action plan can be developed for a steam quality program. Implementation of a steam quality program takes the cooperation of engineers, energy staff, and operations. All these groups must understand the potential benefits and how improvement techniques can be applied.

3.2 Steam Quality Objectives

According to Deacon (1991), the following objectives should be recognized as an integral part of a program to accomplish the steam quality goals listed above:

3.2.1 Steam Pressure

- Steam pressure should be distributed at a high enough pressure as is practical to overcome line losses and satisfy the highest-pressure user.
- Steam pressure should be reduced at usage to as low a pressure as is practical.
- Steam velocity in distribution piping should not exceed $365\ \mathrm{m/s}$.
- Total pressure drop within the distribution piping should not exceed $20\ \%$ of boiler pressure.
- Steam pressure drops should be avoided to preserve total energy content and steam temperature.

3.2.2 Clean Pipelines

- Drip points and dirt legs within the system should be present in the system at regular intervals.
- Control valves should be protected by a strainer which is free from condensation accumulation.

3.2.3 Heat Transfer Potential

- Use steam at the lowest possible pressure to take advantage of the low-pressure latent heat.
- Boiler feed-water should be deaerated to reduce non-condensable gas production.

- Heat transfer devices should incorporate automatic vents to reduce non-condensable accumulations.
- Steam traps should be selected to discharge condensation before it cools and becomes corrosive.
- Automatic vents and steam traps should be located properly to discharge air and non-condensable gases before serious accumulations occur.

3.3 Steam Quality Action Plan

For each objective listed above, these are some suggested steps to take in order to archive a quality steam (Deacon, 1991).

3.3.1 Steam Pressure

- Investigate generating steam at higher than the required pressure up to $16\ \mathrm{bar(g)}$ to take advantage of specific volume. At higher pressures, steam volume decreases, and more steam can flow through a given size pipe. For the same flow, less pressure drop and less energy loss will occur.
- Investigate the steam pressure requirements of major users. Latent heat savings and improved control are possible by utilizing pressure reducing valves, temperature control valves or sensor- control valve combinations.
- Compare pipe sizes of major distribution and branch lines to sizing charts. One major steam user found a boiler expansion could be avoided by adding and re-piping several lines to reduce pressure drops.
- While comparing line sizes also check for steam velocities to be not above the $1.82\ \mathrm{m/s}$ to $3.65\ \mathrm{m/s}$ range and total system pressure drop should not exceed $20\ \%$ of the total maximum pressure of the boiler.

3.3.2 Clean Pipelines

- Drip legs with dirt pockets should be installed throughout the steam distribution system at least every $152.4\ \mathrm{m/s}$ and ahead of valves or regulators.
- A special situation exists ahead of valves that are protected by a strainer. The strainer body is a low point and accumulates condensation naturally, reducing the effective area of the strainer screen. See Figure 3.5, some of the condensations are picked up by the flow created when the valve opens, impacting the valve and seat with dirty condensation. Installing an inverted bucket steam trap on the strainer blow-down drains the condensation, freeing the screen area from blockage and keeps the strainer clean. See Figure 3.6, look for control valves that have perpetual dirt and wear problems to try this simple solution on.

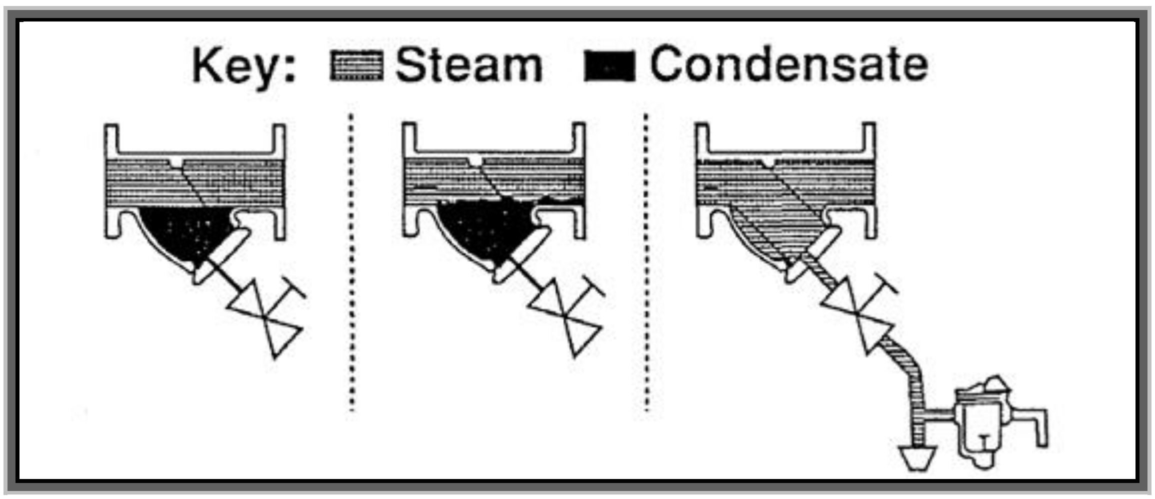


Figure 3.5: Automatic strainer blow-down prevents water carryover, Deacon (1991).

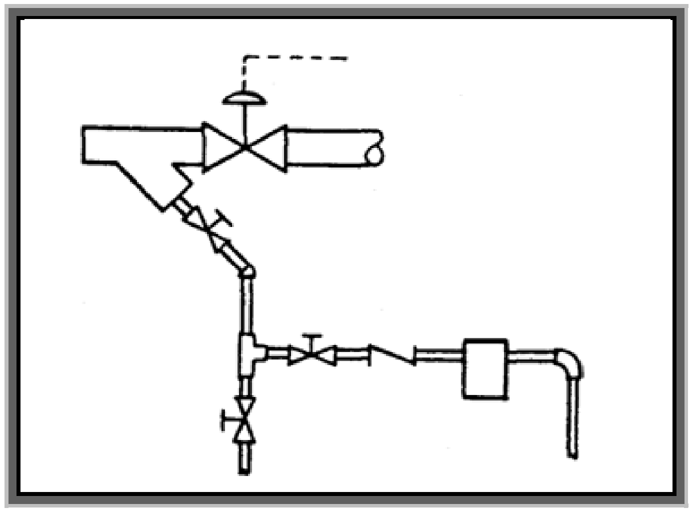


Figure 3.6: Trap draining strainer ahead of control valve, Deacon (1991).

3.3.3 Heat Transfer Potential

- Evaluate heat exchangers for oversizing or poor control. Pressure reducing stations for entire areas might be considered, but a pressure or temperature regulator at the equipment can achieve the same results with some savings. The central reducing station is not needed, the valve can often be smaller due to the greater pressure drop, and large size distribution piping is not needed due to the specific volume of the high-pressure steam.
- Relying on chemical treatment rather than deaerators should not be done. Many corrosion problems that make deaerators are nuisance due to the corrosion the system is supposed to prevent. Never mix makeup water with condensation return, even at the spray head. Always allow the deaerator to freely vent, never restrict the gas flow, and rather use a vent condenser to limit steam venting.
- Bellows-type thermostatic steam traps can be used as automatic air vents on heat exchanger equipment. Air in the system tends to be lighter than steam and does not condense, so it gets pushed to quiet zones by the flowing condensing steam. At these locations, the thermostatic device senses the temperature reduction caused by air. Batch

process cookers, large shell and tube heat exchangers, and large steam coils should incorporate automatic air vents to eliminate air accumulations. See Figure 3.7

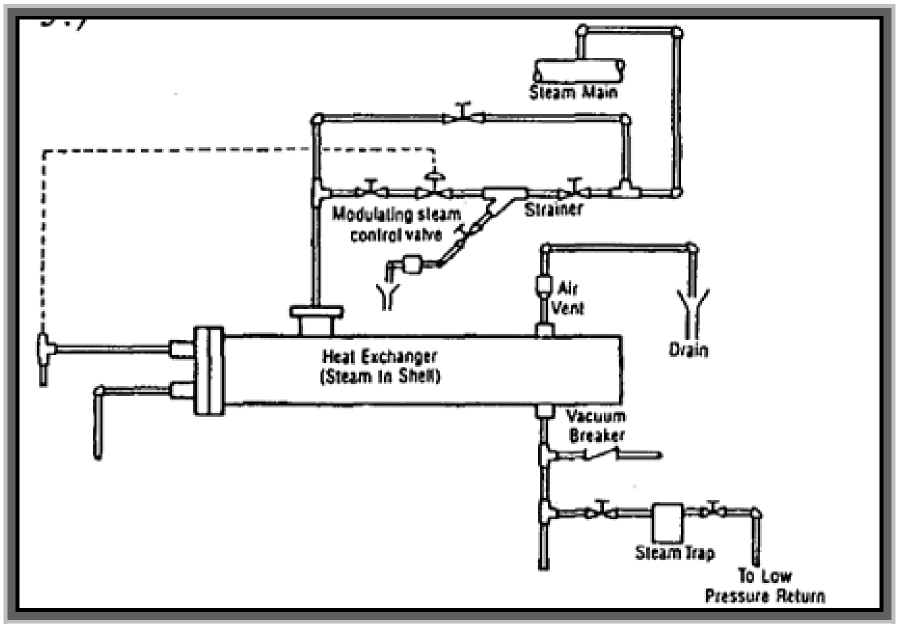


Figure 3.7: Shell and tube heat exchangers (Typical Piping Diagram), Deacon (1991).

- Steam traps should discharge condensation from process applications at or near saturation temperature. Selection of traps that back up or sub-cool condensation will accelerate carbonic acid corrosion, future steam leaks, and maintenance. Properly sized non-sub-cooling traps, such as inverted bucket traps and thermostatic traps will help maximize heat energy transfer.
- Air vents and steam traps must be properly located to function efficiently. When traps are installed, the steps should be followed: Accessible for inspection and repair, below drip points whenever possible, lose to a drip point. To locate air vents properly requires visualization or a good pyrometer.

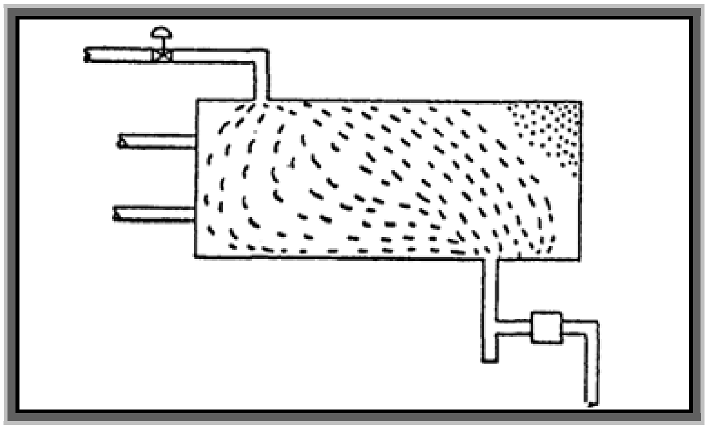


Figure 3.8: Steam pattern in a shell and tube heat exchanger, Deacon (1991).

With a pyrometer, it is possible to measure the skin temperature of the heat exchanger to find the gas accumulations as cold spots. This will be the proper place to locate an air vent. Without a pyrometer, you should visualize the flow within the heat exchanger. Imagine the flow of steam rushing in and condensing. See Figure 3.8, imagine the condensation flow down and out the drain. The proper air vent location is away from these active flow areas. Usually, the best locations are high points, away from the inlet. A non-sub-cooling steam trap properly located can usually take care of the lower quiet zones since the traps will vent air also.

3.4 Benefit of understanding quality and quantity

It was recommended to Isegen SA (Pty) Ltd to follow all the above steps in order to archive quality steam to their process plant. There is a big difference between generating good quality steam and delivering steam that has high usable quality. This difference centres on maximizing energy transfer. The traditional definition of steam quality is the percentage of steam to moisture. Yet high-quality steam should also be usable, efficient and effective. Steam that is delivered at a controlled pressure is clean and has high heat transfer potential are important quality features.

Chapter Four

4. Numerical Evaluation of Steam Consumption

This chapter investigates heat energy losses and steam consumption while flowing through the insulated long-distance pipeline. This study aims to present a basic application of heat transfer analysis to demonstrate the heat energy required and the heat losses during the steam supply to the process plant. The losses of heat energy at different steps of steam flowing through the insulated pipeline with turbulent flow characteristics are given. In this case, the steam is the working fluid, which is generated in the boiler and terminated in the process house. This chapter describes the methodology used, the model development and its validation.

The application of Excel spreadsheet has been used in order to develop the theoretical model of the problem. With the help of the computer package, the given problem was modelled and thoroughly analysed. The heat loss for different values of the thickness of insulation was obtained. The application of Excel spreadsheet was used to prove that as the ambient temperature decreased the heat loss increased and as the ambient temperature increased the heat loss decreased. It was also noticed that as the wind speed increased the heat loss increased.

This study can be used for a real-time assessment of the heat dissipation during steam transfer from one place to another and passing through cold atmospheric conditions. The use of Excel spreadsheet instead of other complex simulation software has proven to be advantageous and efficient. The significance of the numerical analysis method is highlighted and the scope of it being presented as a solution to other possible further problems is discussed.

4.1 Introduction

The pipeline transport is used for the transportation of goods (generally fluids). A quick determination of the heat loss of a pipeline system has always been a difficult problem for engineers and pipe network designers because pipeline systems are complex and the ambient environment can be variable. Lin and Kuo (1988) stated that the mass flow rate through the pipe and the power required for the desired flow rate are highly dependent on the temperature of the fluid it is carrying. The pipe material and the insulation material have great effects on convective and conductive heat transfer through the system. At a particular insulation thickness, viscous dissipation is exactly equal to conductive heat loss. Steam stays at its ideal temperature throughout the length of pipeline and the need for heating stations is reduced. From a design standpoint, this insulation thickness is called optimal. Steam passes through the long, insulated pipelines and terminates at the process house. The processing house consists of the Sizing Plant, PA Plant, MA Plant, Foods Plant etc. at this location. The temperature of the steam at the boiler house is about $180°C$ to $198.3°C$ and at the process house $130°C$ to $143.6°C$, a significant difference. This temperature difference shows that there is unexpected heat transfer dissipation taking place during steam flowing through the pipeline. The project is to investigate the steam energy losses and to minimize heat energy losses by finding the optimal thickness of insulating material (mineral wool).

4.1.1 Problem Statement

The significance of steam is essential in the process industries like cement dry plant, sugar plant, fertilizer, and textile etc. The generation of steam takes place in a boiler house. The generated steam is supplied to the process house through insulated pipelines. When steam flows through insulated pipelines it loses its heat energy through the pipeline which leads to some steam becoming wasted in the process industry. Thus, it is necessary to determine the optimal insulation thickness to avoid heat losses during steam flow with uniform steam velocity.

4.1.2 Solution Methodology

Parametric Modelling in the application of Excel spreadsheet: With the help of an Excel spreadsheet, a parametric model study was carried out and the heat transfer at various thicknesses of insulation was determined. These results were based on an ambient temperature of $20℃$. The heat loss of the pipeline system increased as the insulation thickness decreased and it was validated that heat loss decreased systematically with the increase in insulation thickness.

The model also showed the effects of the ambient temperature and the wind speed on the heat loss along the steam pipeline.

4.1.3 Model Development

The application of Excel spreadsheet model programming: Formulating a mathematical model from the given problem statement is the most important aspect of engineering. So, for this scenario, the numerical evaluation of the heat transfer rate through the pipe wall the equation (4.1) below has been used, the equation is using a suitable solving scheme was carried out. The solution to the problem was obtained using an Excel spreadsheet model.

4.1.4 Theoretical Model

Figure 4.9 shows a part of the pipeline, where the inner and outer surfaces are exposed to hot steam and air respectively. The heat transfer process includes the convection inside the pipe, the conduction through the pipe, and the convection outside the pipe.

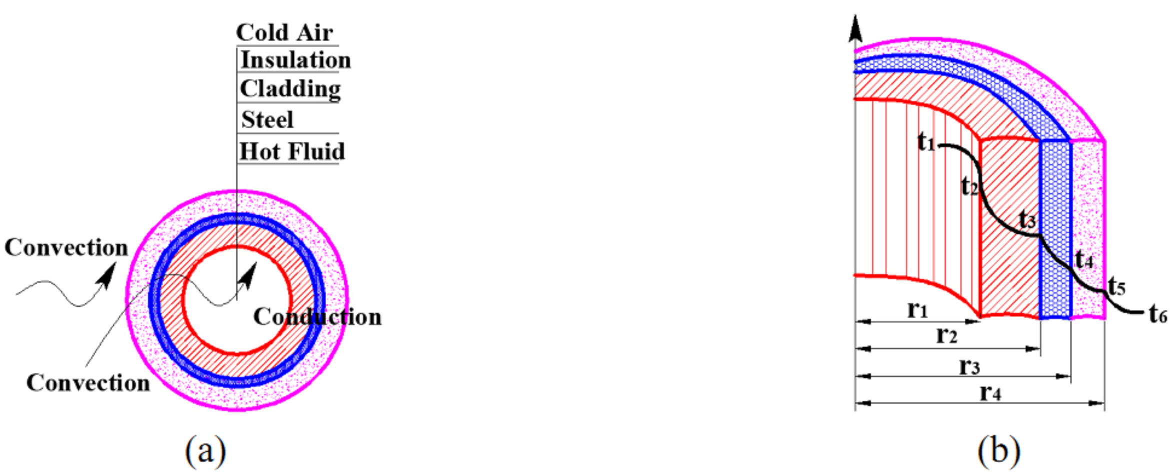


Figure 4.9: Heat transfer process and temperature distribution through insulated steam pipeline wall.

The heat lost per unit pipe length is the heat transfer through the inside steam boundary layer (forced convection), pipe wall, insulation, jacketing conduction and external air boundary layer convection-radiation. According to the thermal resistance theory, the heat transfer rate through the pipe wall is presented as (Bergman, Incropera et al., 2011).

$$\dot{q} = \frac{(t_1 - t_6)}{\frac{1}{2\pi r_{1\,L\,h_1}} + ln\left(\frac{r_2}{r_1}\right)/2\pi\lambda_1 L + ln\left(\frac{r_3}{r_2}\right)/2\pi\lambda_2 L + ln\left(\frac{r_4}{r_3}\right)/2\pi\lambda_3 L + \frac{1}{2\pi r_{4\,L\,h_0}}} \qquad (4.1)$$

Where t stands for the temperature, r is the radius, λ is the thermal conductivity, h is the convection heat transfer coefficient, and L is the length, respectively. In practical analysis, although the outside air temperature t_6 is usually considered as a constant, the inside steam temperature t_1 is a variable since it varies along the flow direction. By analysis and calculation, it is concluded that the temperature difference $(t_1 - t_6)$ may be assumed to be the logarithm mean temperature difference.

4.1.5 Heat transfer inside the pipeline

In industrial and civic heat supply engineering, the most common fluid inside includes hot water and steam. Although the heat transfers of these two fluids are all forced convection, the phase change of steam gives more challenge to the analysis in steam pipeline systems. In addition, the heat transfer inside the pipeline is an internal convection, such that another important characteristic is that the fluid is confined by the surface wall of the pipeline. As a result, the heat transfer phenomena inside pipeline are thus closely associated with the characteristic of the hot fluid and the dimension of the pipeline.

For steam, the condensation of steam has to be considered in the heat transfer process. Condensation is a phase change process from vapor to liquid. It occurs when the steam strikes the pipe well whose surface is at a temperature below the steam saturation temperature and the steam releases latent heat and converts to a liquid phase immediately (Tandon, Varma et al., 1995). Several empirical and semi-empirical correlations were suggested and one of the most widely used correlations is the Ackers Equation as follows (Yang and Webb, 1996).

The first step is to find the Equivalent Reynolds number using equation 4.1a in order to determine the internal convection heat transfer coefficient

$$Re = \frac{\dot{m}\, d_i}{\mu_{steam}} \tag{4.1a}$$

The inside convection heat transfer coefficient for steam $h_{i-steam}$ is thus obtained as

$$h_i = C\,\frac{\lambda_{steam}}{d_i}\,Re^n\,{Pr_{steam}}^{1/3} \tag{4.1b}$$

Bonda and Gavade (2017) recommended the approximate range values of the convective heat transfer coefficient as shown in Table 4.1 below.

Table 4.1: Approximate range values of the convective heat transfer coefficient.

Fluids	U W/m^2.K	Fluids	U W/m^2.K
Water to water	1300-2500	Steam to heavy organics	30-300
Ammonia to water	1000-2500	Light organics to light organics	200-350
Gas to water	10-250	Medium organics to medium organics	100-300
water to compressed air	50-170	Heavy organics to heavy organics	50-200
Water to lubricating oil	110-340	Light organics to heavy organics	50-200
Light organics to water	370-750	Heavy organics to light organics	150-300
Medium organics to water	240-650	Crude oil to gas oil	130-320
Heavy organics to water	25-400	Plate heat exchanger: water to water	3000-4000
Steam to water	2200-3500	Evaporator :steam/water	1500-6000
Steam to ammonia	1000-3400	Evaporator :steam/other fluids	300-2000
Water to condensing ammonia	850-1500	Evaporator of refrigeration	300-1000
Water to Freon-12	280-1000	Condenser : steam/water	1000-4000
Steam to gas	25-240	Condenser : steam/other fluids	300-1000
Steam to light organics	490-1000	Gas boiler	10-50
Steam to medium organics	250-500	Oil bath for heating	30-550

4.1.6 Heat transfer outside the pipeline

The heat transfers from pipe wall to outside air is an external heat convection between solid boundary and moving fluid. In general, pipeline systems are exposed to the surrounding, such that the meteorological conditions have a great influence on the heat loss of the pipeline systems. A well-known empirical correlation contributed by Hilpert is presented as (Rathore and Kapuno, 2011).

$$Nu = CRe_{eq}^{n}\, Pr^{1/3} \tag{4.1c}$$

while Re stands for the Reynolds number, and Pr the Prandtl number respectively. C and n are constants and tabulated in the below Table 4.2 at various Reynolds numbers. This empirical correlation is widely used for $Pr \geq 0.7$.

Table 4.2: The values of C and n (Rathore and Kapuno, 2011).

Re	***C***	***n***
0.4~4	0.989	0.330
4~40	0.911	0.385
40~4000	0.683	0.466
4000~40000	0.193	0.618
40000~400000	0.0266	0.805

According to Equation $4.1c$ and the definition of the Nusselt number (Tao, 2001), the outside convection heat transfer coefficient h_o is thus presented as

$$h_o = C\,\frac{\lambda_{air}}{d_i}\,Re^{n}\,{Pr_{air}}^{1/3} \tag{4.1d}$$

It is very important to determine the Equivalent Reynolds number using equation 4.1e in order to determine the external convection heat transfer coefficient

$$Re = \rho v D/\mu \tag{4.1e}$$

The above-mentioned correlation is a simplified working formula, which facilitates the solution of such problems based on the current conditions on site. The heat lost for total pipe length, $\dot{q}$, is the heat transfer through the inside steam boundary layer (forced convection) pipe wall, insulation and external air boundary layer convection-radiation. The heat flow through the pipe per unit length neglecting the jacketing conduction and the fittings such as valves and bends, Tee piece. The total heat loss of the steam pipeline based on the current conditions would be obtained using the equation (4.1).

4.2 Conduction

According to Bennett *et al*. (1983), heat conduction is the transfer of heat from one part of a body to another part of the same body, or from one body to another body in physical contact with it, without appreciable displacement of the particles of the body. Energy transfer by conduction is accomplished in two ways. The first mechanism is that of molecular interaction, in which the greater motion of a molecule at a higher energy level (temperature) imparts energy to adjacent molecules at lower energy levels. This type of transfer is present, to some degree, in all systems in which a temperature gradient exists and in which molecules of a solid, liquid, or gas are present. The second mechanism of conduction heat transfer is by "free" electrons. The free-electron mechanism is significant primarily in pure-metallic solids; the concentration of free electrons varies considerably for alloys and becomes very low for non-metallic solids. The distinguishing feature of conduction is that it takes place within the boundaries of a body, or across the boundary of a body, into another body placed in contact with the first, without an appreciable displacement of the matter comprising the body.

4.2.1 Convection

According to Kraus (2011), heat transfer by convection occurs in a fluid by the mixing of one portion of the fluid with another portion due to gross movements of the mass of fluid. The actual process of energy transfer from one fluid particle or molecule to another is still one of conduction, but the energy may be transported from one point in space to another by the displacement of the fluid itself. Convection can be further subdivided into free convection and forced convection. If the fluid is made to flow by an external agent such as a fan or pump, the process is called "forced convection". If the fluid motion is caused by density differences which are created by the temperature differences existing in the fluid mass, the process is termed "free convection", or "natural convection". The motion of the water molecules in a pan heated on a stove is an example of a free convection process. The important heat transfer problems of condensing and boiling are also examples of convection, involving the additional complication of a latent heat exchange. It is virtually impossible to observe pure heat conduction in a fluid because as soon as a temperature difference is imposed on a fluid, natural convection currents will occur due to the resulting density differences.

4.2.2 Radiation

Thermal radiation describes the electromagnetic radiation that is emitted at the surface of a body which has been thermally excited. This electromagnetic radiation is emitted in all directions; and when it strikes another body, the part may be reflected, the part may be transmitted, and part may be absorbed. Thus, heat may pass from one body to another without the need for a medium of transport between them. In some instances, there may be a separating medium, such as air, which is unaffected by this passage of energy. The heat of the sun is the most obvious example of thermal radiation.

4.3 Conduction heat loss

Fourier's law states that the rate of flow of heat through a single homogeneous solid is directly proportional to the area A of the section at right angles to the direction of heat flow, and to the change of temperature with respect to the length of the path of the heat flow, $\left(\frac{dy}{dx}\right)$. This is an empirical law based on observation. The law illustrated in figure 4.10 in which a thin slab of material of thickness dx and surface area A has one face at a temperature t and the other at a temperature $(t + dt)$. Then applying Fourier's law we have for the rate of heat flow in the direction x,

$$\dot{Q} \propto A \frac{dt}{dx} \tag{4.2}$$

Or

$$\dot{Q} = -\lambda A \frac{dt}{dx} \tag{4.3}$$

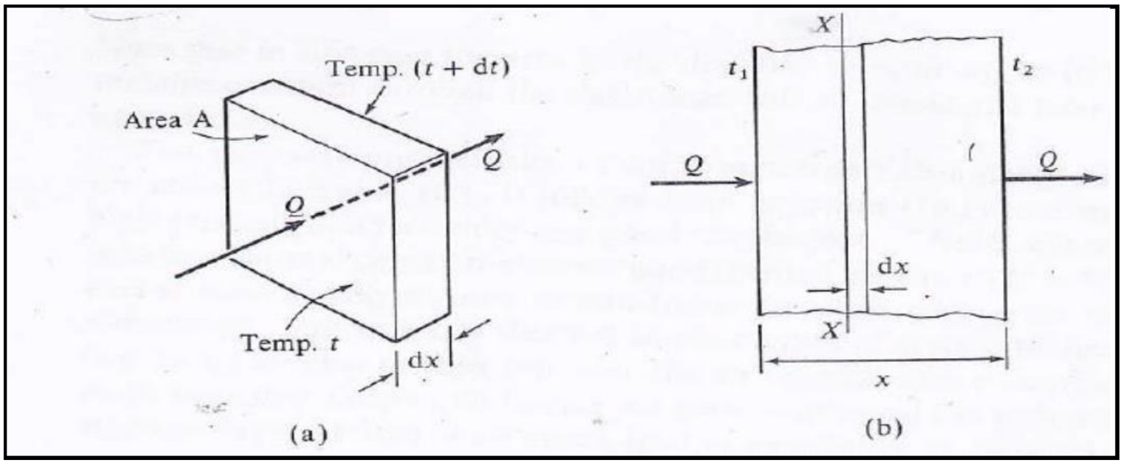


Figure 4.10: The transfer of heat through a slab of material (Eastop and McConkey, 2006).

We have,

$$\dot{Q} = -\lambda A \frac{dt}{dx} \; or \; \dot{Q}(dx) = -\lambda A(dt) \tag{4.4}$$

Integrating,

$$\int_0^x \dot{Q}(dx) = -\int_{t_1}^{t_2} \lambda A \,(dt) \tag{4.5}$$

Or

$$\dot{Q}(x) = -A \int_{t_1}^{t_2} \lambda \,(dt) \tag{4.6}$$

This equation can be solved when the variation of thermal conductivity $\boldsymbol{\lambda}$ with the temperature t is known. Now for most solids, the value of thermal conductivity is near constant over a temperatures range, therefore, $\boldsymbol{\lambda}$ can be considered as constant,

$$\dot{Q}(x) = -A\lambda \int_{t_1}^{t_2} (dt) \tag{4.7}$$

Or

$$\dot{Q} = -\frac{\lambda A}{x}(t_2 - t_1) = \frac{\lambda A}{x}(t_1 - t_2) \tag{4.8}$$

4.4 Newton's law of cooling and overall Heat Transfer Coefficient

In order to consider the rate at which heat is transferred from one fluid to another through a plane wall, it is necessary to know something of the way in which heat is transferred from a solid surface to a fluid and vice versa.

Newton's law of cooling states that heat transfers from a solid surface of area A, at a temperature t_w, to a fluid of temperature t, which is given by,

$$\dot{Q} = hA(t_w - t) \tag{4.9}$$

Where h is called the convection heat transfer coefficient.

Figure 4.11 shows the heat transfer taking place from fluid A to fluid B through a dividing wall of thickness x, thermal conductivity $\boldsymbol{\lambda}$, and a temperature variation across the wall $(t_1 - t_2)$.

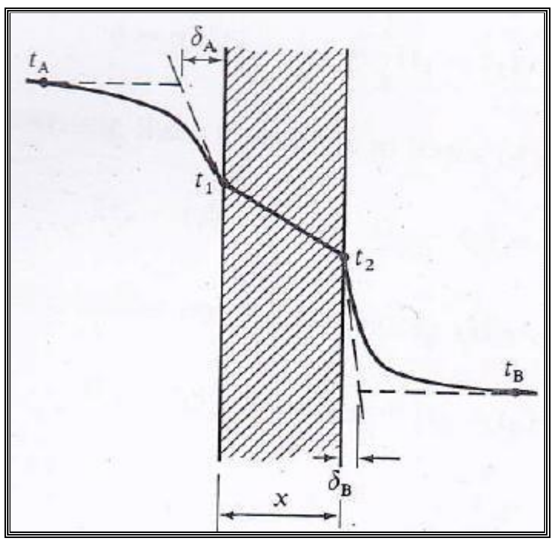


Figure 4.11: Temperature variation of the heat energy from fluid A to B (Eastop and McConckey, 2006).

In fluid A, the temperature decreases rapidly from t_A to t_1 in the region of the wall and similarly, in fluid B, the temperature decreases rapidly from t_2 to t_B in this region of the wall. In most practical cases the fluid temperature is approximately constant through its bulk, apart from a thin film near the solid surface bounding the fluid. The thickness of the fluid film is given by δ_A for the fluid A and δ_B for the fluid B. This is shown by the dotted line on Figure: 4.11. The heat transfer in these films is by conduction only, hence applying equation (4.8) and consider a unit surface area from A to the wall.

The overall heat transfer coefficient equation is divided as follows

From the fluid A to the wall,

$$\dot{q} = \frac{\lambda_B}{\delta_A}(t_A - t_1) \tag{4.10a}$$

From the wall to fluid A,

$$\dot{q} = \frac{\lambda_B}{\delta_A}(t_2 - t_B) \tag{4.10b}$$

Also from equation 4.4, from fluid A to the wall,

$$\dot{q} = h_A(t_A - t_1) \tag{4.10c}$$

From the wall to fluid B,

$$\dot{q} = h_B(t_2 - t_B) \tag{4.10d}$$

Comparing equations $(4.10a)$ and $(4.10c)$, and equations $(4.10b)$ and $(4.10d)$, it can be seen that,

$$h_A = \frac{\lambda_A}{\delta_A} \text{ and } h_B = \frac{\lambda_B}{\delta_B} \tag{4.11}$$

In general, $h = \lambda/\delta$, where δ is the thickness of the stagnant film of fluid on the surface. The heat flow through the wall in Figure 4.10 is given by equation 4.8.

For the unit surface area,

$$q = \frac{\lambda}{x}(t_2 - t_1) \tag{4.12}$$

For steady-state heat transfer, the heat flowing from the fluid A to the wall is equal to the heat flowing through the wall which is also equal to the heat flowing from the wall to the fluid B.

Therefore,

$$\dot{q} = h_A(t_A - t_1) = \frac{\lambda}{x}(t_2 - t_1) = h_B(t_2 - t_B) \tag{4.13}$$

Rewriting these above equations in terms of the temperatures, we obtained

$$\frac{\dot{q}}{h_A} = (t_A - t_1);\ \frac{\dot{q}x}{\lambda} = (t_2 - t_1);\ \frac{\dot{q}}{h_B} = (t_2 - t_B) \tag{4.14}$$

Hence by summing the corresponding side of the three equations,

$$\frac{\dot{q}}{h_A} + \frac{\dot{q}x}{\lambda} + \frac{\dot{q}}{h_B} = (t_A - t_1) + (t_1 - t_2) + (t_2 - t_B) \tag{4.15}$$

Therefore,

$$(t_A - t_B) = \dot{q}\left(\frac{1}{h_A} + \frac{x}{\lambda} + \frac{1}{h_B}\right) \tag{4.16}$$

Or

$$\dot{q} = (t_A - t_B)/\left(\frac{1}{h_A} + \frac{x}{\lambda} + \frac{1}{h_B}\right) \tag{4.17}$$

By analogy with equation 4.9 this can be written:

$$\dot{Q} = U(t_A - t_B) \tag{4.18}$$

Or

$$\dot{Q} = UA(t_A - t_B) \tag{4.19}$$

Where

$$\frac{1}{U} = \left(\frac{1}{h_A} + \frac{x}{\lambda} + \frac{1}{h_B}\right) \tag{4.20}$$

U is called the overall heat transfer coefficient and it has the same units as h.

The resistance analogy method, the heat flow is caused by the temperature difference whereas the current flow is caused by the potential difference V. Hence, it is possible to postulate a thermal resistance analogous to an electrical resistance. From Ohm's law, we have

$$V = IR \text{ or } I = \frac{V}{R} \tag{4.21}$$

Where V is the potential difference, I is the current and R is the resistance. Comparing the above equations with equation (4.8) with thermal resistance,

$$R = \frac{x}{\lambda A} \tag{4.22}$$

Where R is the thermal resistance, $\dot{Q}$ as analogous to I and $t_2 - t_1$ as analogous to V. To find the resistance of the fluid film it is necessary to compare Ohm's law with equation (4.9) given the thermal resistance of a fluid film,

$$R = \frac{1}{h_A} \tag{4.23}$$

4.5 The heat loss in the case of a hot insulated circular pipe

The heat flow resistance equation through the cylindrical wall is derived as follows. Eastop and McConkey (2006) consider a cylinder of internal radius r_1, and external radius r_2 as shown in Figure 4.12. Let the inside and outside surface temperatures be t_1 and t_2, respectively. Consider the heat flow through a small element, thickness d_r at any r, where the temperature is t. Let the conductivity of the material be $\boldsymbol{\lambda}$.

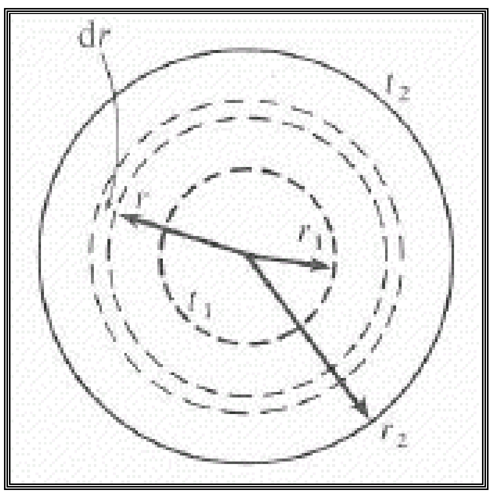


Figure 4.12: The insulated circular steam pipe (Eastop and McConckey, 2006).

Then applying the equation 4.3, for unit length in the axial direction, we have

$$\dot{Q} = -\lambda A \frac{dt}{dx} = -\lambda(2\pi r \; x \; 1)\frac{dt}{dx} \tag{4.24}$$

Or

$$\dot{Q}\frac{dr}{r} = -2\pi\lambda \, dt \tag{4.25}$$

Integrating between the inside and the outside surfaces,

$$\dot{Q}\int_{r_1}^{r_2}\frac{dr}{r} = -2\pi\lambda\int_{t_1}^{t_2} dt \tag{4.26}$$

Where $\dot{Q}$ and $\boldsymbol{\lambda}$ are both constant,

Therefore,

$$\dot{Q}\ \ln\frac{r_2}{r_1} = -2\pi\lambda(t_2 - t_1) = 2\pi\lambda(t_1 - t_2) \tag{4.27}$$

Or

$$\dot{Q} = \frac{2\pi\lambda(t_1 - t_2)}{\ln\frac{r_2}{r_1}} \tag{4.28}$$

Now from equation 4.8

$$\dot{Q} = \frac{\lambda A}{x}(t_1 - t_2) \tag{4.29}$$

Substituting a mean area A_m in this equation, and also substituting for the thickness $x = (r_1 - r_2)$, we have,

$$\dot{Q} = \frac{\lambda A_m(t_1 - t_2)}{(r_2 - r_1)} \tag{4.30}$$

Comparing this equation with equation 4.8, then,

$$\dot{Q} = \frac{\lambda A_m(t_1 - t_2)}{(r_2 - r_1)} = \frac{2\pi\lambda(t_1 - t_2)}{\ln\frac{r_2}{r_1}} \tag{4.31}$$

Therefore,

$$\frac{A_m}{r_2 - r_1} = \frac{2\pi}{\ln\frac{r_2}{r_1}} \tag{4.32}$$

Or

$$A_m = \frac{2\pi(r_2 - r_1)}{\ln\frac{r_2}{r_1}} = \frac{A_2 - A_1}{\ln\frac{r_2}{r_1}} \tag{4.33}$$

Thus, the solution can be achieved by substituting the logarithmic mean area A_m into equation 4.8. The logarithmic mean area formula also shows that a logarithmic mean radius can be given by as follows

$$r_m = \frac{r_2 - r_1}{\ln\frac{r_2}{r_1}} \tag{4.34}$$

In the case of a composite cylinder (e.g. a metal pipe with several layers of lagging) the most convenient approach is again that of the electrical analogy, so by using equation 4.22.

From equation 4.28 applying the electrical analogy *(I = V/R)*, it can be seen that,

$$R = \frac{\ln\frac{r_2}{r_1}}{2\pi\lambda} \tag{4.35}$$

The fluid film on the inside and outside surface can be treated using equation 4.23.

$$R_o = \frac{1}{h_O A_o} \tag{4.36}$$

Where A_o is the outside surface area $2\pi r_2$ referring to Figure 4.6, and h_o is the heat convection coefficient for the outside surface.

$$R_i = \frac{1}{h_o A_i} \tag{4.37}$$

Where A_i is the inside surface area, $2\pi r_1$ and h_i is the convection heat transfer coefficient for the inside surface. It can be seen from equation 4.11. The heat transfer rate depends on the ratio of the ${}^{r_2}/{r_1}$ and not on the difference $(r_2 - r_1)$. The smaller ratio ${}^{r_2}/{r_1}$ the higher is the heat flow for the same temperature difference.

$$\dot{Q} = \frac{2\pi\lambda(t_1 - t_2)}{\ln\frac{r_2}{r_1}} \tag{4.38}$$

4.6 The heat loss through radiation

According to Eastop and McConckey (1993), the fourth power of its absolute temperature of a black body is directly proportional to the emissive power. The thermal radiation that occurs during the steam distribution and uses to the manufacturing process transported through the insulation pipeline is a direct result of the random movements of saturated steam atoms and molecules. The saturated steam atoms and molecules are composed of charged protons and electrons. Hence, the random movements of the protons and electrons result in the electromagnetic radiation being emitted. This carries energy away from the steam pipeline to the environmental surroundings. The steam pipeline is considered to be a grey body because it is a hypothetical source that radiates as a black body but with an emissivity being less than one and with a constant wavelength. The steam pipeline does not absorb all the incident radiation but emits the radiation in constant proportion to the corresponding blackbody radiation.

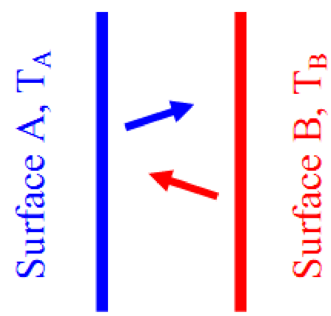


Figure 4.13: Relationship between absorptivity and emissivity (Eastop & McConkey, 1993).

This principle is as follows

$$E_B = \sigma T^4 \tag{4.39}$$

The value of σ is $5.67\ x10^{-8}\ W/m^2K^4$ and this value is constant. The rate of energy emitted by a non-black body is then given by:

$$E = \varepsilon\sigma T^4 \tag{4.40}$$

where ε is the emissivity of the non-black body.

Considering the body of emissivity ε_1 at a temperature T_1 that is completely surrounded by the black surroundings at a lower temperature T_2. The energy leaving the body is completely absorbed by the surroundings and from equation (4.40) the rate of energy emission is given by

$$\varepsilon_1\sigma A_1 T_1^4 \tag{4.41}$$

Now the fraction of an energy which is absorbed by the second body depends on the absorption of the body one. For a grey body $\alpha = \varepsilon$ at all temperatures and thus

$$Rate\ of\ energy\ absorption = \varepsilon_1\,\sigma A_1 T_1^4 - \varepsilon_2\sigma A_2 T_2^4 \tag{4.42}$$

In this case, the pipeline surroundings area, where the emitting of radiation heat transfer occurs was assumed to be equal to the pipeline area. Therefore, the net radiation loss rate of the pipeline radiating energy to its cooler surroundings can be expressed as

$$\dot{Q} = \varepsilon_1\,\sigma A_1 T_1^4 - \varepsilon_2\sigma A_2 T_2^4 \tag{4.43}$$

4.7 Heat loss across the cylindrical section

The heat resistance across the cylindrical shape of the steam pipeline is given by:

$$R = \ln\frac{r_2/r_1}{2\pi\lambda} \tag{4.44}$$

Where (r_1) is the internal radius and $(r_2)$is the outside surface radius. The thermal conductivity is represented by $\boldsymbol{\lambda}$. The heat energy first flows through the pipe wall, then flows through the insulation and finally through the aluminium cladding cover, before it dissipates into the pipeline surroundings.

4.8 The numerical model development

The saturated steam pipeline is made of carbon steel with a thermal conductivity of $43\ W/m$K **(see Appendix A)** and has a wall thickness of $7.11\ mm$ thickness. The numerical model was developed based on an ambient temperature of 20°C and $14.5\ bar(g)$ saturated steam. The steam pipe has the mineral wool insulation of $50\ mm$ thick with a thermal conductivity of $0.06\ W/m$K **(see Appendix F).** It is covered with $2\ mm$ thick aluminium cladding metal, which has a thermal conductivity of $237\ W/m$K **(see Appendix A)**. The pipeline had an outside diameter of $168.3\ mm$. The temperature of the saturated steam ranged between 200.1°C at the start and 198.9°C at the end. This was due to an acceptable pressure drop estimate of 0.1bar per 100 metres, however, this is for pipe that is expected to be in a good condition on the inside **(see Appendix Q)**. The inside heat transfer coefficient, h_i, and the outside heat transfer coefficient, h_o, were also taken into consideration during the model prediction analysis. Therefore, the total heat loss along the $250\ m$ long saturated steam pipeline calculations were obtained based on the above parameters, as well as the convection heat transfer values calculated below.

4.8.1 Calculation of heat loss through the insulated steam pipeline

The heat transfer through the inside steam boundary layer (forced convection), pipe wall, insulation and external air boundary layer convection-radiation. A heat transfer process can be considered as a heat loss due to the insulations materials limitations (inefficiency). Figure 4.14 below illustrates a schematic drawing of the materials in the steam pipeline wall.

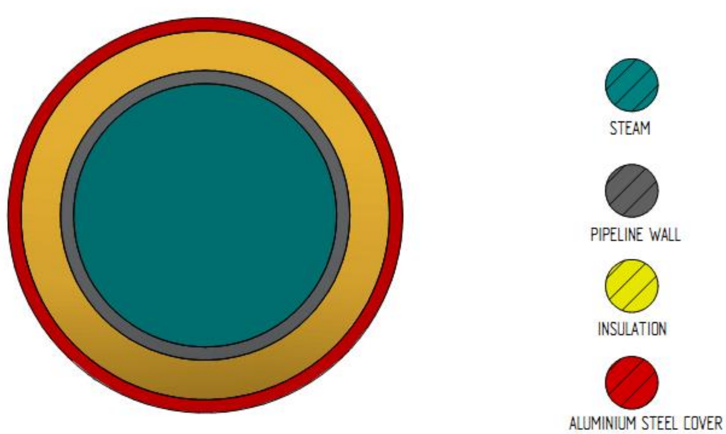


Figure 4.14: Schematic drawing of the materials in the steam pipeline wall.

4.8.2 The inside convection heat transfer coefficient for steam

The first step is to find the Equivalent Reynolds number Re is given by (Yang and Webb, 1996)

$$Re = \frac{\dot{m}\, d_i}{\mu_{steam}}$$

Table 4.2: Reynolds Number Calculations.

Items	Values	Units
$\dot{m}$=	2.22	kg/s
d=	0.154	m
μ=	0.0000157	kg/ms
Re=	**21775.7**	

According to (Yang and Webb, 1996), several empirical and semi-empirical correlations were suggested and one of the most widely used correlations is as follows,

where C = 0.0265, n = 0.8 for Re > 50 000,

C = 5.03, n = 1/3 for Re < 50 000

The inside convection heat transfer coefficient for steam is thus obtained as

$$h_i = C\,\frac{\lambda_{steam}}{d_i}\,Re^n\,{Pr_{steam}}^{1/3}$$

Table 4.3: Inside Convection Heat Transfer Coefficient.

Item	Value	Units
C=	5.03	
λ=	0.0375	W/mK
d=	0.154	m
Re=	21775.7	
Pr=	1.22	
n=	0.333	
const=	0.333	
h_i=	**36.42**	**W/m² K**

4.8.3 The outside convection heat transfer coefficient for air

The Reynolds number was calculated based on the yearly wind speed in the Durban area, which ranges from 3 to $10\ m/s$ **(See Appendix J)** and dynamic viscosity of $1.846x10^{-5}$ and also air density of $1.177\ kg/m^3$ **(see Appendix C)**.

$$Re = \rho v D/\mu$$

Table 4.4: The Reynolds Number for outside convection heat transfer coefficient.

ρ=	1.177	1.177	1.177	1.177	1.177	1.177	1.177	1.177	kg/m³
v=	3	4	5	6	7	8	9	10	m/s
D=	0.2723	0.2723	0.2723	0.2723	0.2723	0.2723	0.2723	0.2723	m
μ=	1.85E-05	1.85E-05	1.85E-05	1.85E-05	1.846E-05	1.846E-05	1.846E-05	1.85E-05	kg/ms
Re=	**52085.12**	**69446.83**	**86808.53**	**104170.24**	**121531.94**	**138893.65**	**156255.36**	**173617.06**	

When investigating the heat transfer coefficient of the outer surface, the Reynolds number, Prandtl number **(see Appendix C)**, and thermal conductivity of air **(see Appendix C)** were taken into consideration. The outside convection heat transfer coefficient for air is thus obtained as

$$h_o = C\ \frac{\lambda_{air}}{d_i}\ Re^n\ {Pr_{air}}^{1/3}$$

Table 4.5: The outside convection heat transfer coefficient.

At wind speed	3	4	5	6	7	8	9	10	m/s
C=	0.0266	0.0266	0.0266	0.0266	0.0266	0.0266	0.0266	0.0266	
λ=	0.026	0.026	0.026	0.026	0.026	0.026	0.026	0.026	W/mK
d=	0.2723	0.2723	0.2723	0.2723	0.2723	0.2723	0.2723	0.2723	m
R_e=	52085.12	69446.83	86808.53	104170.2	121531.94	138893.65	156255.36	173617.1	
P_r=	0.707	0.707	0.707	0.707	0.707	0.707	0.707	0.707	
n=	0.805	0.805	0.805	0.805	0.805	0.805	0.805	0.805	
const=	0.333	0.333	0.333	0.333	0.333	0.333	0.333	0.333	
h_o=	**14.31**	**18.04**	**21.59**	**25.00**	**28.30**	**31.51**	**34.65**	**37.72**	**W/m² K**

The table above shows the effects of wind speed on the heat transfer coefficient. It can be seen that as the wind speed increased, the convection heat transfer coefficient increased. In the model analysis, the heat loss per metre was compared between 3 and $10\ m/s$ from the prediction model was 131 and $136\ W/m$ which was a $3.7\ \%$ difference. It was found that both the inner and outer convection heat transfer coefficients had a very small effect on the heat loss per metre compared to the insulation resistance. The insulation resistance has the major effect on the heat loss per metre, based on the analysis that was made on the model by changing the thermal conductivity of the insulation between 0.06 to $0.08\ W/mK$, the heat loss at $3\ m/s$ wind speed changes from 131 to $168.9\ W/m$.

4.8.4 Heat losses through the valves' body including through the joining flanges.

(Mc Indicativ, 2006) stated that when referring to heat losses of a pipeline system one must take into account not only losses that correspond to the pipeline, but also those corresponding to the related elements (valves, fittings, and that are not insulated). To be able to take this into account, the related elements must be considered as an equivalent length.

4.8.5 The equivalent length of fittings

Spirax Sarco recommended that the allowance equivalent for each pair of the flanges should be $0.3\ m$ and each valve is $0.464\ m$ pipe. The current study has four pairs of flanges and two valves. Therefore, the total equivalent length of the fittings is $2.128\ m$ of pipe, and this amount was included in the total pipe length, as shown in the model below on Table 4.6. The numerical model data heat loss calculations are shown here based on $3\ m/s$. The other wind speeds of 4 to $10\ m/s$ can be found in **(Appendix M).**

4.8.6 Simplified Analytical Model

This study aims to develop a heat transfer evaluation system that will predict and quantify heat losses for various conditions of weather conditions. Based on the heat transfer theory, a theoretical model was established using the application of an Excel spreadsheet. Therefore, the model was developed for the assessment of the heat loss of pipeline systems.

The model was divided into $10\ m$ lengths and the heat loss for each $10\ m$ length was calculated and added together to calculate the total heat loss of the steam pipeline. Therefore, the following assumptions were made to set up the model,

- The pipeline was assumed to be perfectly thermally insulated along its length.
- Valves and flanges were uninsulated.
- The temperature of the saturated steam in the pipeline is constant at any cross-section and depends only on the pressure at that point along the length.
- The wind speed was between 3 to $10\ m/s$.
- The ambient temperature was ranging between 10 to 30°C based on the South African (Durban) weather service and, the numerical model table below was developed based on the average ambient temperature of 20°C.
- The mass flow rate of $8\ t/h$ was based on the boiler operating parameters.
- The pressure gauge of $14.5\ bar(g)$ was based on the boiler operating parameters.
- The heat transfer coefficient of the outer surface (h_o) was dependent on the Reynolds number which was calculated based on the yearly wind speed in the Durban area which ranges from 3 to $10\ m/s$, the Prandtl number, and the thermal conductivity of air.

Table 4.6: Numerical model data heat loss calculations at the average ambient temperature of 20℃ and wind speed of 3m/s.

The total heat loss of the steam pipeline based on the current conditions is thus obtained as

Pressure loss = 0.1 bar/100m ref : https://www.spiraxsarco.com/learn-about-steam/steam-distribution/pipes-and-pipe-sizing,figure 10.2.7

$$\dot{q} = \frac{(t_1 - t_6)}{\frac{1}{2\pi r_{1L\,h_i}} + ln\left(\frac{r_2}{r_1}\right)/2\pi\lambda_1 L + ln\left(\frac{r_3}{r_2}\right)/2\pi\lambda_2 L + \ ln\left(\frac{r_4}{r_3}\right)/2\pi\lambda_3 L + \frac{1}{2\pi r_{4L\,h_o}}}$$

Input	NB150 Insulated Steam Pipeline Section									Condensate drain point at 100m	NB150 Insulated Steam Pipeline Section		Equivalent length of pipe fittings	NB150 Insulated Steam Pipeline Section							Condensate drain point at 200m	NB150 Insulated Steam Pipeline Section						Manmanifold uninsulated pipeline systems	Units
$P_{sat\,(abs)}$ =	15.450	15.440	15.430	15.420	15.410	15.400	15.390	15.380	15.370	15.360	15.350	15.340	15.331	15.331	15.321	15.311	15.301	15.291	15.281	15.271	15.261	15.251	15.241	15.241	15.231	15.221	15.211	15.201	bar
t_s=	200.1	200.1	200.0	200.0	200.0	199.9	199.9	199.8	199.8	199.7	199.7	199.6	199.6	199.6	199.5	199.5	199.4	199.4	199.3	199.3	199.2	199.2	199.1	199.1	199.1	199.0	199.0	198.9	°C
t_a=	20																												°C
h_i=	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	W/m^2K
h_o=	14.3090																												W/m^2K
k_c=	43																												W/mK
k_{ins}=	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06		0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06		0.06	0.06	0.06	0.06		W/mK
$k_{clad(Alum)}$=	237	237	237	237	237	237	237	237	237	237	237	237		237	237	237	237	237	237	237	237	237		237	237	237	237		W/mK
r_1=	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.10137	m
r_2=	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.10955	m
r_3=	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415		0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415		0.13415	0.13415	0.13415	0.13415		m
r_4=	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615		0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615		0.13615	0.13615	0.13615	0.13615		m
$\dot{Q}$ =	**130.93**	**130.90**	**130.87**	**130.83**	**130.80**	**130.77**	**130.73**	**130.70**	**130.66**	**130.62**	**130.59**	**130.55**	**948.98**	**130.52**	**130.48**	**130.45**	**130.41**	**130.37**	**130.34**	**130.30**	**130.27**	**130.23**	**946.64**	**130.20**	**130.16**	**130.12**	**130.09**	**1234.63**	**W/m**
L=	10	10	10	10	10	10	10	10	10	10	10	9.354	0.646	10	10	10	10	10	10	10	10	9.354	0.646	10	10	10	10	1.5	m
$\dot{Q}$ =	**1.31**	**1.31**	**1.31**	**1.31**	**1.31**	**1.31**	**1.31**	**1.31**	**1.31**	**1.31**	**1.31**	**1.22**	**0.61**	**1.31**	**1.30**	**1.30**	**1.30**	**1.30**	**1.30**	**1.30**	**1.30**	**1.22**	**0.61**	**1.30**	**1.30**	**1.30**	**1.30**	**1.85**	**kW**
t_{surf}=	30.7	30.7	30.7	30.7	30.7	30.7	30.7	30.7	30.7	30.7	30.7	30.7	97.5	30.7	30.7	30.7	30.7	30.7	30.6	30.6	30.6	30.6	97.3	30.6	30.6	30.6	30.6	120.9	°C
																				Theoritical Condensate running load	0.0070	kg cond/s	25.12	Kg/h					
																				Measured Condensate running load	0.0203	kg cond/s	73.20	Kg/h	2.91	x as much	**Total Heat loss**	**35.54**	**kW**

4.8.7 The effects of the ambient temperature and wind speed on the heat loss of the pipeline system

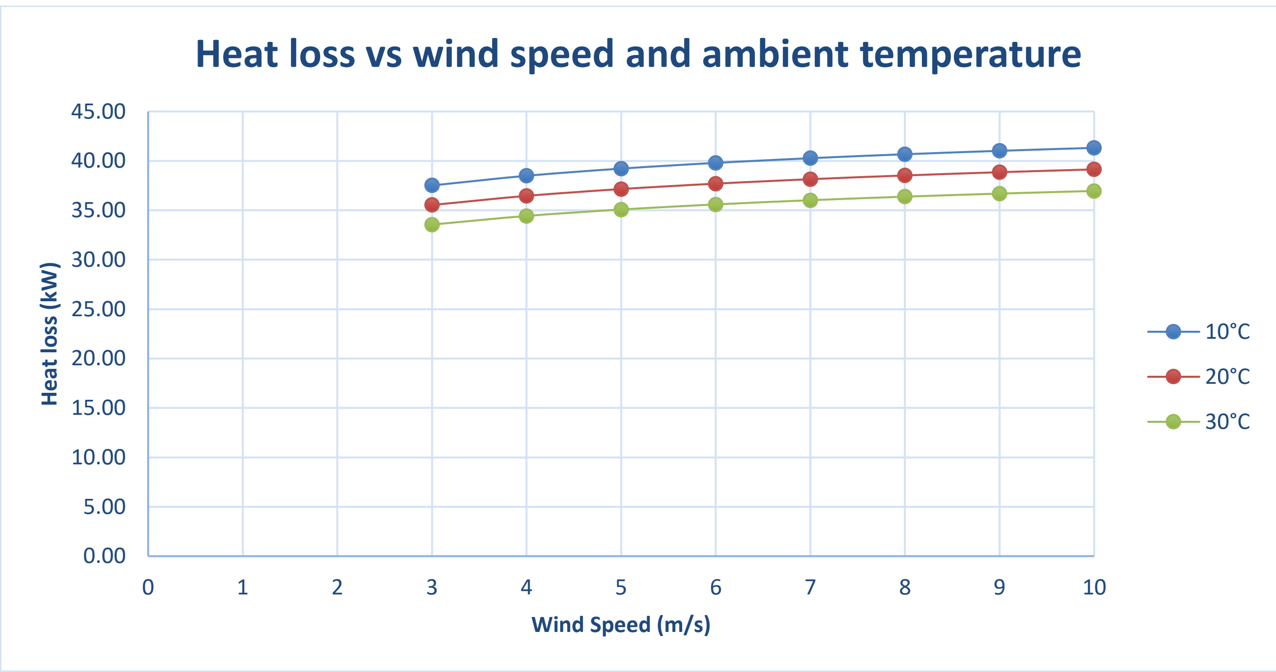
The data used in this investigation was obtained from South African Weather Services **(see Appendix K).** The annual wind speed measurements were taken at Durban South Merebank station from January 2014 to December 2015. The annual wind speed values are shown in **(Appendix J)** ranges from 3 to $10\ m/s$.

Table 4.7 and Graph 4.1 below show the effects of the wind speed on the heat loss of the pipeline system subjected to the different ambient temperatures. The ambient temperature of the Durban weather range between 10 to 30℃, so these temperatures were also tested in the model. The model average ambient air temperature is initially set to be 20℃. It can be seen that as the ambient temperature decreased the heat loss increased and as the ambient temperature increased the heat loss decreased. It can also be seen that as the wind speed increased the heat loss increased.

However, the outside convection heat transfer coefficient ranged between 14.31 to $37.72 W/m^2\ K$. It can be seen that the heat loss variation in each case is 10.1 %, based on the $3\ m/s$ wind speed as shown in **(Appendix M).**

Table 4.7: The effects of the ambient temperature and wind speed on the heat loss of the pipeline system.

Ambient Temperature	**10°C**	**20°C**	**30°C**
Wind speed (m/s)	**Heat Loss (kW)**	**Heat Loss (kW)**	**Heat Loss (kW)**
3	37.52	35.54	33.56
4	38.49	36.46	34.43
5	39.22	37.15	35.08
6	39.80	37.70	35.60
7	40.27	38.15	36.02
8	40.67	38.53	36.38
9	41.02	38.85	36.69
10	41.32	39.14	36.96



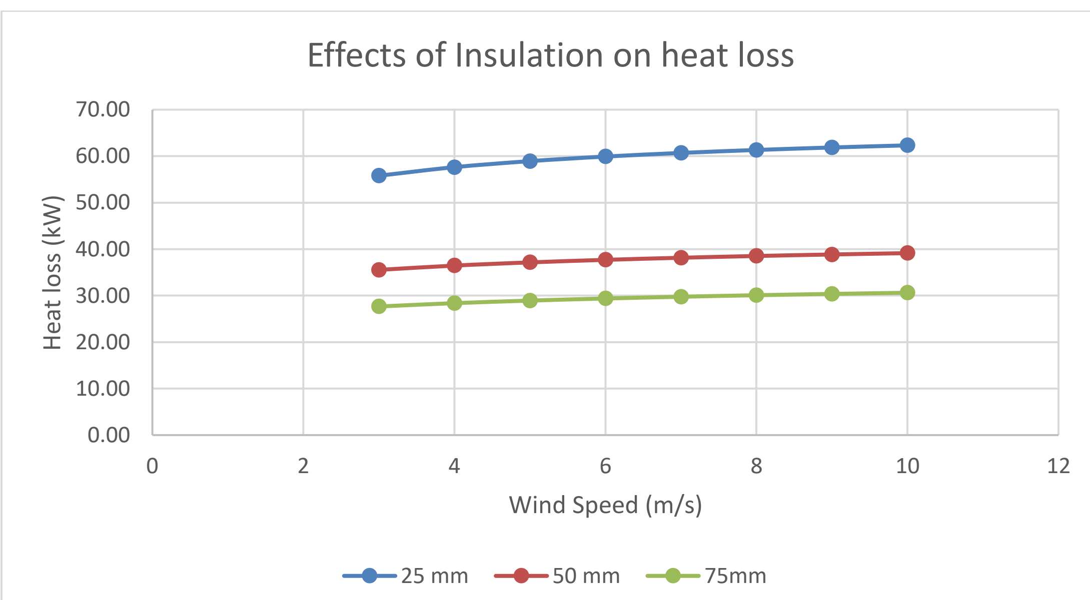
Graph 4.14: Heat loss vs wind speed and ambient temperature.

4.8.8 The effects of insulation on the heat loss of the pipeline system

These results are based on an ambient temperature of 20℃. As shown in Table 4.8 and Graph 4.2 below, the model shows that the heat loss of the pipeline system increases as the insulation thickness decreases and also as the wind speed increases. Furthermore, if the thickness of the insulation increases the heat loss decreases and that is resulting in less heat transfer to the ambient air.

Table 4.8: The effects of insulation on the heat loss of the pipeline system.

Thickness of Insulation	25 mm	50 mm	75mm
Wind Speed (m/s)	**Heat Loss (kW)**	**Heat Loss (kW)**	**Heat Loss (kW)**
3	55.80	35.54	27.66
4	57.64	36.46	28.38
5	58.93	37.15	28.93
6	59.91	37.70	29.38
7	60.69	38.15	29.76
8	61.33	38.53	30.09
9	61.87	38.85	30.37
10	62.33	39.14	30.62



15: Effect of insulation on heat loss.

4.9 Model Analysis Results

Spirax Sarco provides typical heat losses from insulated pipelines, which is $139.8\ W/m$ at a temperature difference of $180°C$, with $50\ mm$ of insulation (**See Appendix L).** The model's predicted heat losses, at a wind speed of $3\ m/s$ and 20°C, were found to be similar to those of the Spirax Sarco data seen in the Reference Data Heat Loss column in Table 4.9 below, which assumed still air at $20°C$ and $50\ mm$ insulation. As seen in the table below there is only a 6.3 % difference between the Spirax Sarco data and the model.

Table 4.9: Heat losses from the insulated pipeline.

Ambient Temperature (^{0}C)	**Reference Data Heat Loss (W/m)** Ref: Appendix L	**Model Prediction Heat Loss (W/m)**	**Difference Percentage (%)**
20	139.8	131	6.3

The steam condenses as heat is lost from the pipeline to the environment. The rate of condensation depends on such parameters as the steam temperature, the ambient temperature, and the efficiency of the lagging. After considering these and other factors including the pipe fittings, wind speed, and temperatures acting on the pipeline, the model was found to be similar to typical steam condensation rates as compared to the Spirax Sarco data seen in the reference data steam condensation column in the table below. The table 4.10 below shows the model prediction of condensate rate generated per $100\ m$ of well-insulated pipeline and compares it to the Spirax Sarco typical expected values for condensate generated for a pipeline with $50\ mm$ thick insulation with an ambient temperature of $20°C$ and $14.5\ bar(g)$ saturated steam **(see Appendix B)**.

Table 4.10: Amounts of steam condensed each hour per 100 m of insulated steam main.

Ambient Temperature (^{0}C)	**Reference Data Steam Condensation (kg/h)** Ref: Appendix B	**Model Prediction Condensation Rate (kg/h)**	**Difference Percentage (%)**
20	25.09	25.12	0.12

The model's predicted condensate rate per $100\ m$ of pipeline and $3\ m/s$ wind speed is close to the Spirax Sarco published data in still air at $20℃$ and $14{,}5\ b$(g) steam pressure, with only a 0.12% difference. However, the condensate that was measured during the experiment was found to be $73.20\ kg$ per hour which is 2.9 times compared to the model prediction condensate rate data. It can be seen that the condensate measured data values and the numerical prediction do not correlate. Therefore, these results can be caused by the following possibilities.

- A trapped pocket of the steam condenses is ineffective the $15\ \mathrm{mm}$ pipe has sufficient capacity it is unlikely to capture much of the condensation moving along the main at high speed.
- A missing, wet, or damaged insulation that could cause condensate accumulation and exceed steam traps' capacities.
- Pressure and Temperature drop along the steam main pipeline due to the age of the insulation and damage of cladding.

4.10 Surface Temperature Readings

The performance of insulation to provide personnel safety is indicated by the intensity of the outside surface temperatures. The lower the outside surface temperatures exhibited, the better the performance of the insulation materials, and consequently the more effective it is in providing personnel safety against burns.

The instrument that was used to take the temperature readings on the outside of the insulation was an infrared thermometer. The steam field tests mainly involve the measurements of the temperatures of the outside surface of the cladding. The measurements were taken at four distances along the pipeline, at intervals of $50\ m$. The surface temperature readings at each position along the pipe, at the X and Y axis position, were between 25 to 32℃, as seen in Table 4.11 below. The ambient temperature readings were also recorded as were provided by the Durban weather service. Table 4.11 below illustrates the correlation between the measured surface temperatures at different points of the pipeline and the model prediction surface temperatures based on the various atmospheric weather conditions.

Table 4.11: Insulated steam pipeline surface temperatures measurements.

Date	***Time***	***Fluid Transported***	***Position (x,y) at a distance of 50m;100m; 150m;200m***	***Measured Pipe Surface Temperature (℃) at (x)***	***Measured Pipe Surface Temperature (℃) at (y)***	***Model Predicted Surface Temperature (℃) at 3m/s***	***Percentage Difference between Measured and Predicted Surface Temperature (%)***	***Recorded Ambient temperatures (℃) for Durban weather***
4-Jun-18	Day	Saturated Steam	XY	30	30	30.5	1.6	20
4-Jun-18	Night	Saturated Steam	XY	26	25	26.9	3.7	16
11-Jun-18	Day	Saturated Steam	XY	31	30	32.6	4.9	22
11-Jun-18	Night	Saturated Steam	XY	26.8	27	26.9	0.4	16
15-Jun-18	Day	Saturated Steam	XY	30	30	36.3	17.4	26
15-Jun-18	Night	Saturated Steam	XY	26	25	21.3	14.8	10
20-Jun-18	Day	Saturated Steam	XY	31	30.5	31.6	1.9	21
20-Jun-18	Night	Saturated Steam	XY	26.5	25.8	26	1.9	15
25-Jun-18	Day	Saturated Steam	XY	29	30	31.6	5.1	21
25-Jun-18	Night	Saturated Steam	XY	25	26	22.2	11.2	11
29-Jun-18	Day	Saturated Steam	XY	30	30	32.6	7.9	22
29-Jun-18	Night	Saturated Steam	XY	25.3	25.7	22.2	12.3	11

4.10.1 Differences in surface temperatures

The investigation of determining the correlation between the measured surface temperatures at different points of the pipeline and the model prediction surface temperatures based on the various atmospheric weather conditions were determined. Based on the results that were obtained It can be seen that if the ambient medium temperatures that range between 15℃ to 21℃ have a reasonable comparison, but the lower temperature as 10℃ to below and the high temperatures that are above 22℃ do not correlate so well. However, this study was based on the average ambient temperature of 20℃ which falls under the good category where the correlation between the measured surface temperatures and the model prediction surface temperatures were found tied up well with the 1.6 % difference.

4.11 Numerical results and discussion

The numerical analysis was carried out to investigate the heat energy losses and condensate generation while flowing through the insulated steam pipeline between the ambient temperature and the operational steam temperature.

The total heat loss along the $250\,m$ long saturated steam pipeline calculations were successfully obtained and were compared to the Spirax Sarco data. During the analysis of the effects of the ambient temperature and wind speed on the heat loss of the pipeline, it has shown that as the ambient temperature decreased the heat loss increased, and as the ambient temperature increased the heat loss decreased. It can also be seen that as the wind speed increased the heat loss increased.

The condensate rate was physically measured during the experiment and was also investigated to the numerical model data. The model's predicted condensate was matching up with the Spirax Sarco published data with only a 0.12% difference. However, the condensate that was measured during the experiment was found to be 2.9 times compared to the model prediction condensate rate data. It can be seen that the condensate measured data values and the numerical prediction do not correlate. Therefore, further investigation needs to be carried out to be able to state the accuracy of the models. Furthermore, the effects of the insulation on the heat loss of the pipeline system were also investigated. The model shows that the heat loss of the pipeline system increases as the insulation thickness decreases and also as the wind speed increases. Furthermore, if the thickness of the insulation increases the heat loss decreases and that is resulting in less heat transfer to the ambient air.

Steam waste is costly in both an environmental and financial sense and, therefore, it requires prompt attention in order to ensure that the steam system is working at its optimum efficiency with a minimum impact on the environment. It is imperative that Isegen SA (Pty) Ltd uses appropriate methods to reduce the steam consumption, taking into account all the parameters that impact the cost fuel, condensation, power generation. It will eventually reduce the fuel costs, the emission surcharges as well as maximize the process efficiency.

Chapter Five

5. Experimental Evaluation

This chapter contains vital information on the type of experiments conducted at Isegen SA (Pty) Ltd Foods plant. It also shows the procedure for obtaining experimental data which is the basis for this research. The concept of transfer rate is the basic difference between heat transfer and thermodynamics. Thermodynamics does not answer the question of “how fast” a change is accomplished, but it can provide a solution to a system in equilibrium. Therefore, experimental analysis has been used to determine heat losses in a thermal system in order to predict and minimize thermal losses in the industry.

Calculation of heat losses from the identified pipeline segments was made with the use of a numerical evaluation. The aforementioned heat loss and temperature difference are determined for meteorological conditions occurring at the moment of pipeline inspection. The calculated quantities constitute a reference level during the evaluation of an existing pipeline. For a given nominal diameter of pipeline and temperature of the fluid inside the pipeline, the permitted value of heat loss for a unit length of pipeline or recommended insulation layer thickness were determined on the basis of the aforementioned as per BS 5422:2009 standards. Taking into consideration the thickness of thermal insulation and other parameters, insulation conduction resistance was calculated for thermal insulation complying with BS 5422:2009 standard requirements.

5.1 Gathering Information

- Study and verify line drawings.
- Read Process and Instrumentation drawings (P & ID’s).
- Look at “Isometric as-built” drawings.
- Confirm that blueprints are current.
- Review facility insulation specifications.
- Note equipment with special insulation requirements. This likely will give the original types and locations of insulation.
- Discuss the measurement system used by the plant.

5.2 The Appropriate Technique of Draining Steam Mains

Condensation is formed in the steam pipeline during the steam transport into the process plant as the result of saturated steam losing heat energy, known as latent energy, to the pipeline internals and the surrounding environment. Steam condenses into water when the latent heat of steam is transferred to heat the pipeline internals and the surrounding environment. The condensation contains a significant amount of sensible heat that can account for about 15% to 35% of the initial heat energy contained in the steam. The condensation is returned to the hot-well tank and, therefore, less make-up water is required and the sensible heat contained therein helps to improve the boiler efficiency.

Isegen SA (Pty) Ltd is experiencing problems with the condensation recovery system and steam leaking from the pipeline is often ignored. Leaks can be costly in both the economic and environmental sense and therefore need prompt attention to ensure the steam system is working at its optimum efficiency with a minimum impact on the environment.

Figure 5.16 illustrates the steam loss for various sizes of the hole at various pressures. This loss can be readily translated into a fuel-saving based on the annual hours of operation.

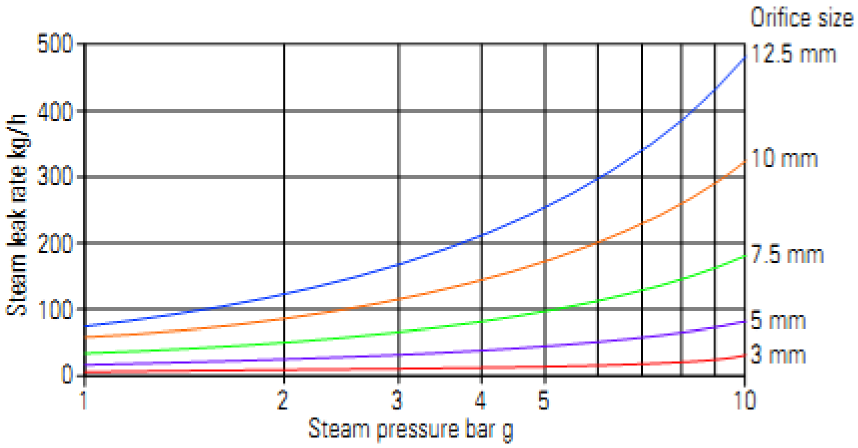


Figure 5.16: Steam leaking rate through holes (https://www.spiraxsarco.com/learn-about-steam/steam-distribution/steam-mains-and-drainage).

There are excessive heat energy losses that occur when draining the condensation during the steam transport. As mentioned above, Isegen SA (Pty) Ltd uses the recovered high-temperature condensation as part of the boiler feed-water, this maximizes the boiler output temperature because less heat energy is required to convert water into steam. These heat energy losses can negatively affect the steam energy available. The steam amount required in the process area will be less.

Steam traps are the most effective and efficient method of draining condensate from a steam distribution system. The steam traps selected must suit the system in terms of

- **Pressure rating:** Pressure rating is easily dealt with; the maximum possible working pressure at the steam trap will either be known or should be established.
- **Capacity:** Capacity, that quantity of condensate to be discharged, which needs to be divided into two categories; warm-up load and running load.
- **Suitability:** A mains drain trap should consider these constraints; discharge temperature, frost damage, and water hammer.

5.3 Experimental Setup and Procedures

The steam field tests mainly involve the measurements of the temperatures of the outside surfaces of the claddings and quantifying the condensate along the main steam pipeline. The infrared thermometer was used to measure the outside cladding temperature along the insulated steam pipeline. The pressure was read from the gauge pressure from the boiler and on the pipeline. This enabled the heat losses from the insulated steam pipeline to be calculated. The measuring wheel and measuring tape was used to measure the pipeline length. Relevant data about the pipeline on which the tests were carried out in the foods plant section at Isegen SA (Pty) Ltd are summarized below.

Steam transmission pipeline:

The outer diameter of the steam pipe: $(6") \ 168.3 \ \mathrm{mm}$

Pipe wall thickness: $7.11 \ \mathrm{mm}$

Operation steam pressure: $14.5 \ \mathrm{bar(g)}$

The thermal conductivity of mineral wool insulation: $0.06 \ \mathrm{W/mK}$

Pipe length: $250 \ \mathrm{m}$

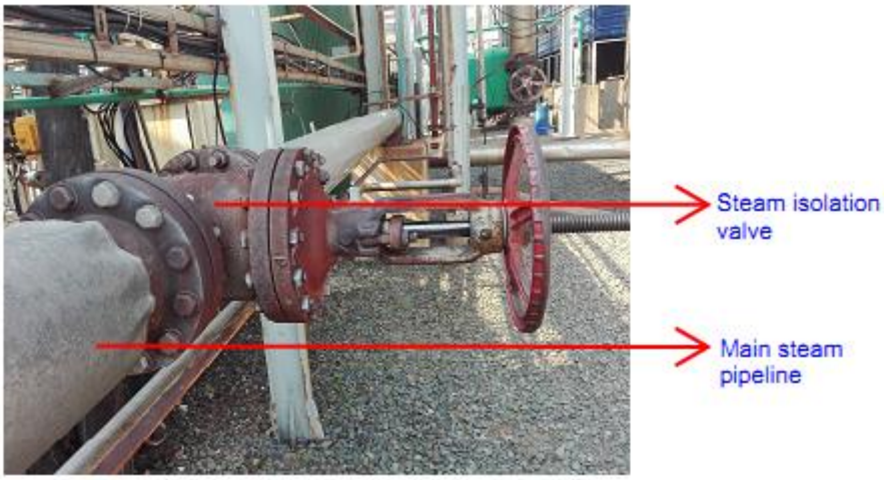


Figure 5.17: Isegen SA steam main pipeline.

A meeting was held with the Process Technical Engineer at Isegen SA (Pty) Ltd to discuss possible optimization of the existing condensation recovery system on the steam systems since there were excessive steam losses. Permission was granted to optimize the existing condensation recovery system, after lengthy discussions with Isegen SA (Pty) Ltd technical team. Two trials were carried out after researching about condensation recovery systems, which needs to be divided into two categories; warm-up load and running load.

However, the steam loss was established directly, by measuring the mass of condensate collected in a container over a period of time. Therefore, the container was weighed before and after in order to measure the quantity of condensate collected. Steam was then supplied to the plant, and any condensate is discharged below the water level in the container to condense any flash steam. By noting the increase in weight over time, the mean condensate collected can be determined. Although this method gives the mean rate of condensate collected, if the weight of condensate is noted at regular intervals during the test, the corresponding condensate rates can be calculated. Any obvious peaks will become apparent and can be taken into account when deciding on the capacity of the associated equipment. It is important to note that the test was conducted with the condensate discharging into an atmospheric system. Under normal operation the condensate would have a back pressure and the steam trap capacity would relate to the expected differential pressure. Five tests were performed under similar conditions in order to obtain more reliable results. Figure 5.18 shows a steam trap set and drain point of the current system.

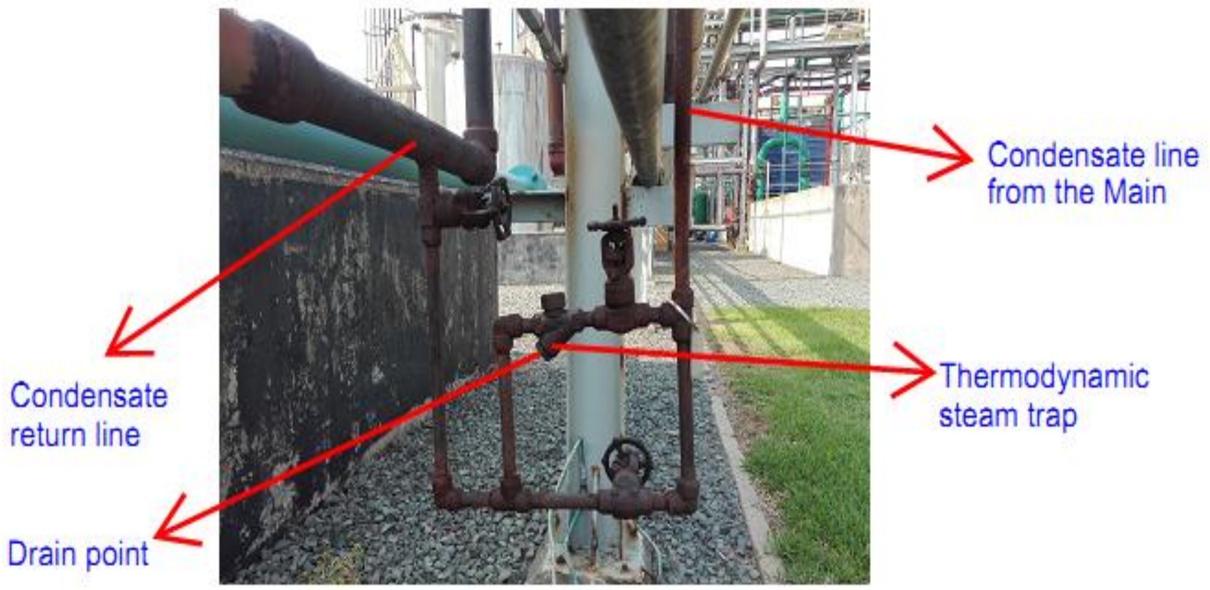


Figure 5.18: Isegen SA steam trap set.

5.3.1 Warm-up Load

During start-up, the pipework is cooler than the steam. This leads to heat transfer from the steam to the pipe and because the air surrounding the pipes is also cooler than the steam, heat transfer between the pipework and surrounding air also occurs. At the same time, steam that is in contact with cooler pipes will be condensed. The rate of condensation is at its maximum during start-up owing to the fact that the temperature difference between steam and the pipework is at its greatest. This condensation rate is called “starting load”. In the first instance, the pipeline needs to be brought up to operating temperature. This can be determined by calculation, knowing the mass and specific heat capacity of the pipeline and fittings.

The Table 5.12 shows the amount of condensation generated when bringing 50 m of main up to working temperature, 50 m being the maximum recommended distance between trapping points.

Table 5.12: Amount of steam condensed to warm-up 50 m of schedule 40 pipe.

Evaluation on Condensation			
Pressure (kPa)	**Pipe size (mm)**	**Condensation mass (kg)**	**Time (min)**
1350	150	28	12
1400	150	39	15
1450	150	42	18
1500	150	57	21

The results are based on the ambient temperature of 20 °C. The values shown in the above table are in kilograms. To determine the average condensing rate, the time taken for the process must be considered. In this case, the warm-up process required 57 kg of steam and was to take 21 minutes, then the average condensing rate would be

$$\boldsymbol{Average\ condensing\ rate} = {}^{60}/_{21}\ x\ 57\ \mathrm{kg} = 162\ \mathrm{kg/h}$$

5.3.2 Running Load

Once the pipework is warm the temperature difference between the steam and the pipework will naturally reduce. However, some condensation will continue to occur since the pipeline loses heat to its surroundings. This condensation rate is called "running load". Condensation that falls at the bottom of the pipe is carried along by steam and also by gravity to specific points from where it is drained. The same process occurs once steam flows from the distribution network into the utilization section. The energy of the steam is used in warming up the equipment and product and heat transfer continues even after operating temperature is achieved. Because condensation forms due to heat transfer it must be removed from both the distribution network and the utilization section. This condensate is an energy source and can be re-used as hot boiler feed water. Not recovering this condensate is simply a waste of energy and hence it is recommended to recover all the condensate and return it to the boiler feed water tank and complete the steam and condensation loop.

Once the steam main is up to operating temperature, the rate of condensation is mainly a function of the pipe size and the quality and the thickness of the insulation. Again, with sufficient data, the heat losses can be determined.

5.3.2.1 Measuring the mass of condensate.

Steam consumption can also be established directly, by measuring the mass of condensate collected in a drum over a period of time. This may provide a more accurate method than using theoretical calculations if the flash steam losses (which are not taken into account) are small, and can work for both non-flow and flow type applications. In this case an empty drum and platform scales were placed under the drain valve. This method was quite easy to set up and can be relied upon to give accurate results.

The drum was first weighed with a sufficient quantity of cold water to prevent burn. Steam was then supplied to the process plant, and any condensate was discharged below the water level in the container to condense any flash steam by noting the increase in weight over period of time. The weight of condensate was noted at regular intervals during the test, the average condensing rates can be calculated. Any obvious peaks will become apparent and can be taken into account when deciding on the capacity of associated equipment. It is important to note that the test is conducted with the condensate discharging into an atmospheric system.

Care was taken to ensure that only condensate produced during the test run is measured. The test should run for as long as possible in order to reduce the effect of errors of measurement. Five tests were conducted under similar conditions and the results average was obtained and used to calculate the average condensate rate. Care must be taken here, particularly if the level change is small or if losses occur due to flash steam. Table below shows the test results that were carried out.

Table 5.13 shows typical amounts of steam condensed per 100 m of the steam main at various pressures and also shows the test results that were carried out.

Table 5.13: Amount of steam condensed during operation of 100 m of schedule 40 pipe.

Date	Pressure (kPa)	Pipe size (mm)	Condensation mass (kg)	Time (min)	Condensate in 100m (Kg/h)	Average kg/min
6-Aug-18	1350	150	12	10	72	1.2
6-Aug-18	1400	150	18	15	72	1.2
6-Aug-18	1450	150	29	20	87	1.45
10-Aug-18	1350	150	14	12	70	1.17
10-Aug-18	1400	150	20	17	70.6	1.18
10-Aug-18	1450	150	28	22	76.4	1.27
15-Aug-18	1350	150	13	12	65	1.08
15-Aug-18	1400	150	21	18	70	1.17
15-Aug-18	1450	150	25	21	71.4	1.19
24-Aug-18	1350	150	15	13	69.2	1.15
24-Aug-18	1400	150	22	19	69.5	1.16
24-Aug-18	1450	150	27	23	70.4	1.17
30-Aug-18	1350	150	12	10	72	1.2
30-Aug-18	1400	150	20	17	70.6	1.18
30-Aug-18	1450	150	26	21	74.3	1.24
Total Average						1.20

The results are based on the ambient temperature of 20 °C. The average condensing rate determined in the same way as that shown above for 'running load'. As mentioned before that the weight of condensate was noted at regular intervals during the test, the average condensing rates was found to be as seen below:

$$\boldsymbol{Average\ condensing\ rate} = 1.20\ x\ 60 = 73.2 \text{ kg/h}$$

The average condensate rate was found to be 73.20 kg per hour which is 2.9 times compared to the model prediction condensate rate data. It can be seen that the condensate measured data values and the numerical prediction do not correlate.

According to Spirax Sarco, steam condenses as heat is lost from the pipe to the environment. The rate of condensation depends on the steam temperature, the ambient temperature and the efficiency of the lagging.

The model's predicted condensate rate per $100\ m$ of pipeline and $3\ m/s$ wind speed it was in comparison the Spirax Sarco published data in still air at 20°C and $14{,}5\ b$(g) steam pressure, and indicated a difference of 2.9 times compared to the Spirax Sarco published data. Therefore, the steam condensation rate for $100\ m$ insulated steam main pipeline is expected to be approximately $25\ kg/h$ **(see Appendix B)**.

The currently in use thermodynamic steam trap was able to remove the actual condensate during the tests at its maximum allowable pressure of 250 bar(g), **(see Appendix I)** for more details. The steam pressure drop did occur in the steam main line throughout the steam transport. Steam is an essential resource for Isegen SA (Pty) Ltd. It provides convenient, reliable and cost-effective energy. As such an indispensable tool, it is economically friendly to run a steam system at its optimum efficiency. Therefore, reducing the steam loss can lead to considerable savings in terms of energy and water resources. This optimization reduces the boiler fuel consumption and minimizes the plant's carbon footprint.

Spirax Sarco stated that the return of condensate to the boiler feed tank is commonly recognised as a highly effective way to improve the efficiency of the steam plant. Formed by condensed steam, liquid condensate needs to be drained from pipelines and equipment to avoid the risk of water hammer. Water hammer is a risk in a poorly drained steam main, where condensate collects and forms a slug of water. This water is incompressible unlike steam and can cause damage when carried along by the high-speed steam.

Condensate is a valuable resource and the recovery of even small quantities is often economically justifiable. Condensate recovery offers several benefits. It saves energy and reduces fuel costs, reduces water charges, and chemical treatment costs.

5.3.3 Distributed Control Systems.

Siemens briefly highlighted the introduction of the Distributed Control Systems (DCS), DCS are dedicated systems used in manufacturing processes that are continuous or batch-oriented. Process control of large industrial plants has evolved through many stages. Initially, control would be from panels local to the process plant. However, this required a large manpower resource to attend to these dispersed panels, and there was no overall view of the process. The next logical development was the transmission of all plant measurements to a permanently-manned central control room. Effectively this was the centralisation of all the localised panels, with the advantages of lower manning levels and easier overview of the process. Often the controllers were behind the control room panels, and all automatic and manual control outputs were transmitted back to the plant. However, whilst providing a central control focus, this arrangement was inflexible as each control loop had its own controller hardware, and continual operator movement within the control room was required to view different parts of the process. Below figure 5.19 shows the Isegen control room set up.

Figure 5.19: Isegen DCS Control Room.

With the coming of age of electronic processors and graphic displays, it became possible to replace these discrete controllers with computer-based algorithms, hosted on a network of input/output racks with their own control processors. These could be distributed around the plant, and communicate with the graphic display in the control room or rooms.

The introduction of DCSs allowed easy interconnection and re-configuration of plant controls such as cascaded loops and interlocks, and easy interfacing with other production computer systems. It enabled sophisticated alarm handling, introduced automatic event logging, removed the need for physical records such as chart recorders, allowed the control racks to be networked and thereby located locally to plant to reduce cabling runs, and provided high-level overviews of plant status and production levels.

5.3.3 DCS in Boiler Control System

Various parameters such as temperature, flow, level, and pressure have to be controlled proficiently to achieve greater efficiency. Compared to manual operation, automation prevents human errors and makes the process more efficient. DCS is made use of to automate the boiler control system. The Trend data acquire different types of process data and displays time-series changes in a graph shown in Figures 5.20. The time-series changes for the acquired process data are referred to as trend data.

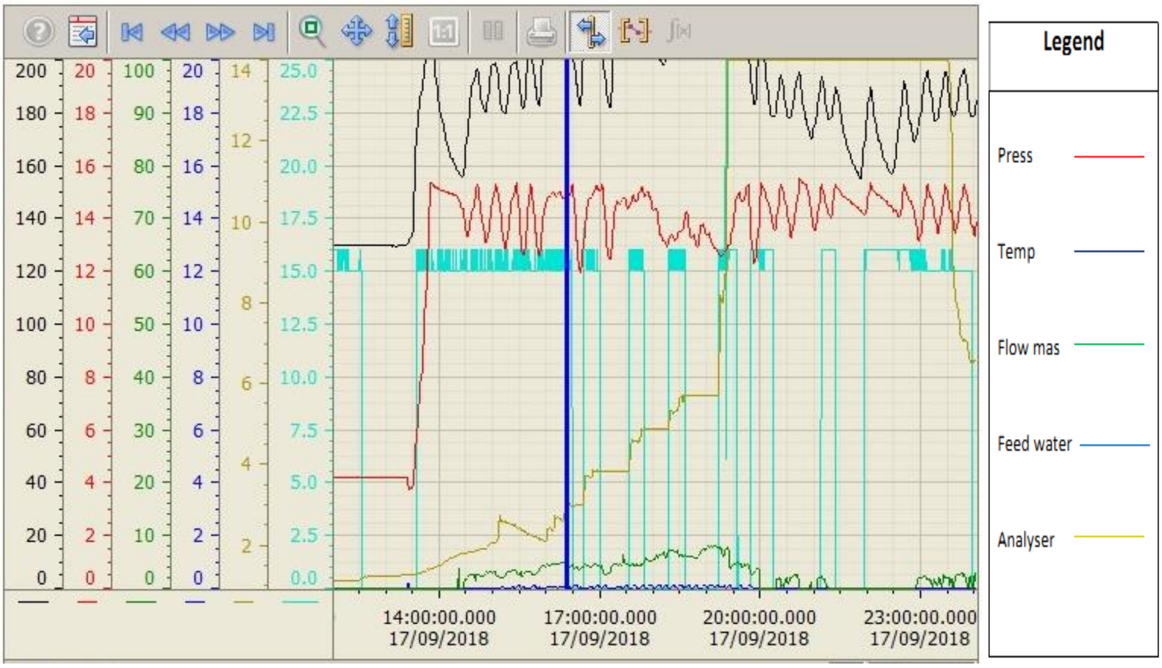


Figure 5.20: Boiler Tend Data from DCS.

Isegen, mainly use DSC system to control and monitors the boiler steam pressure and temperature and steam mass flowrate on a daily basis in order to prevents human errors and makes the process more efficient. The trends graphical proved that there was an effect of the temperature drops during the steam transportation from boiler house to process area especial at night and in winter when the ambient temperature is low **(see Appendix G)** for more details results.

However, some caution is required in interpreting the precise nature of this relationship. To provide clarity on how to ready the plant tends, on the right-hand side legends are provided while on the bottom there is date and time and on the left-hand side the numbers are allocated starting with Temperatures (℃), Pressure $(Bar.g)$, flow mass (kg/s) lastly feed water temperature (℃).

Data collection will eventually be used as a management tool to monitor and control energy consumption. In addition, some form of the electronic processor will exist which can receive, process and display the information. This processor may also receive additional signals for pressure and/or temperature to enable density compensation calculations to be made. Data may need to be gathered over a period of time to give an accurate picture of the process costs and trends.

5.4 Discussion of Experimental Results

The effect of thermal insulation conditions on the heat losses taking place at Isegen SA (Pty) Ltd steam pipeline network was studied. This was done by quantifying the magnitude of such losses and their comparison with the overall energy losses taking place during the steam transport from the boiler house to the process area. The appropriate analysis for the experimental measurements, lead to obtaining the heat transfer coefficient and heat losses at different operating parameters. The study was based on the results of a field inventory of the conditions of the pipelines thermal insulation.

This section presents the results and the outcomes of the experimental work performed at Isegen. The heat transfer in the pipeline was first analysed numerically to determine heat losses in a thermal system in order to predict and minimize thermal losses in the plant. The heat losses due to insulation inefficiencies have also been investigated.

Condensate is such a valuable resource that the recovery of even relatively small quantities is economically justifiable. Even the discharge from a single steam trap is often worth recovering. The average condensate rate was found to be 73.20 kg per hour which is 2.9 times compared to the model prediction condensate rate data and the Spirax Sarco reference data.

Further investigation will be required of the condition of the insulation along the pipeline should be carried out to determine what condition it is in to try to determine what seems to be causing so much condensate to be generated and removed. This excess condensate may be causing the problems in the process downstream as it may be being carried into the lines there, reducing the energy transfer to the process.

5.6 New Proposed Method

The aim of the study was to determine heat losses along the steam insulated pipeline system and minimize heat losses in the plant as well as also to improve the boiler efficiency. The Spirax Sarco (Pty) Ltd technical team was invited on site to provide more technical information regarding steam and steam condensation system recovery. The following points were discussed and proposed to be followed at all times: The steam lines should be arranged to fall in the direction of flow, at not less than $100\ mm$ per 10 metres of pipe (1:100). Steam lines rising in the direction of flow should slope at not less than $250\ mm$ per 10 metres of pipe $(1:40)$. Steam lines should be drained at regular intervals of $30-50\ m$ at any low points in the system. Where drainage has to be provided in the straight length of pipe, then a large bore pocket should be used to collect condensation. If strainers are to be fitted, then they should be fitted on their sides. Branch connections should always be taken from the top of the main from where the driest steam is taken. Traps selected should be robust enough to avoid water hammer damage and frost damage. The proper pipe alignment and drainage were proposed to the Isegen SA (Pty) Ltd technical team.

5.6.1 Steam Mains and Drainage

Throughout the length of a hot steam main, an amount of heat will be transferred to the environment, and this will depend on the parameters. With steam systems, this loss of energy represents inefficiency, and thus pipes are insulated to limit these losses. Whatever the quality or thickness of insulation, there will always be a level of heat loss, and this will cause steam to condensate along the length of the main.

In addition, the steam will become wet as it picks up water droplets, which reduces its heat transfer potential. If water is allowed to accumulate, the overall effective cross-sectional area of the pipe is reduced, and steam velocity can increase.

5.6.2 Steam Mains and Drainage

The subject of drainage from steam lines is covered in the UK British Standard BS 806:1993, Section 4.12. BS 806 stated that, whenever possible, the main should be installed with a fall of not less than $1:100$ ($1\ \mathrm{m}$ fall for every $100\ \mathrm{m}$ run), in the direction of the steam flow. This slope will ensure that gravity, as well as the flow of steam, will assist in moving the condensation towards drain points where the condensate may be safely and effectively removed see Figure: 5.21.

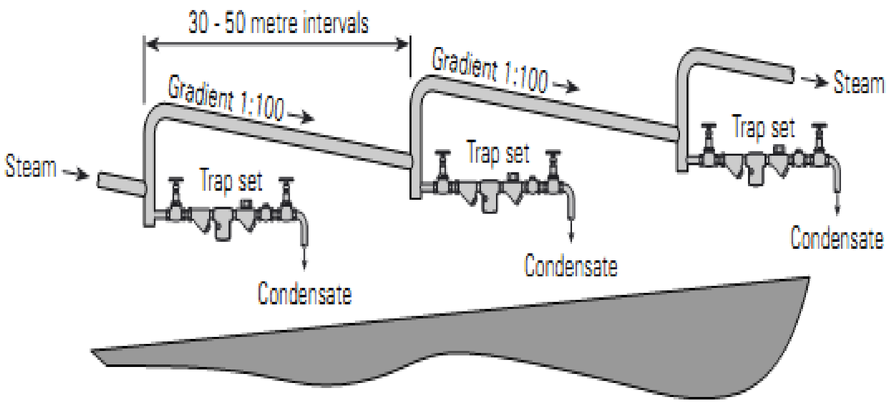


Figure 5.21: Typical steam main installation (https://www.spiraxsarco.com/learn-about-steam/steam-distribution/steam-mains-and-drainage).

5.6.3 Drain Points

The drain point must ensure that the condensation can reach the steam trap. Careful consideration must, therefore, be given to the design and location of drain points.

Consideration must also be given to condensation remaining in a steam main at shutdown when steam flow ceases. Gravity will ensure that the water (condensate) will run along sloping pipework and collect at low points in the system. Steam traps should, therefore, be fitted to these low points.

The amount of condensation formed in a large steam main under start-up conditions is sufficient to require the provision of drain points at intervals of $30\ \mathrm{m}$ to $50\ \mathrm{m}$, as well as natural low points such as at the bottom of rising pipework.

Isegen current drain point setup, the steam is flowing along the main dragging condensation along with it. Figure 5.22 is describing the Isegen current setup of the trap pocket, a $15\ \mathrm{mm}$ drain pipe connected directly to the bottom of a main.

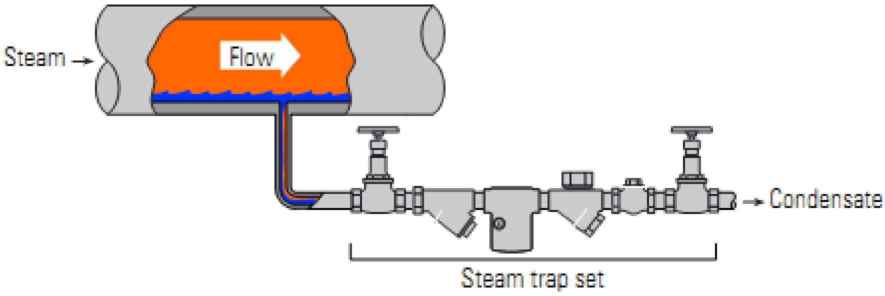


Figure 5.22: Trap pocket too small (https://www.spiraxsarco.com/learn-about-steam/steam-distribution/steam-mains-and-drainage).

Although the $15\ \mathrm{mm}$ pipe has sufficient capacity, it is unlikely to capture much of the condensate moving along the main at high speed and this arrangement was found ineffective.

The new proposed steam trap set is more reliable solution for the removal of condensation as shown in Figure 5.23. The trap line should be at least 25 to $30\ \mathrm{mm}$ from the bottom of the pocket for steam mains up to $100\ \mathrm{mm}$, and at least $50\ \mathrm{mm}$ for larger mains. This allows space below for any dirt and scale to settle.

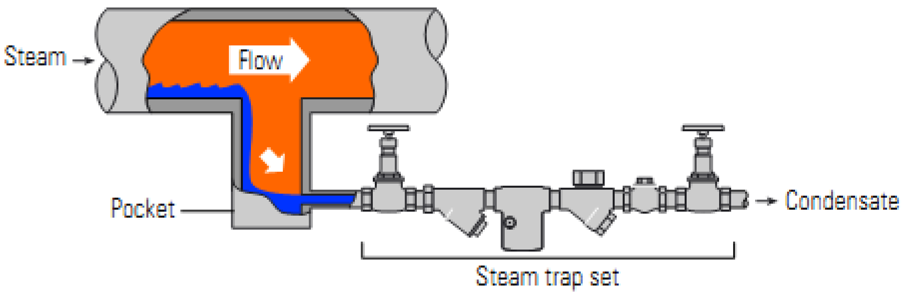


Figure 5.23: Trap pocket properly sized (https://www.spiraxsarco.com/learn-about-steam/steam-distribution/steam-mains-and-drainage).

The bottom of the pocket may be fitted with a removable flange or blowdown valve for cleaning purposes.

5.6.4 Steam Traps

The primary purpose of the steam trap is to discharge condensation, whilst not allowing live steam to escape. Due to the wide variety of applications under which steam traps are required to operate, they come in many shapes and sizes to suit those applications, including

- Thermostatic (operated by changes in fluid temperature)
- Thermodynamic (operated by changes in fluid dynamics)
- Mechanical (operated by changes in fluid density)

Steam traps are shown in Figure 5.24 below. The thermostatic trap is included because it is ideal where there is no choice but to discharge condensation into a flooded return pipe.

Figure 5.24: Steam traps suitable for steam mains drainage (https://www.spiraxsarco.com/learn-about-steam/steam-distribution/steam-mains-and-drainage).

5.7 Branch Lines

Branch lines are normally much shorter than steam mains. As a general rule, therefore, provided the branch line is not more than 10 metres in length, and the pressure in the main is adequate, it is possible to size the pipe on a velocity of 25 to $40\ \mathrm{m/s}$, and not to worry about the pressure drop.

5.7.1 Branch Line Connections

Branch line connections, taken from the top of the main, carry the driest steam Figure 5.25. If connections are taken from the side, or even worse from the bottom as in Figure 5.26(a), they can accept the condensation and debris from the steam main. The result is very wet and dirty steam reaches the equipment, which will affect performance in both the short and long term.

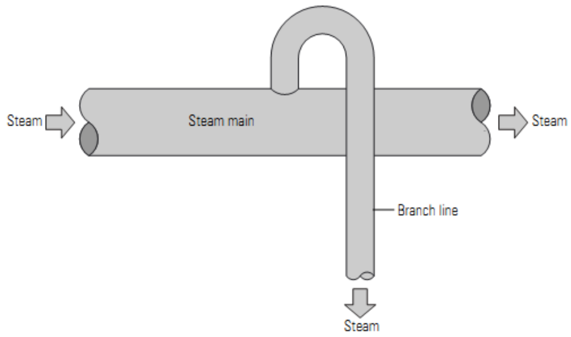


Figure 5.25: Branch line (https://www.spiraxsarco.com/learn-about-steam/steam-distribution/steam-mains-and-drainage).

The valve in Figure 5.26(b) should be positioned as near to the off-take as possible to minimize condensation lying in the branch line if the plant is likely to be shut down for extended periods.

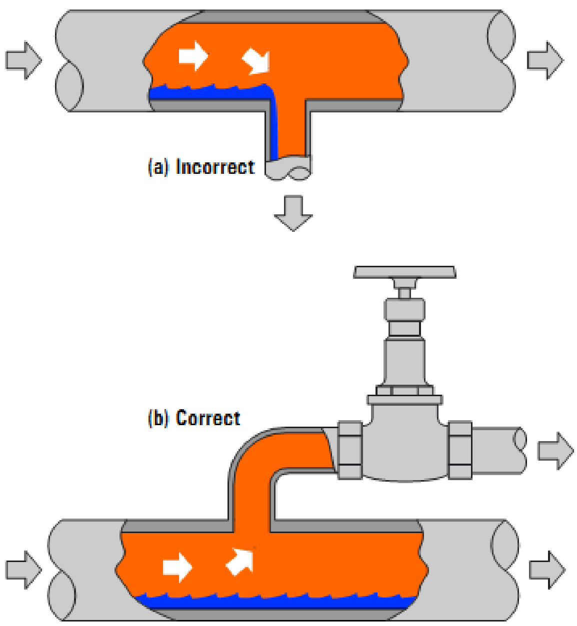


Figure 5.26: Steam off-take (https://www.spiraxsarco.com/learn-about-steam/steam-distribution/steam-mains-and-drainage).

5.7.2 Strainers

When a new pipeline is installed, it is not uncommon for fragments of casting sand, packing, jointing, welding rods, and even nuts and bolts to be accidentally deposited inside the pipe. In the case of an old pipeline, there will be rust, and in hard water districts, a carbonate deposit. Occasionally, pieces will break loose and pass along the pipeline with the steam to rest inside a piece of steam using equipment. This may, for example, prevent a valve from opening or closing correctly. Steam using equipment may also suffer permanent damage through wiredrawing, the cutting action of high-velocity steam and water passing through a partly open valve. Once wiredrawing has occurred, the valve will never give a tight shut-off, even if the dirt is removed.

It is therefore wise to fit a line-size strainer in front of every steam trap, flow meter, reducing valve and regulating valve. The illustration shown in Figure 5.27 shows a cutting section through a typical strainer.

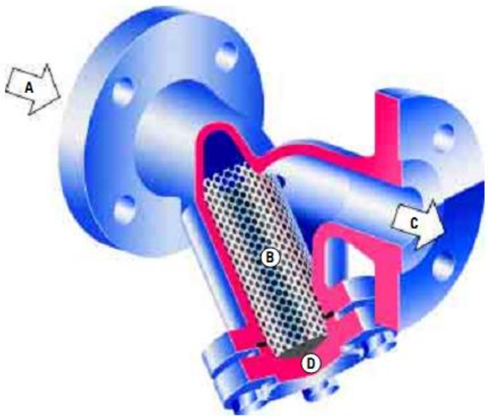


Figure 5.27: Cut section through a Y-type strainer (https://www.spiraxsarco.com/learn-about-steam/steam-distribution/steam-mains-and-drainage).

Steam flows from the inlet ‘A' through the perforated screen ‘B' to the outlet ‘C'. While steam and water will pass readily through the screen, dirt cannot. The cap ‘D' can be removed, allowing the screen to be withdrawn and cleaned at regular intervals. A blowdown valve also is fitted to cap ‘D' to facilitate regular cleaning.

Straners can, however, be a source of wet steam as previously mentioned. To avoid this situation, strainers should always be installed in steam lines with their baskets to the side.

5.8 Return Condensation to the Boiler

When steam transfers its heat in a manufacturing process, heat exchanger, or heating coil, it reverts to a liquid phase called condensate. An attractive method of improving the power plant energy efficiency is to increase the condensation return to the boiler.

Returning hot condensate to the boiler makes sense for several reasons. As more condensate is returned, less make-up water is required, saving fuel, make-up water, and chemicals and treatment costs. Less condensation discharged into a sewer system reduces disposal costs. Return of high purity condensation also reduces energy losses due to boiler blow down. Significantly, fuel savings occur as most returned condensation is relatively hot (54℃ to 100℃), reducing the amount of cold make-up water (10℃ to 16℃) that must be heated.

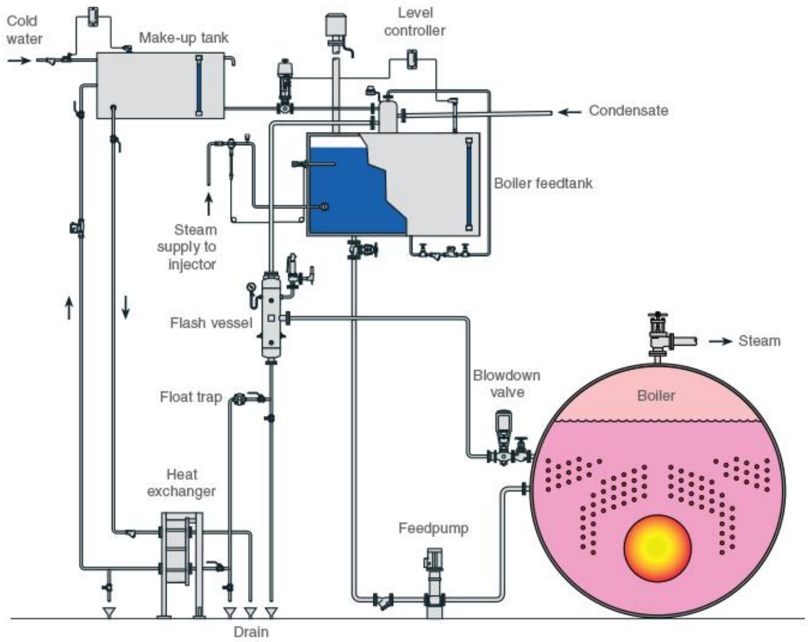


Figure 5.28: Typical heat recovery from boiler (https://www.spiraxsarco.com/learn-about-steam/condensate-recovery/flash-steam).

5.9 Insulation Condensation Return Lines

The insulation of condensation return lines allows for higher temperature condensation recovery to the hot well, which renders savings in fuel. Further, it reduces radiation energy transfer and thus contributes to a drop in the surrounding temperature in the workplace.

Chapter Six

6 Cost Analysis of Heat Energy loss and Production lost

Accurately determining heat energy loss and production lost: monitoring and managing energy use in a plant, evaluating proposed design changes to the generation and the process itself, and for continuing to identify competitive advantages through plant efficiency improvements. Steam costs are highly dependent on the path that steam follows in the generation and distribution system. Simulation models are simple, convenient, and reliable tools to follow these paths, calculate the correct costs, and to optimize the system. The method used for evaluating utility costs has a dramatic effect on project economics, and therefore the investment decision. Improper utility pricing can lead to bad decisions; good projects can be discarded, and bad projects can be implemented. Regrettably, this is relatively common. To avoid such mistakes, it is imperative that plant engineers and managers use appropriate methods for steam pricing, taking into account all the parameters that impact energy costs, fuel, condensation, power generation, and cooling water when evaluating the project.

The need to save energy is now necessary due to the recent increase in fuel and energy costs. The market strategy of Isegen SA (Pty) Ltd is to be one of the low-cost Malic Acid producers in order to maintain global competitiveness. However, Isegen has found it very difficult to maintain global competitiveness due to the significant rise in fuel and energy cost. Thus, optimizing the energy efficiency of the factory provides a very high potential to reduce their product prices while maintaining the quality.

Isegen SA (Pty) Ltd is using Sasol natural gas as a boiler consumption fuel. Department of Energy (2013) noted that the benefits of using natural gas are multiple and include inter alia being cleaner-burning, offering reduced emissions, and being consistent with regard to the quality/energy content.

6.1 Natural Gas

According to Sasol, the natural gas primarily consists of methane. It is found in deep onshore or offshore underground natural rock formations as a conventional gas or trapped within impermeable rock formations as unconventional shale gas. In its natural state, it can be transported in pipelines; or it can also be liquefied by refrigeration to produce Liquefied Natural Gas (LNG) for alternative transportation purposes.

South Africa has a history of gas as an energy source dating back to 1892. In 1966 the South African gas distribution company (now Sasol Gas) was formed to market and distribute pipeline gas on a broader scale. Initially, gas was sourced from industrial coal to gas processes. In 2004 the first natural gas from Mozambique arrived in Secunda through the over-border pipeline. Today the bulk of gas is supplied from natural (imported) sources as opposed to synthetically (locally) produced sources. South Africa has gas trade agreements with Namibia and Mozambique. Currently, natural gas is supplied from the Temane and Pande Mozambican gas field to South Africa.

According to the Department of Energy (2017), natural gas is already used in South Africa to produce diesel at the Mossel Bay facility as well as natural gas supplemented Coal-to-Liquid (CTL) at Sasol in Secunda. By expanding the Mossel Bay facility capability for South Africa, more liquid fuel in the form of diesel is produced locally. In 2001, the government of South Africa and Sasol Limited signed an agreement concerning the Mozambican gas pipeline. Sasol, which dominates the piped gas market in South Africa, has been exempted from regulated pricing for the past 10 years as the provisions of the agreement have prevailed over the Gas Act since 2004. The rationale was, among others, to promote the development of a commercial and competitive piped gas industry with active participation by the private sector, while at the same time introducing natural gas into the South African economy at the lowest cost and as fast as possible.

As per Greenpeace - The Advanced Energy Revolution (2015), the gas industry is a capital-intensive industry and Sasol and its partners have invested over R21 billion in the natural gas industry to date and continue to make further investments. The manufacturing circle estimates that gas accounted for about 20 % of input costs for some large manufacturers, and major users consumed 59 % of gas Sasol supplied to the domestic market.

Table 6.1 presents the South African historical natural gas prices supplied by Sasol Gas for 2017. The tariffs are grouped into six customers of average annual consumption measured in Rand per Gigajoule.

Table 6.1: 2017 monthly natural gas price in Rand per Gigajoule (https://www.sapia.org.za/Overview/Old-fuel-prices).

Period	Maximum price	Class 1	Class 2	Class 3	Class 4	Class 5	Class 6
		33 GJ	333 GJ	3 333 GJ	33 333 GJ	333 333 GJ	1 054 093 GJ
JAN	129.97	96.62	86.97	72.47	48.31	43.48	38.64
FEB	129.97	96.62	86.97	72.47	48.31	43.48	38.64
MAR	129.97	96.62	86.97	72.47	48.31	43.48	38.64
APR	128.98	95.89	86.31	71.91	47.94	43.14	38.35
MAY	128.98	95.89	86.31	71.91	47.94	43.14	38.35
JUN	128.98	95.89	86.31	71.91	47.94	43.14	38.35
JUL	122.42	91.01	81.92	68.25	45.50	40.94	36.39
AUG	122.42	91.01	81.92	68.25	45.50	40.94	36.39
SEP	122.42	91.01	81.92	68.25	45.50	40.94	36.39
OCT	135.07	83.42	83.41	75.30	50.20	45.17	40.15
NOV	135.07	83.42	83.41	75.30	50.20	45.17	40.15
DEC	135.07	83.42	83.41	75.30	50.20	45.17	40.15

The national Energy Regulations only approves a price ceiling, implying that the actual price charged to customers should not exceed the maximum price. Actual price is determined based on contractual negotiations between a licensee and its customers, and the negotiated price should comply with section 22 of the Gas Act.

6.2 Effects of the steam tracing and plant downtime

Isegen has challenges of the frequent downtime on the foods plant section. Production is lost due to inefficient operation of the steam trace lines which use the steam supplied by the steam system.

When product in a pipeline is at a higher temperature than the air surrounding it, heat will pass through the wall of the pipeline from the product to the surrounding air. This heat loss will cause the temperature of the product to fall. Insulating the pipeline will lower the rate at which heat is lost, but unfortunately, no insulation is 100 % efficient. To make up the heat lost from the product pipeline, the steam tracer lines are attached to the product line. However, due to the condensate that was measured during the experiment from the steam mains and the heat loss that was predicted on the numerical analysis that would affect the steam tracing to be inefficient.

The blockages that occur in the product pipeline prevent the transport of substances and need to be removed on a regular basis to ensure smooth operation and that leads to downtime of the plant to clear the lines and also lost production due to the downtime. These cases normally occur at night when the ambient temperatures are low and to resolve the situations it normally takes the company two days. To clear the pipeline blockage, Isegen normally used the mechanical processes of gas flame heating and high-pressure cleaning, this method is used because it is much safer and it is effective against solid blockages.

6.2.1 Production loss

Based on the information that was provided by Isegen, the Foods plant produces 14 000 tons of Malic Acid a year, each ton cost about $R850{,}00$. According to Isegen, the average major downtimes that occur annually are 14 times. Therefore, a month the plant produces 1167 tonnes and a day produces about 38 tons of Malic Acid.

6.2.2 Daily production loss

$$Cost\ loss\ per\ day = 38\ tones\ x\ R850 = R32\ 300.00$$

6.2.3 Annually production loss due to a number of downtimes

$$Annually\ production\ loss = R32\ 300\ x\ 14\ = R452\ 200.00$$

6.3 Heat Loss in an Insulated Pipe

The expected heat loss through the insulated pipeline was obtained by inputting the basic information into the model shown in chapter four, such as the ambient temperature, wind speed, and material properties. In the model, the heat loss of the pipeline system was seen to increase as the insulation thickness decreased. This is in line with the general heat transfer theory relating to Fourier's Law in relation to cylindrical heat transfer.

6.3.1 Savings due to Insulation replacement

The annual heat energy loss cost from a pipe with 50 mm thick insulation compared with a 75 mm thick insulation was determined seen in Tables 6.2 and 6.3, assuming a system operating for 7320 hours a year. The current maximum price value of natural gas is $R135.07$ per GJ.

$$Annual\ Heat\ Energy\ loss\ Cost = heat\ loss\ x\ Fuel\ cost\ x\ Number\ of\ operating\ hours\ a\ year$$

Table 6.2: Heat Energy at 50mm thick of Insulation.

Thickness of Insulation	50 mm			
Wind Speed (m/s)	**Heat Loss (kW)**	**Fuel Cost per (R/kJ)**	**Number of Hours/year**	**Heat Energy Loss Cost (R/Year)**
3	35.54	0.00013507	7320	R126,489.22
4	36.46	0.00013507	7320	R129,777.09
5	37.15	0.00013507	7320	R132,230.86
6	37.70	0.00013507	7320	R134,175.77
7	38.15	0.00013507	7320	R135,776.99
8	38.53	0.00013507	7320	R137,130.33
9	38.85	0.00013507	7320	R138,296.44
10	39.14	0.00013507	7320	R139,316.29

Table 6.3: Heat Energy at 75mm thick of Insulation.

Thickness of Insulation	75mm			
Wind Speed (m/s)	**Heat Loss (kW)**	**Fuel Cost per (R/kJ)**	**Number of Hours/year**	**Heat Energy Loss Cost (R/Year)**
3	27.66	0.00013507	7320	R98,462.26
4	28.38	0.00013507	7320	R101,007.10
5	28.93	0.00013507	7320	R102,979.23
6	29.38	0.00013507	7320	R104,584.00
7	29.76	0.00013507	7320	R105,930.91
8	30.09	0.00013507	7320	R107,086.17
9	30.37	0.00013507	7320	R108,093.20
10	30.62	0.00013507	7320	R108,982.18

$$Thickness\ Cost\ Saving = 50\ mm\ heat\ energy\ loss\ cost - 75\ mm\ heat\ energy\ loss\ cost$$

Table 6.4: Total energy saving due to insulation results.

Wind Speed (m/s)	50mm Heat Energy Loss Cost (R/Year)	75mm Heat Energy Loss Cost (R/Year)	Thickness Cost Saving (R/Year)	Difference Percentage (%)
3	R126,489.22	R98,462.26	R28,026.96	22
4	R129,777.09	R101,007.10	R28,769.99	22
5	R132,230.86	R102,979.23	R29,251.63	22
6	R134,175.77	R104,584.00	R29,591.76	22
7	R135,776.99	R105,930.91	R29,846.08	22
8	R137,130.33	R107,086.17	R30,044.16	22
9	R138,296.44	R108,093.20	R30,203.24	22
10	R139,316.29	R108,982.18	R30,334.12	22
Average			**R29,508.49**	**22**

This measure can be to use thicker insulating material or to make a careful analysis of the proper insulation material. Crucial factors in choosing insulating material including a low thermal conductivity, dimensional stability under temperature change, resistance to water absorption, and resistance to combustion. Improving the insulation on the existing pipeline of heat distribution systems could save an average of 22 % of energy loss costs due to thicker insulation.

6.4 Upgrade cost proposed project

It was recommended to Isegen that it would be ideal to implement the upgrades on the existing steam pipeline in order to minimise the heat loss and the condensate. Each improvement implemented will affect the savings realised by the other improvements. The steam pipeline upgrade proposal cost is shown in the table below, the more detailed documents are provided on the relevant appendixes as it was recommended in chapter 5.

Table 6.5: Steam pipeline upgrade proposal cost.

Item	Work	Reference	Cost
1	Steam Pipeline inspection and design work	**Appendix N**	R277 357.00
2	Steam pipeline upgrade	**Appendix O**	R141 174.00
3	Cost of insulation and cladding installation	**Appendix P**	R275 632.00
The total cost of the upgrade			**R697 163.00**

6.5 Payback period

The payback period has been determined after carefully considering all the important factors that caused the heat loss and production losses, however, it was very important to consult other experts to assist with the cost in upgrading the steam pipeline system. Thus, the payback period is the simplest method to assess the risk associated with the investment and the time required to get the initial outlay recovered. The payback period is calculated as follows,

$$Annual\ total\ savings = Insulation\ replacement + Production\ lost$$

$$Annual\ total\ savings = R29\ 508{,}49 + R452\ 200{,}00$$

$$Annual\ total\ savings = R481\ 708{,}49$$

Therefore,

$$Payback\ period\ (years) = Cost\ of\ project/\ Annual\ total\ savings$$

$$Payback\ period\ (years) = R697\ 163{,}00/R481\ 708{,}49$$

$$Payback\ period\ (years) = 1{,}45\ years$$

Isegen will be recovered in approximately one and a half years which seems a reasonable payback duration for the type of investment.

6.6 Results Discussions

Accurately determining energy saving due to insulation is important for monitoring and managing energy use in a plant, for evaluating proposed design changes to the generation of steam and for continuing to identify competitive advantages through plant efficiency improvements. By comparing the current pipeline system and the model prediction system on the insulation energy results, a 22 % difference was found. As a result, energy can be saved by increasing the thickness of the insulation from 50 to 75 mm.

The main opportunities for improvement were found to be inefficient operation of the steam trace lines and steam pipeline modification, improvement of the insulation thickness and the estimated direct cost is approximately about $R697\ 163.00$, However, by implementing this upgrade proposal will result in improving economic performance for businesses through sustainable business practices.

The energy savings and efficient condensate recovery are very vital for Isegen SA (Pty) Ltd since most of the heating systems are indirect steam. This means a large quantity of the sensible heat energy of the steam distributed into the factory is not fully recovered due to the leaks on the condensate system. Therefore, it is necessary for Isegen to implement this study to save fuel consumption and production lost and also steam usage.

Chapter Seven

7.1 Conclusions

This study aimed to investigate the effect of the atmospheric conditions on the temperature drop across heat treatment systems. A numerical analysis model was developed to investigate the heat energy losses and condensate generation during the transport of steam through the plants main insulated steam pipeline, between the boiler house and process area of the plant. The experimental data was also captured for comparison between the model prediction condensate and the measured condensate.

The total heat loss along the $250\,m$ long saturated steam pipeline calculations were successfully obtained and were compared to the Spirax Sarco data. During the analysis of the effects of the ambient temperature and wind speed on the heat loss of the pipeline, it was proven that as the ambient temperature decreased the heat loss increased, and as the ambient temperature increased the heat loss decreased. It can also be seen that as the wind speed increased the heat loss increased. Furthermore, the effects of the insulation on the heat loss of the pipeline system were also investigated. The model shows that the heat loss of the pipeline system increases as the insulation thickness decreases and also as the wind speed increases. Furthermore, if the thickness of the insulation increases the heat loss decreases and that is resulting in less heat transfer to the ambient air. Based on the model prediction results it was recommended to use $75\,mm$ thickness of insulation to minimise the heat losses.

The condensate rate was physically measured during the experiment and was also investigated in the numerical model data. The model's predicted condensate was matching up with the Spirax Sarco published data with only a $0.12\,\%$ difference. However, the condensate that was measured during the experiment was found to be 2.9 times compared to the model prediction condensate rate data. It can be seen that the condensate measured data values and the numerical prediction do not correlate.

The cost analysis conducted was based on the heat energy loss and production lost. An in-depth analysis has been carried out to observe insight into factors that influence capital budgeting decisions. The results of the survey and its analysis have been provided in chapter 6. The cost analysis represents a reasonable payback period of approximately 1.45 years. This led Isegen SA (Pty) Ltd to consider implementing this research in their factory **(see Appendix R for the Management of Change Form).** Our research suggests that even a proactive maintenance program can save considerable amounts of energy.

7.2 Future work

The present study of investigating the effect of the atmospheric conditions on the temperature drop across heat treatment systems (piping and vessels) can be extended to study further in future. The work initiated in this thesis can be continued in various directions that may be generally classified as experimental infrastructure improvement and development, as well as experimental validation methodology and performance. This future work can expand the scope of the experimental studies and complete the validation of the mathematical models.

The improvements and future developments of the experimental infrastructure can be summarized in the following guidelines.

- Data on the condition of the steam system and the return of condensation should be conducted in at least three similar plants.
- Improved water management.
- Chemical savings.

The savings estimated in the study were cumulative and based upon the operating parameters stated. Each improvement implemented will affect the savings realized by the other improvements. When evaluated alone, each improvement concept would likely have an even bigger individual saving than calculated in this study.

It is recommended to bring to action the methodology that has been proposed in chapter five. This methodology is applicable to steam efficiency improvement project activities with the following conditions

- Steam efficiency is improved by replacement and/or repair of a $15\ \mathrm{mm}$ drain pipe connected directly to the bottom of a main by an at least $50\ \mathrm{mm}$ for larger mains and this allows space below for any dirt and scale to settle and the return collection and re-utilization of condensation.
- The regular maintenance of steam traps or the return of condensation is not common practice or required under regulations in the respective country.
- Data on the condition of steam traps and the return of condensation is accessible in at least three similar other plants.
- Replace the insulation and the cladding sheets.

For more information regarding the proposed methodology and its consideration please see in chapter five from 5.6 to 5.8. This baseline methodology shall be used in conjunction with the approved monitoring methodology steam system efficiency improvements by replacing steam smaller drain pipeline from the main with the bottom of the pocket that may be fitted with a removable flange or blowdown valve for cleaning purposes.

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APPENDICES

Appendix A: The thermal conductivity of various materials

Table 14: Table 20A: The thermal conductivity of various materials (http://www.spiraxsarco.com).

Material	Thermal conductivity (W/m K)
Copper (pure)	399
Gold (pure)	317
Aluminum (pure)	237
Iron (pure)	80.2
Carbon steel (1 %)	43
Stainless Steel (18/8)	15.1
Glass	0.81
Plastics	0.2 – 0.3
Wood (shredded/cemented)	0.087
Cork	0.039
Water (liquid)	0.6
Ethylene glycol (liquid)	0.26
Hydrogen (gas)	0.18
Benzene (liquid)	0.159
Air	0.026

Appendix B: Reference Data Steam Condensation

According to Spirax Sarco, the steam condensate for 100 m insulated steam main pipeline can be expressed as follows:

$$\dot{m}_s = \frac{3.6\,\dot{Q}Lf}{h_{fg}}\ (kg/h)$$

For further reference: https://www.spiraxsarco.com/learn-about-steam/steam-engineering-principles-and-heat-transfer/steam-consumption-of-pipes-and-air-heaters#article-top

Steam condense as heat is lost from the pipe to the environment. The rate of condensation depends on the steam temperature, the ambient temperature and the efficiency of the lagging.

Input	**Value**	**Units**
$\dot{Q}$=	1925	W/m
The Factor that provides a solution in kg/h=	3.6	
L=	100.6	m
f=	0.07	
h_{fg}	1945	kJ/kg
$\dot{m}_s$	**25.09**	**kg/h**

Appendix C: The properties of dry air at low pressure

Table 15: The properties of dry air at low pressure (Rogers and Mayhew, 1995)

T [K]	c_p [kJ/kg K]	c_v [kJ/kg K]		μ 10^{-5}[kg/m s]	k 10^{-5}[kW/m K]	Pr	at 1 atm ρ [kg/m^3]	at 1 atm ν 10^{-5}[m^2/s]
175	1.0023	0.7152	1.401	1.182	1.593	0.744	2.017	0.586
200	1.0025	0.7154	1.401	1.329	1.809	0.736	1.765	0.753
225	1.0027	0.7156	1.401	1.467	2.020	0.728	1.569	0.935
250	1.0031	0.7160	1.401	1.599	2.227	0.720	1.412	1.132
275	1.0038	0.7167	1.401	1.725	2.428	0.713	1.284	1.343
300	1.0049	0.7178	1.400	1.846	2.624	0.707	1.177	1.568
325	1.0063	0.7192	1.400	1.962	2.816	0.701	1.086	1.807
350	1.0082	0.7211	1.398	2.075	3.003	0.697	1.009	2.056
375	1.0106	0.7235	1.397	2.181	3.186	0.692	0.9413	2.317
400	1.0135	0.7264	1.395	2.286	3.365	0.688	0.8824	2.591
450	1.0206	0.7335	1.391	2.485	3.710	0.684	0.7844	3.168
500	1.0295	0.7424	1.387	2.670	4.041	0.680	0.7060	3.782
550	1.0398	0.7527	1.381	2.849	4.357	0.680	0.6418	4.439
600	1.0511	0.7640	1.376	3.017	4.661	0.680	0.5883	5.128
650	1.0629	0.7758	1.370	3.178	4.954	0.682	0.5430	5.853
700	1.0750	0.7879	1.364	3.332	5.236	0.684	0.5043	6.607
750	1.0870	0.7999	1.359	3.482	5.509	0.687	0.4706	7.399
800	1.0987	0.8116	1.354	3.624	5.774	0.690	0.4412	8.214
850	1.1101	0.8230	1.349	3.763	6.030	0.693	0.4153	9.061
900	1.1209	0.8338	1.344	3.897	6.276	0.696	0.3922	9.936
950	1.1313	0.8442	1.340	4.026	6.520	0.699	0.3716	10.83
1000	1.1411	0.8540	1.336	4.153	6.754	0.702	0.3530	11.76
1050	1.1502	0.8631	1.333	4.276	6.985	0.704	0.3362	12.72
1100	1.1589	0.8718	1.329	4.396	7.209	0.707	0.3209	13.70
1150	1.1670	0.8799	1.326	4.511	7.427	0.709	0.3069	14.70
1200	1.1746	0.8875	1.323	4.626	7.640	0.711	0.2941	15.73
1250	1.1817	0.8946	1.321	4.736	7.849	0.713	0.2824	16.77
1300	1.1884	0.9013	1.319	4.846	8.054	0.715	0.2715	17.85
1350	1.1946	0.9075	1.316	4.952	8.253	0.717	0.2615	18.94
1400	1.2005	0.9134	1.314	5.057	8.450	0.719	0.2521	20.06
1500	1.2112	0.9241	1.311	5.264	8.831	0.722	0.2353	22.36
1600	1.2207	0.9336	1.308	5.457	9.199	0.724	0.2206	24.74
1700	1.2293	0.9422	1.305	5.646	9.554	0.726	0.2076	27.20
1800	1.2370	0.9499	1.302	5.829	9.899	0.728	0.1961	29.72
1900	1.2440	0.9569	1.300	6.008	10.233	0.730	0.1858	32.34
2000	1.2505	0.9634	1.298	—	—	—	0.1765	—
2100	1.2564	0.9693	1.296	—	—	—	0.1681	—
2200	1.2619	0.9748	1.295	—	—	—	0.1604	—
2300	1.2669	0.9798	1.293	—	—	—	0.1535	—
2400	1.2717	0.9846	1.292	—	—	—	0.1471	—
2500	1.2762	0.9891	1.290	—	—	—	0.1412	—
2600	1.2803	0.9932	1.289	—	—	—	0.1358	—
2700	1.2843	0.9972	1.288	—	—	—	0.1307	—
2800	1.2881	1.0010	1.287	—	—	—	0.1261	—
2900	1.2916	1.0045	1.286	—	—	—	0.1217	—
3000	1.2949	1.0078	1.285	—	—	—	0.1177	—

Appendix D: The properties of water and steam

Table 16: The properties of water and steam (Rogers and Mayhew, 1995)

T [°C]	p_s [bar]	v_f 10^{-2}[m³/kg]	c_{pf} [kJ/kg K]	c_{pg} [kJ/kg K]	μ_f 10^{-6}[kg/m s]	μ_g 10^{-6}[kg/m s]	k_f 10^{-6}[kW/m K]	k_g 10^{-6}[kW/m K]	$(Pr)_f$	$(Pr)_g$
0.01	0.006112	0.10002	4.210	1.86	1752	8.49	569	16.3	12.96	0.97
5	0.008719	0.10001	4.204	1.86	1501	8.66	578	16.7	10.92	0.96
10	0.01227	0.10003	4.193	1.86	1300	8.83	587	17.1	9.29	0.96
15	0.01704	0.10010	4.186	1.87	1136	9.00	595	17.5	7.99	0.96
20	0.02337	0.10018	4.183	1.87	1002	9.18	603	17.9	6.95	0.96
25	0.03166	0.10030	4.181	1.88	890	9.35	611	18.3	6.09	0.96
30	0.04242	0.10044	4.179	1.88	797	9.52	618	18.7	5.39	0.96
35	0.05622	0.10060	4.178	1.88	718	9.70	625	19.1	4.80	0.96
40	0.07375	0.10079	4.179	1.89	651	9.87	632	19.5	4.30	0.96
45	0.09582	0.10099	4.181	1.89	594	10.0	638	19.9	3.89	0.95
50	0.1233	0.1012	4.182	1.90	544	10.2	643	20.4	3.54	0.95
55	0.1574	0.1015	4.183	1.90	501	10.4	648	20.8	3.23	0.95
60	0.1992	0.1017	4.185	1.91	463	10.6	653	21.2	2.97	0.95
65	0.2501	0.1020	4.188	1.92	430	10.7	658	21.6	2.74	0.95
70	0.3116	0.1023	4.191	1.93	400	10.9	662	22.0	2.53	0.96
75	0.3855	0.1026	4.194	1.94	374	11.1	666	22.5	2.36	0.96
80	0.4736	0.1029	4.198	1.95	351	11.3	670	22.9	2.20	0.96
85	0.5780	0.1032	4.203	1.96	330	11.4	673	23.3	2.06	0.96
90	0.7011	0.1036	4.208	1.97	311	11.6	676	23.8	1.94	0.96
95	0.8453	0.1040	4.213	1.99	294	11.8	678	24.3	1.83	0.97
100	1.01325	0.1044	4.219	2.01	279	12.0	681	24.8	1.73	0.97
105	1.208	0.1048	4.226	2.03	265	12.2	683	25.3	1.64	0.98
110	1.433	0.1052	4.233	2.05	252	12.4	684	25.8	1.56	0.99
115	1.691	0.1056	4.240	2.07	241	12.6	686	26.3	1.49	0.99
120	1.985	0.1060	4.248	2.09	230	12.8	687	26.8	1.42	1.00
125	2.321	0.1065	4.26	2.12	220	13.0	687	27.3	1.36	1.01
130	2.701	0.1070	4.27	2.15	211	13.2	688	27.8	1.31	1.02
135	3.131	0.1075	4.28	2.18	203	13.4	688	28.3	1.26	1.03
140	3.614	0.1080	4.29	2.21	195	13.5	688	28.8	1.22	1.04
145	4.155	0.1085	4.30	2.25	188	13.7	687	29.4	1.18	1.05
150	4.760	0.1091	4.32	2.29	181	13.9	687	30.0	1.14	1.07
160	6.181	0.1102	4.35	2.38	169	14.2	684	31.3	1.07	1.09
170	7.920	0.1114	4.38	2.49	159	14.6	681	32.6	1.02	1.12
180	10.03	0.1128	4.42	2.62	149	15.0	676	34.1	0.97	1.15
190	12.55	0.1142	4.46	2.76	141	15.3	671	35.7	0.94	1.18
200	15.55	0.1157	4.51	2.91	134	15.7	665	37.5	0.91	1.22
210	19.08	0.1173	4.56	3.07	127	16.0	657	39.4	0.88	1.25
220	23.20	0.1190	4.63	3.25	121	16.3	648	41.5	0.86	1.28
230	27.98	0.1209	4.70	3.45	116	16.7	639	43.9	0.85	1.31
240	33.48	0.1229	4.78	3.68	111	17.1	628	46.5	0.84	1.35
250	39.78	0.1251	4.87	3.94	107	17.5	616	49.5	0.85	1.39
260	46.94	0.1276	4.98	4.22	103	17.9	603	52.8	0.85	1.43
270	55.05	0.1302	5.10	4.55	99	18.3	589	56.6	0.86	1.47
280	64.19	0.1332	5.24	4.98	96	18.8	574	61.0	0.88	1.53
290	74.45	0.1366	5.42	5.46	93	19.3	558	66.0	0.90	1.60
300	85.92	0.1404	5.65	6.18	90	19.8	541	72.0	0.94	1.70
320	112.9	0.1499								
340	146.1	0.1639								
360	186.7	0.1894								
370	210.5	0.2225								
374.15	221.2	0.317								

Appendix E: The steam table

Table 17E: The steam table (Rogers and Mayhew, 1995)

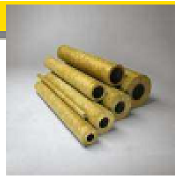
Saturated Water and Steam

p [bar]	T_s [°C]	v_g [m³/kg]	u_f [kJ/kg]	u_g [kJ/kg]	h_f [kJ/kg]	h_{fg} [kJ/kg]	h_g [kJ/kg]	s_f [kJ/kg K]	s_{fg} [kJ/kg K]	s_g [kJ/kg K]
1.0	99.6	1.694	417	2506	417	2258	2675	1.303	6.056	7.359
1.1	102.3	1.549	429	2510	429	2251	2680	1.333	5.994	7.327
1.2	104.8	1.428	439	2512	439	2244	2683	1.361	5.937	7.298
1.3	107.1	1.325	449	2515	449	2238	2687	1.387	5.884	7.271
1.4	109.3	1.236	458	2517	458	2232	2690	1.411	5.835	7.246
1.5	111.4	1.159	467	2519	467	2226	2693	1.434	5.789	7.223
1.6	113.3	1.091	475	2521	475	2221	2696	1.455	5.747	7.202
1.7	115.2	1.031	483	2524	483	2216	2699	1.475	5.707	7.182
1.8	116.9	0.9774	491	2526	491	2211	2702	1.494	5.669	7.163
1.9	118.6	0.9292	498	2528	498	2206	2704	1.513	5.632	7.145
2.0	120.2	0.8856	505	2530	505	2202	2707	1.530	5.597	7.127
2.1	121.8	0.8461	511	2531	511	2198	2709	1.547	5.564	7.111
2.2	123.3	0.8100	518	2533	518	2193	2711	1.563	5.533	7.096
2.3	124.7	0.7770	524	2534	524	2189	2713	1.578	5.503	7.081
2.4	126.1	0.7466	530	2536	530	2185	2715	1.593	5.474	7.067
2.5	127.4	0.7186	535	2537	535	2182	2717	1.607	5.446	7.053
2.6	128.7	0.6927	541	2539	541	2178	2719	1.621	5.419	7.040
2.7	130.0	0.6686	546	2540	546	2174	2720	1.634	5.393	7.027
2.8	131.2	0.6462	551	2541	551	2171	2722	1.647	5.368	7.015
2.9	132.4	0.6253	556	2543	556	2168	2724	1.660	5.344	7.004
3.0	133.5	0.6057	561	2544	561	2164	2725	1.672	5.321	6.993
3.5	138.9	0.5241	584	2549	584	2148	2732	1.727	5.214	6.941
4.0	143.6	0.4623	605	2554	605	2134	2739	1.776	5.121	6.897
4.5	147.9	0.4139	623	2558	623	2121	2744	1.820	5.037	6.857
5.0	151.8	0.3748	639	2562	640	2109	2749	1.860	4.962	6.822
5.5	155.5	0.3427	655	2565	656	2097	2753	1.897	4.893	6.790
6	158.8	0.3156	669	2568	670	2087	2757	1.931	4.830	6.761
7	165.0	0.2728	696	2573	697	2067	2764	1.992	4.717	6.709
8	170.4	0.2403	720	2577	721	2048	2769	2.046	4.617	6.663
9	175.4	0.2149	742	2581	743	2031	2774	2.094	4.529	6.623
10	179.9	0.1944	762	2584	763	2015	2778	2.138	4.448	6.586
11	184.1	0.1774	780	2586	781	2000	2781	2.179	4.375	6.554
12	188.0	0.1632	797	2588	798	1986	2784	2.216	4.307	6.523
13	191.6	0.1512	813	2590	815	1972	2787	2.251	4.244	6.495
14	195.0	0.1408	828	2593	830	1960	2790	2.284	4.185	6.469
15	198.3	0.1317	843	2595	845	1947	2792	2.315	4.130	6.445
16	201.4	0.1237	857	2596	859	1935	2794	2.344	4.078	6.422
17	204.3	0.1167	870	2597	872	1923	2795	2.372	4.028	6.400
18	207.1	0.1104	883	2598	885	1912	2797	2.398	3.981	6.379
19	209.8	0.1047	895	2599	897	1901	2798	2.423	3.936	6.359
20	212.4	0.09957	907	2600	909	1890	2799	2.447	3.893	6.340
22	217.2	0.09069	928	2601	931	1870	2801	2.492	3.813	6.305
24	221.8	0.08323	949	2602	952	1850	2802	2.534	3.738	6.272
26	226.0	0.07689	969	2603	972	1831	2803	2.574	3.668	6.242
28	230.0	0.07142	988	2603	991	1812	2803	2.611	3.602	6.213
30	233.8	0.06665	1004	2603	1008	1795	2803	2.645	3.541	6.186
32	237.4	0.06246	1021	2603	1025	1778	2803	2.679	3.482	6.161
34	240.9	0.05875	1038	2603	1042	1761	2803	2.710	3.426	6.136
36	244.2	0.05544	1054	2602	1058	1744	2802	2.740	3.373	6.113
38	247.3	0.05246	1068	2602	1073	1729	2802	2.769	3.322	6.091
40	250.3	0.04977	1082	2602	1087	1714	2801	2.797	3.273	6.070

Appendix F: The Insulation Thermal Conductivity of Mineral wool

Table 18F: The insulation thermal conductivity of mineral wool (www.isover.co.za).

U Thermo Pipe



Thermal conductivity according to EN 12 667

T [℃]	50	100	150	200	300	400
λ [W/(m.K)]	**0,037**	**0,043**	**0,051**	**0,060**	**0,083**	**0,111**

Maximum service temperature

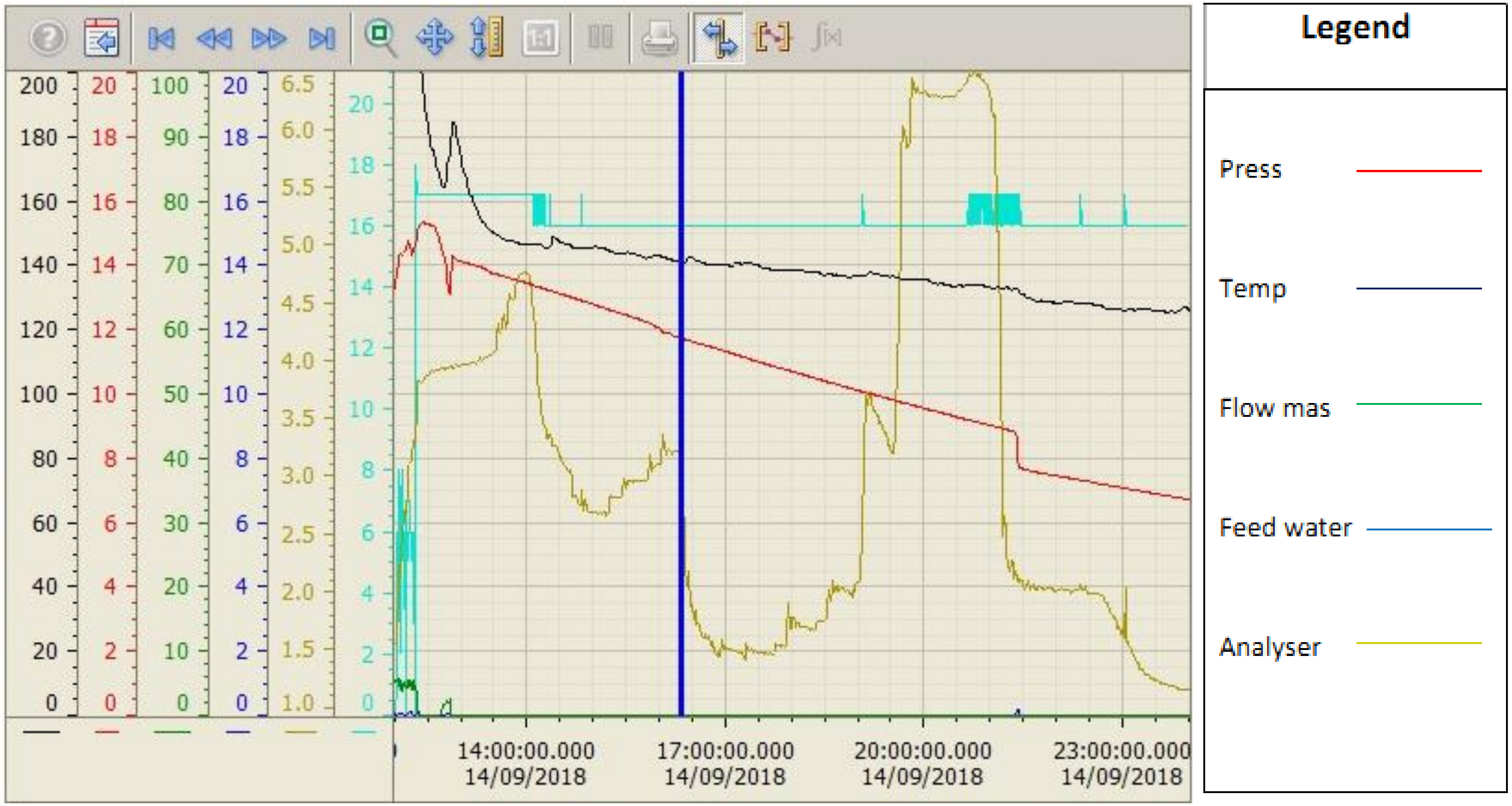
T_{max} = 660℃ under 500 Pa according to EN 14 706

Non combustible, fire class A1 according to EN 13 501

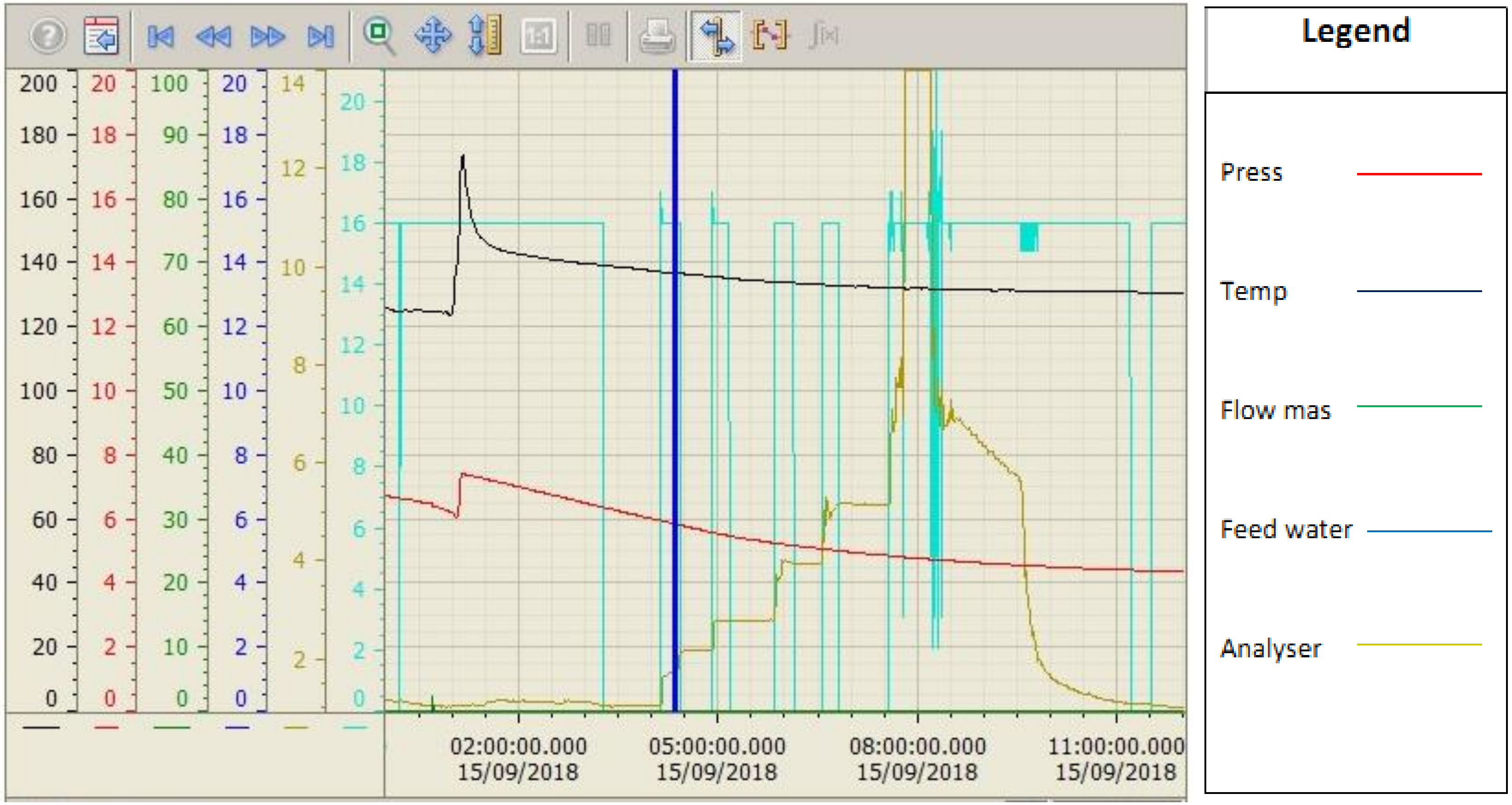
Non hazardous to health EUCEB certificate

ISOVER SAINT-GOBAIN

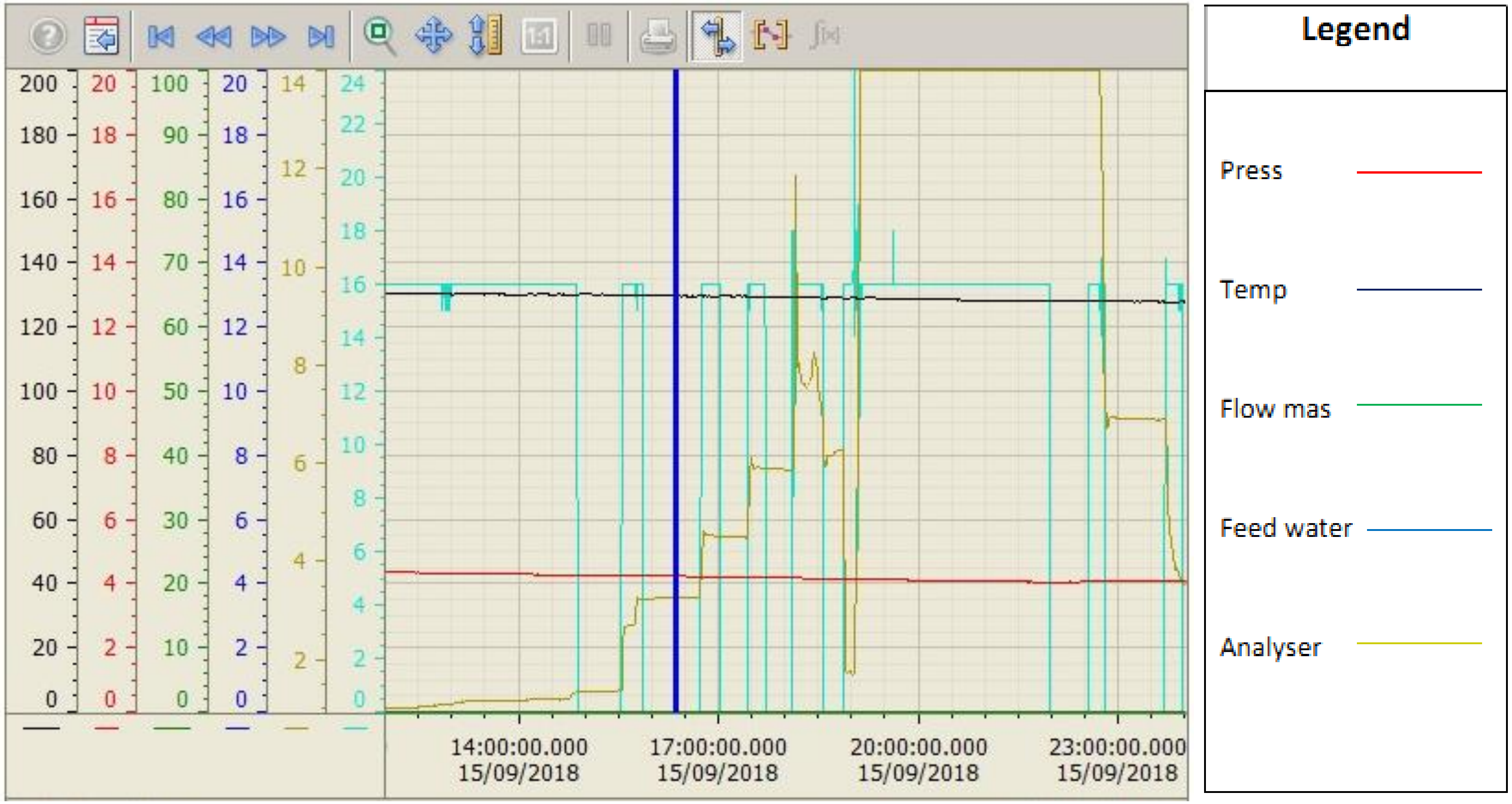
Appendix G: The boiler trend overview of the flow pressure and operating temperature; steam flow On the 14/09/2018 between 14:00 to 23:00, (Siemens Win CC of Isegen SA Pty Ltd).



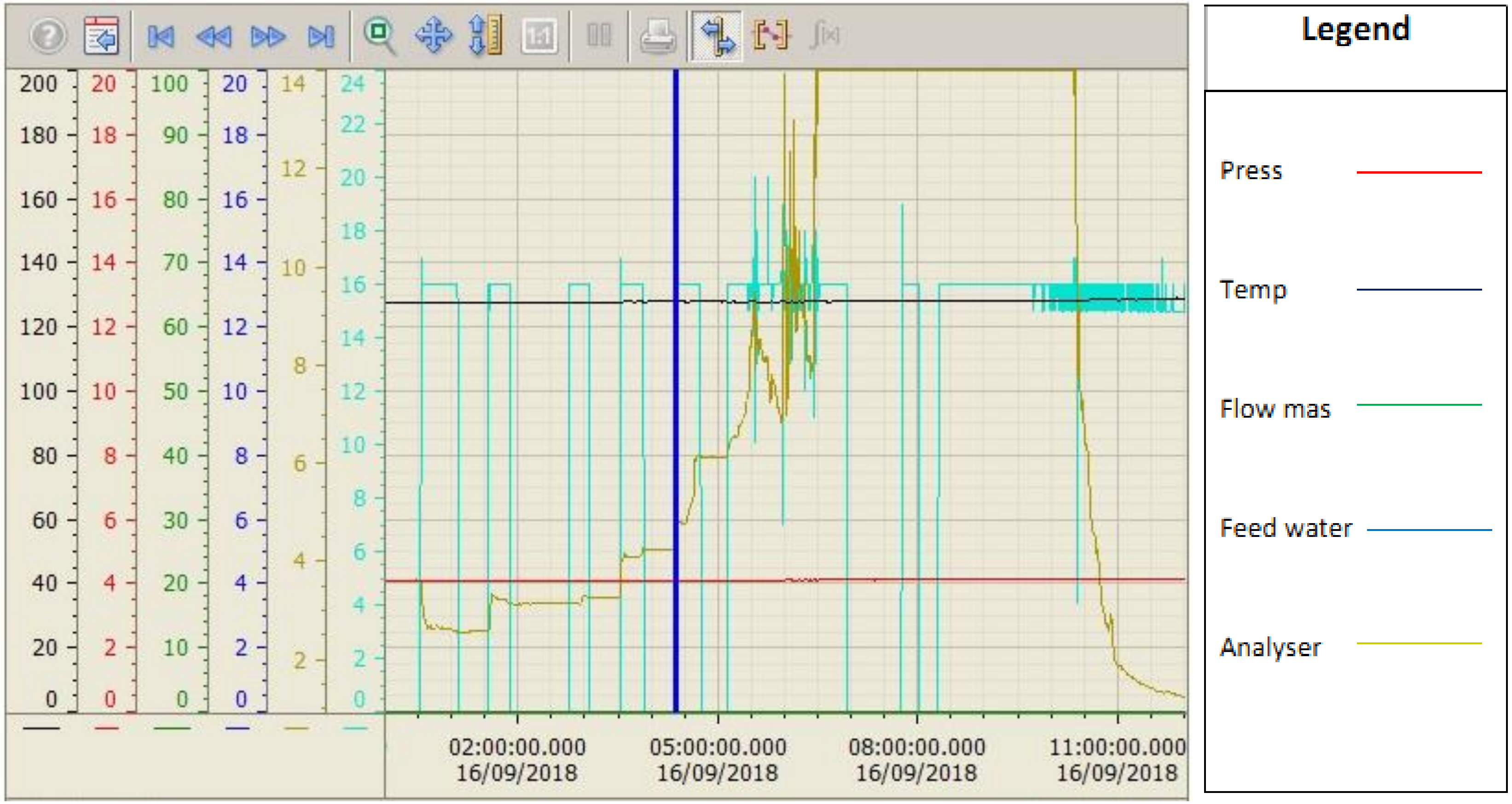
Appendix G: The boiler trend overview of the flow pressure and operating temperature; steam flow On the 15/09/2018 between 02:00 to 11:00, (Siemens Win CC of Isegen SA Pty Ltd).



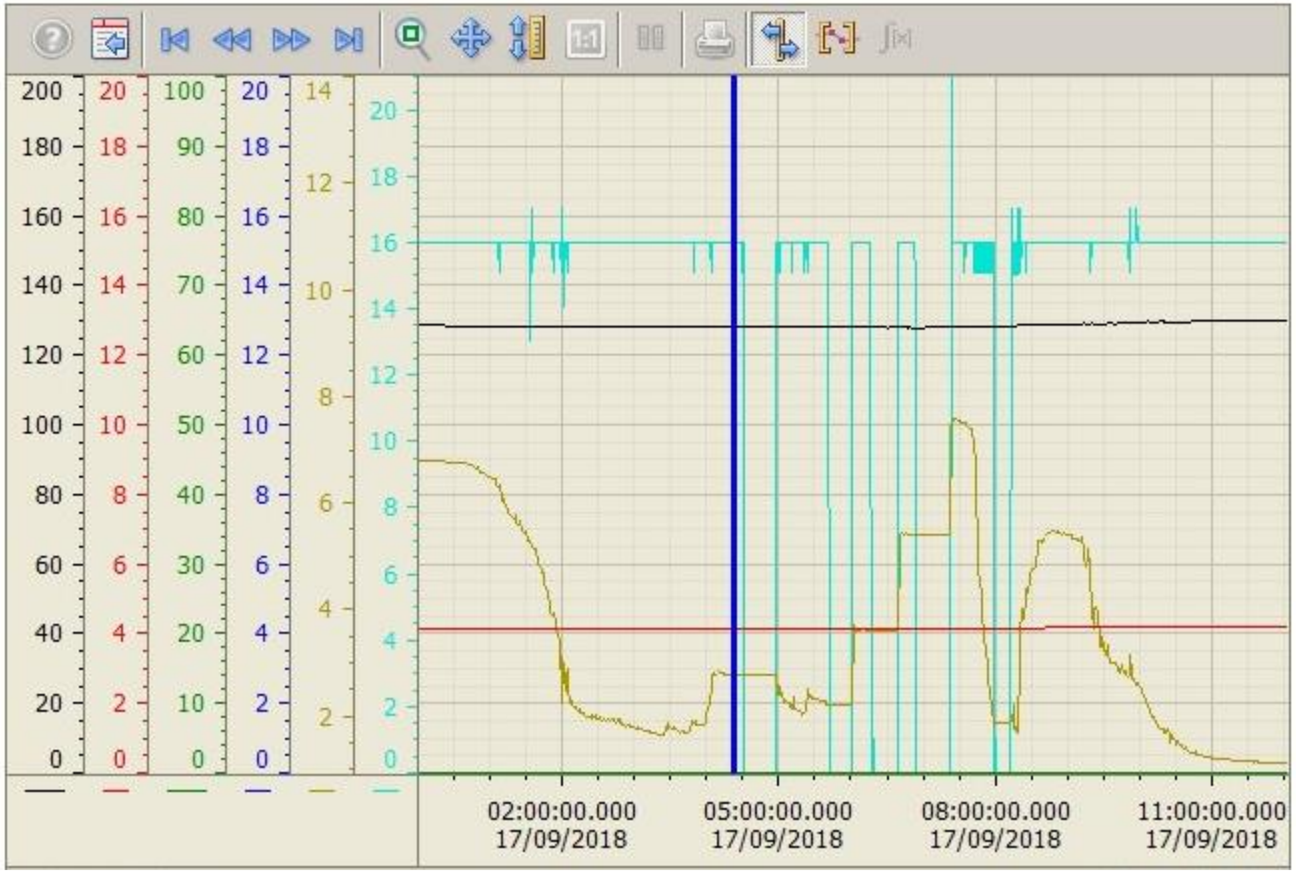
Appendix G: The boiler trend overview of the flow pressure and operating temperature; steam flow On the 15/09/2018 between 14:00 to 23:00, (Siemens Win CC of Isegen SA Pty Ltd).

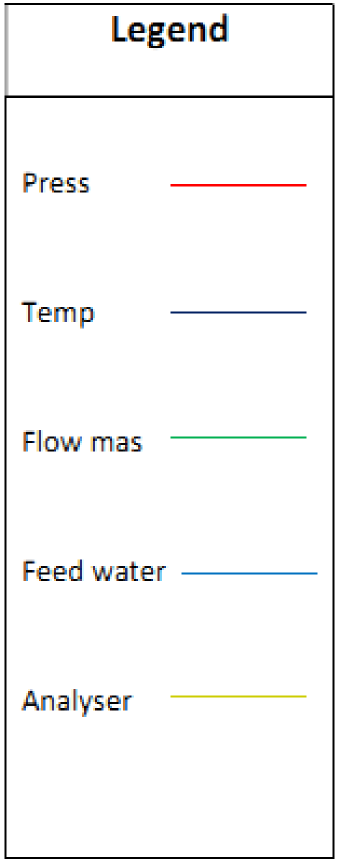


Appendix G: The boiler trend overview of the flow pressure and operating temperature; steam flow On the 16/09/2018 between 02:00 to 11:00, (Siemens Win CC of Isegen SA Pty Ltd).

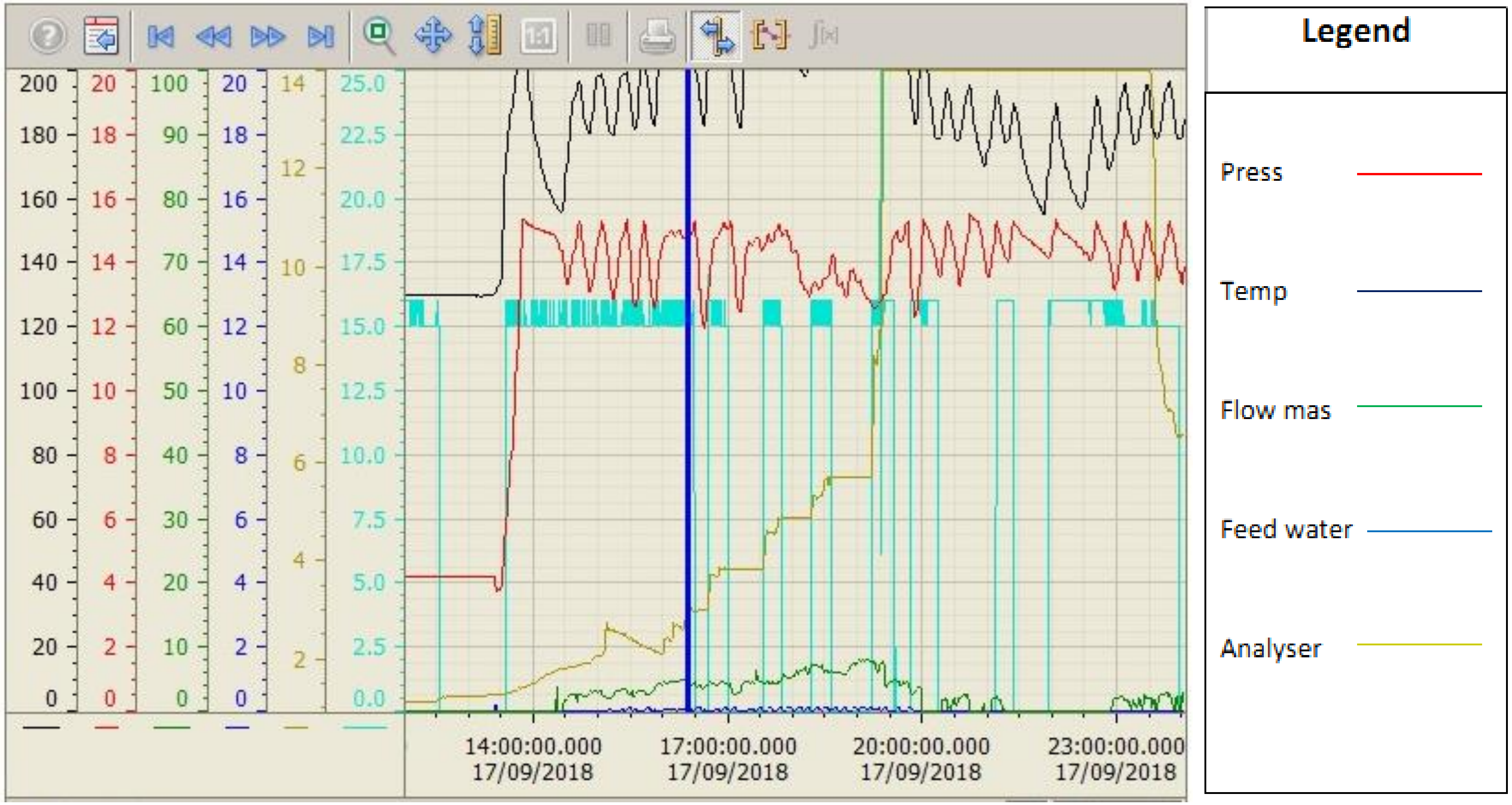


Appendix G: The boiler trend overview of the flow pressure and operating temperature; steam flow On the 17/09/2018 between 02:00 to 11:00, (Siemens Win CC of Isegen SA Pty Ltd).

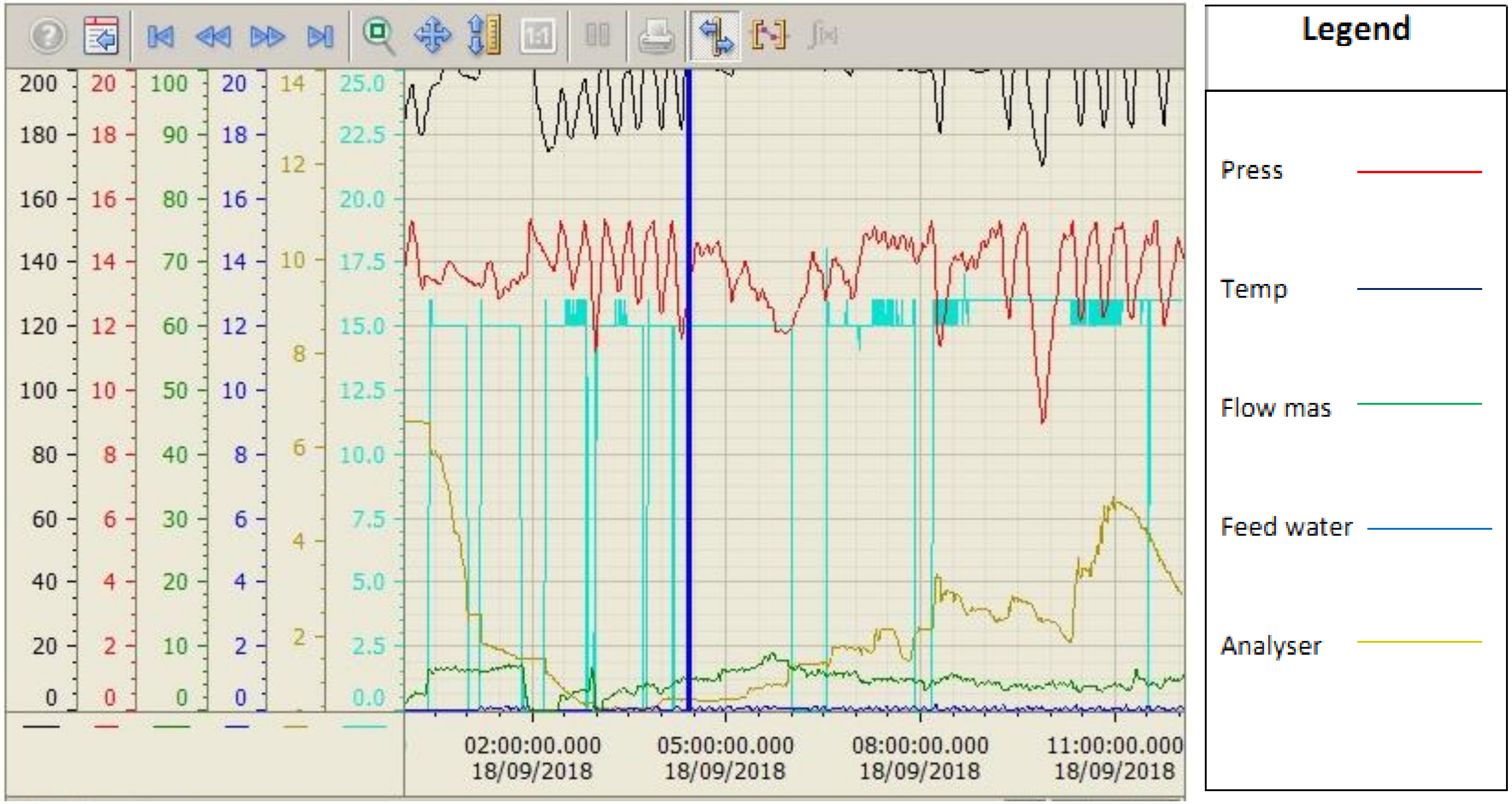




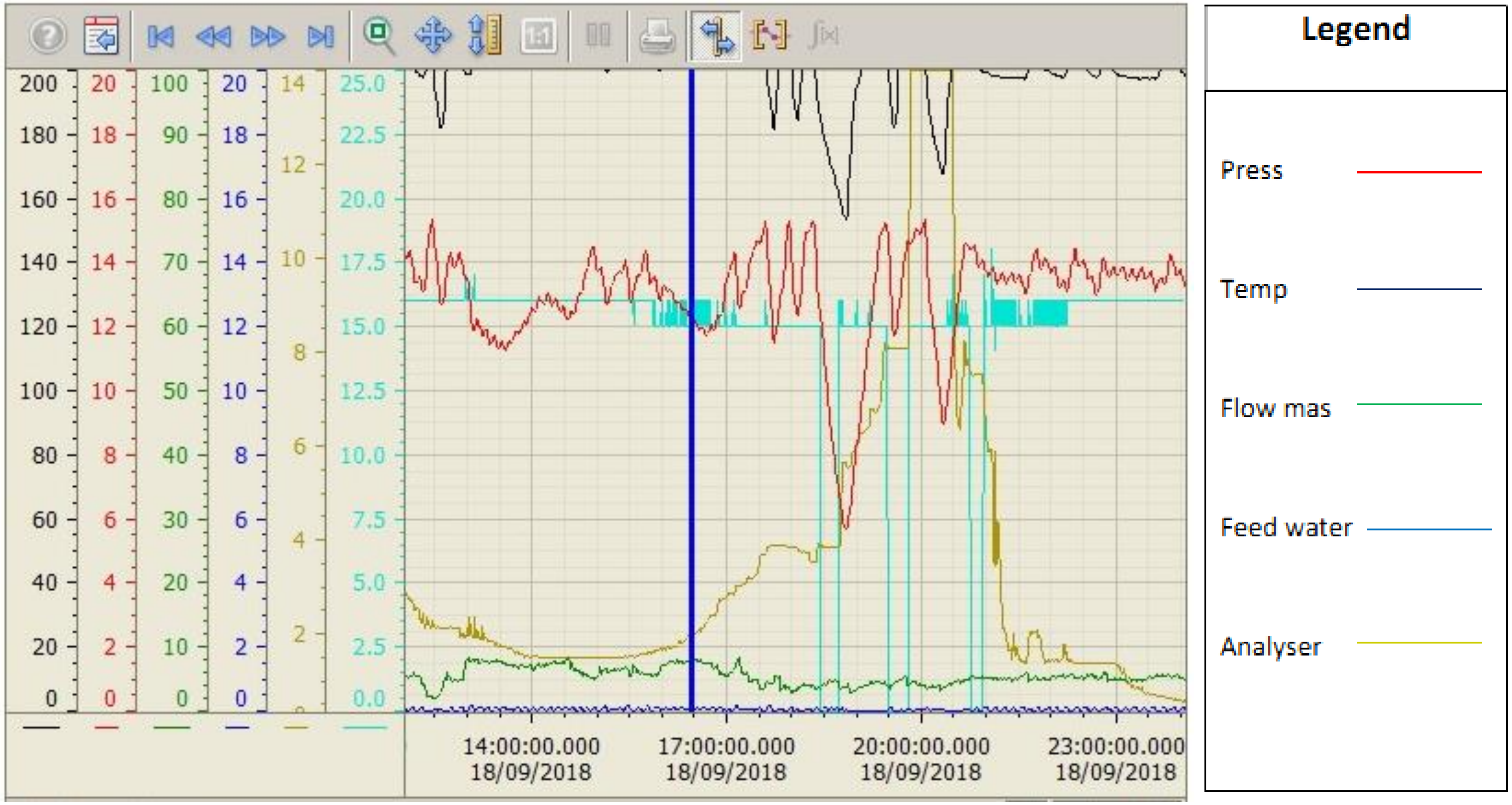
Appendix G: The boiler trend overview of the flow pressure and operating temperature; steam flow On the 17/09/2018 between 14:00 to 23:00, (Siemens Win CC of Isegen SA Pty Ltd).



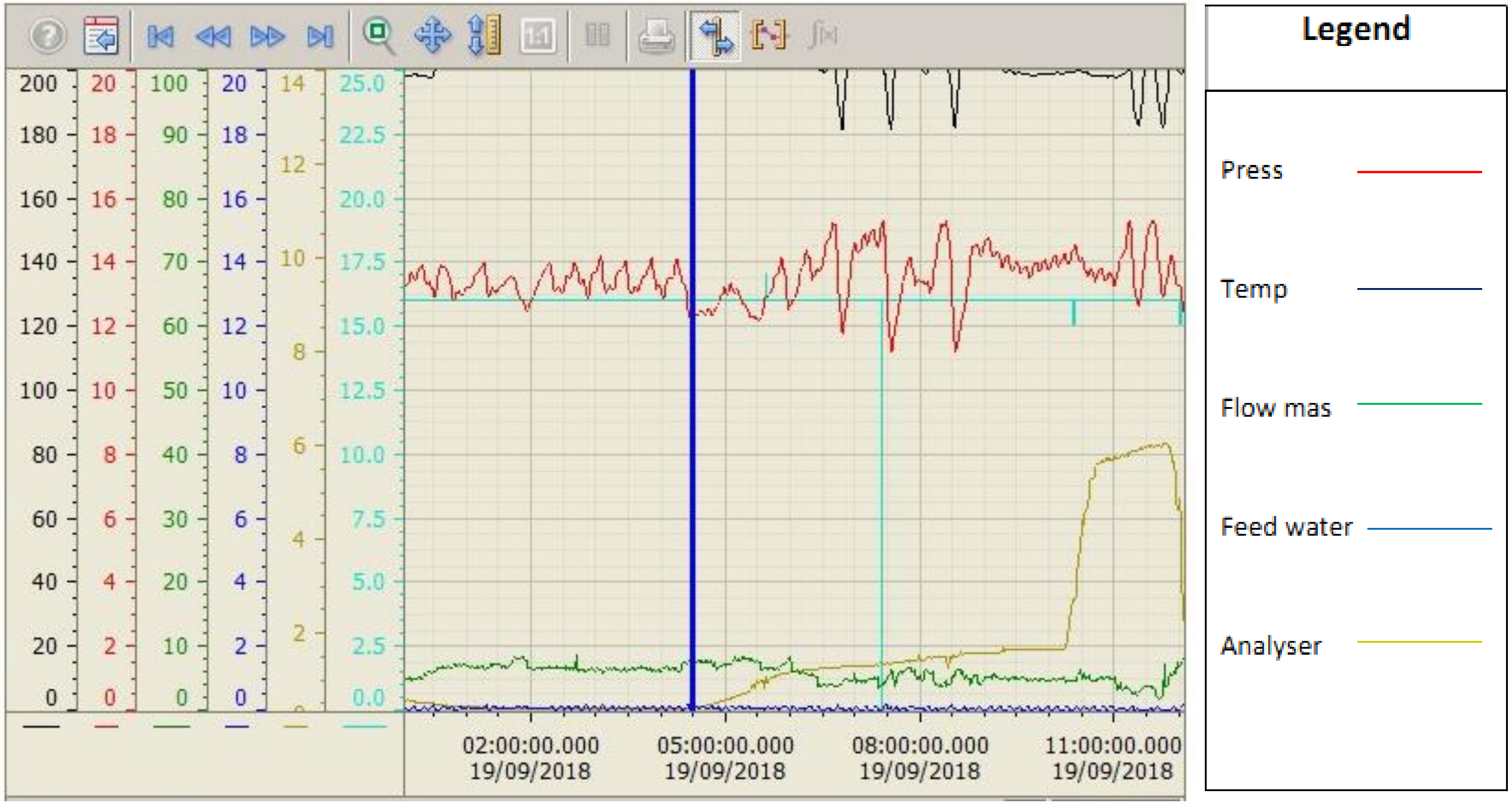
Appendix G: The boiler trend overview of the flow pressure and operating temperature; steam flow On the 18/09/2018 between 02:00 to 11:00, (Siemens Win CC of Isegen SA Pty Ltd).



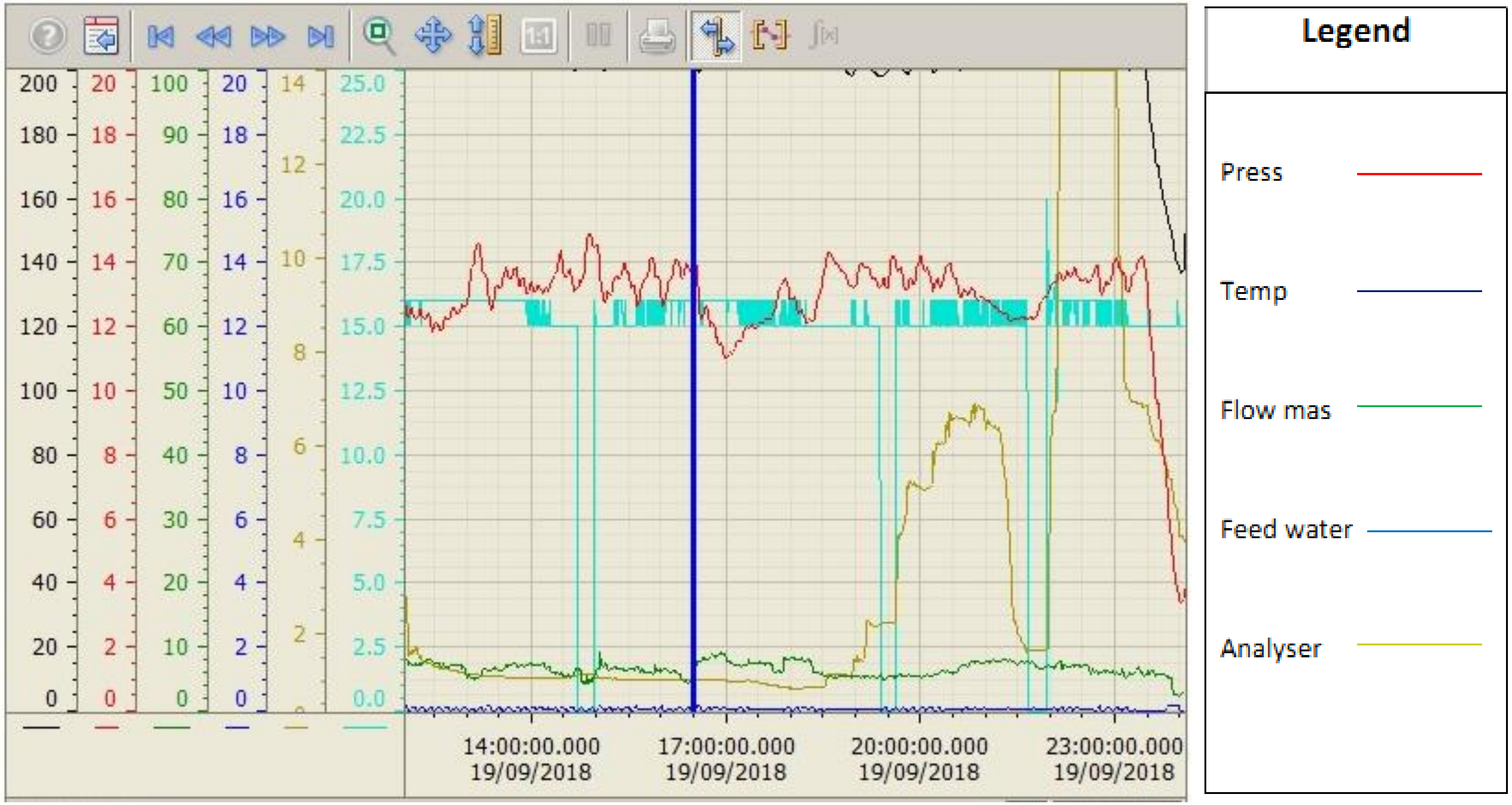
Appendix G: The boiler trend overview of the flow pressure and operating temperature; steam flow On the 18/09/2018 between 14:00 to 23:00, (Siemens Win CC of Isegen SA Pty Ltd).



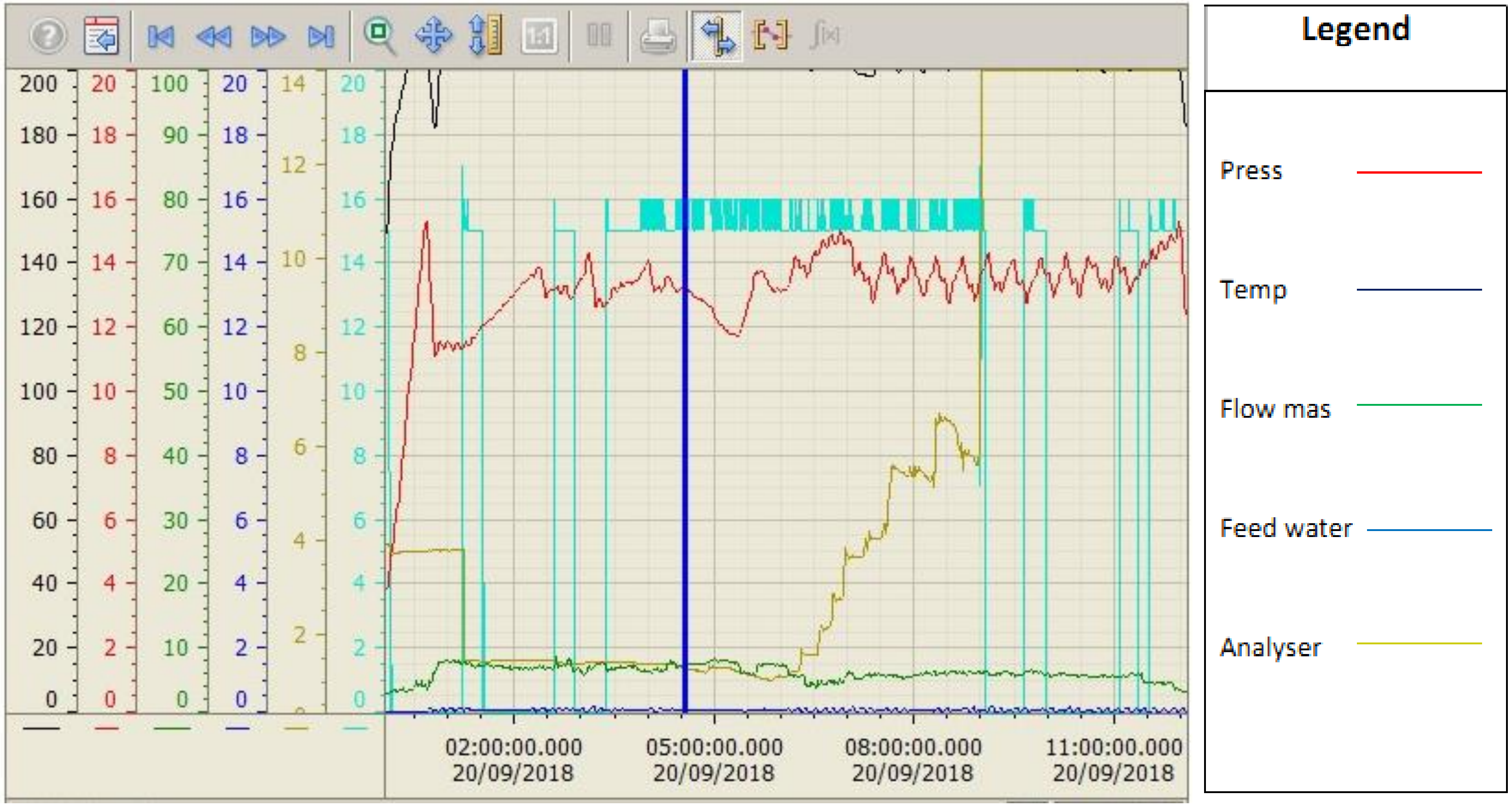
Appendix G: The boiler trend overview of the flow pressure and operating temperature; steam flow On the 19/09/2018 between 02:00 to 11:00, (Siemens Win CC of Isegen SA Pty Ltd).



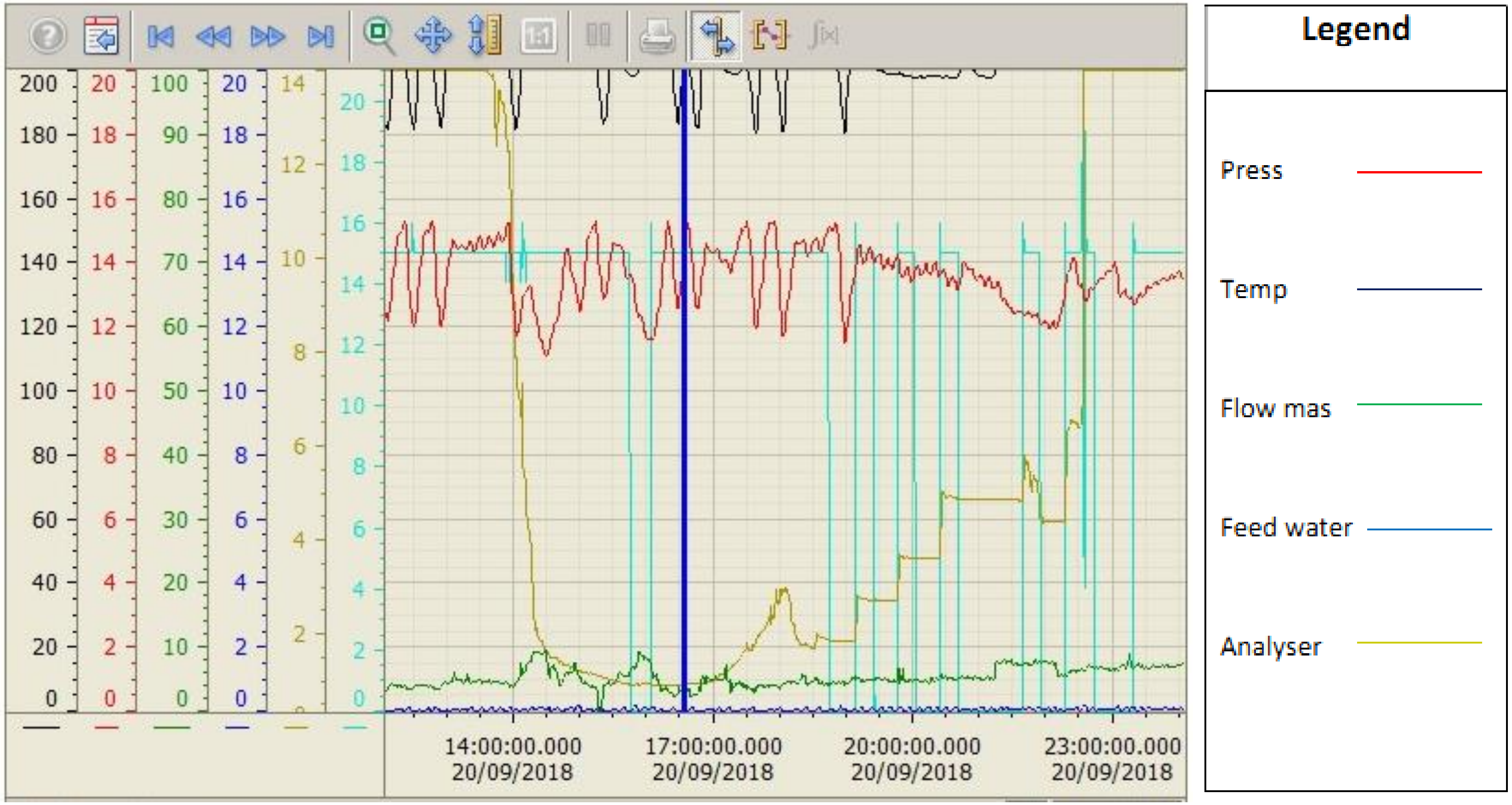
Appendix G: The boiler trend overview of the flow pressure and operating temperature; steam flow On the 19/09/2018 between 14:00 to 23:00, (Siemens Win CC of Isegen SA Pty Ltd).



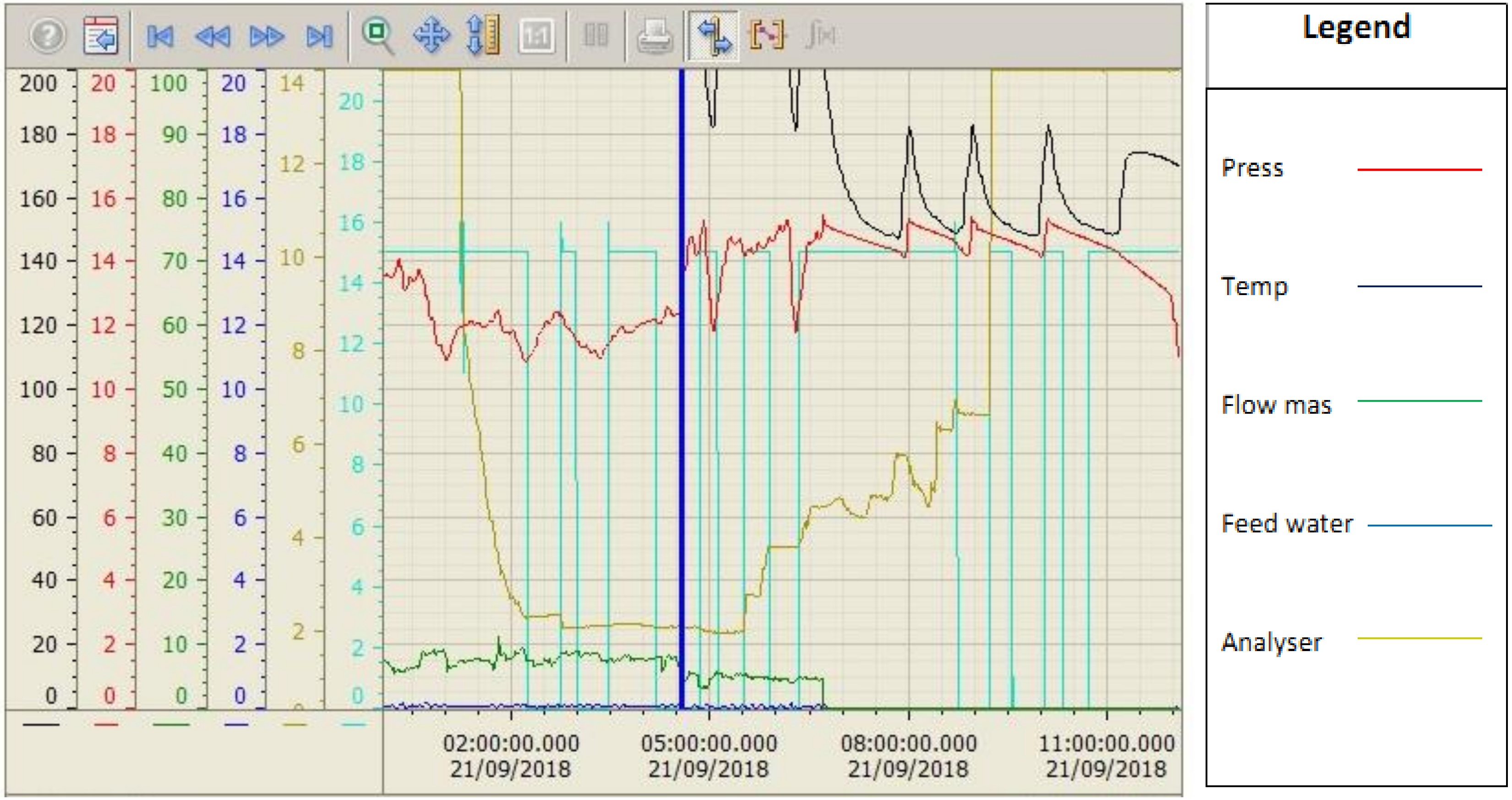
Appendix G: The boiler trend overview of the flow pressure and operating temperature; steam flow On the 20/09/2018 between 02:00 to 11:00, (Siemens Win CC of Isegen SA Pty Ltd).



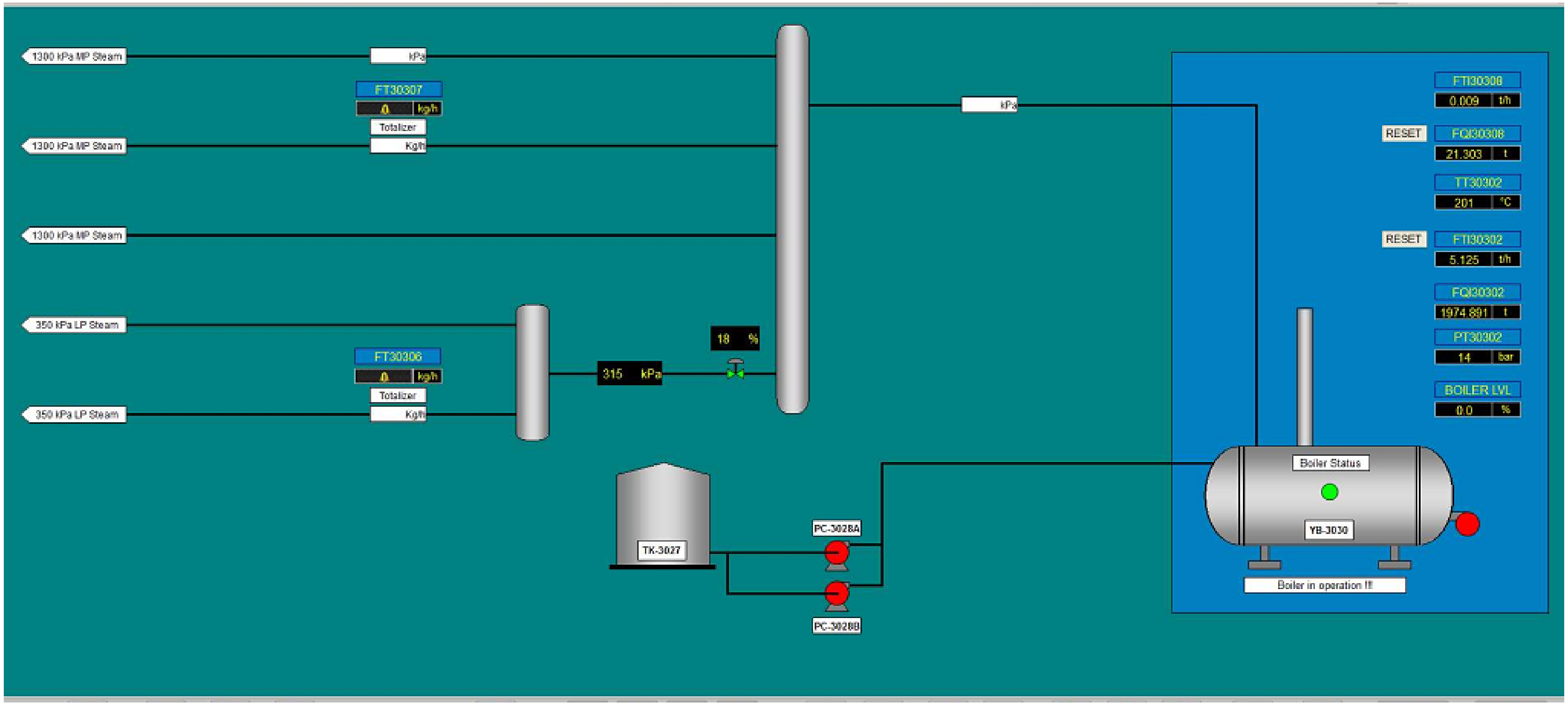
Appendix G: The boiler trend overview of the flow pressure and operating temperature; steam flow On the 20/09/2018 between 14:00 to 23:00, (Siemens Win CC of Isegen SA Pty Ltd).



Appendix G: The boiler trend overview of the flow pressure and operating temperature; steam flow On the 21/09/2018 between 02:00 to 11:00, (Siemens Win CC of Isegen SA Pty Ltd).



Appendix H: The Boiler Steam Distribution System.



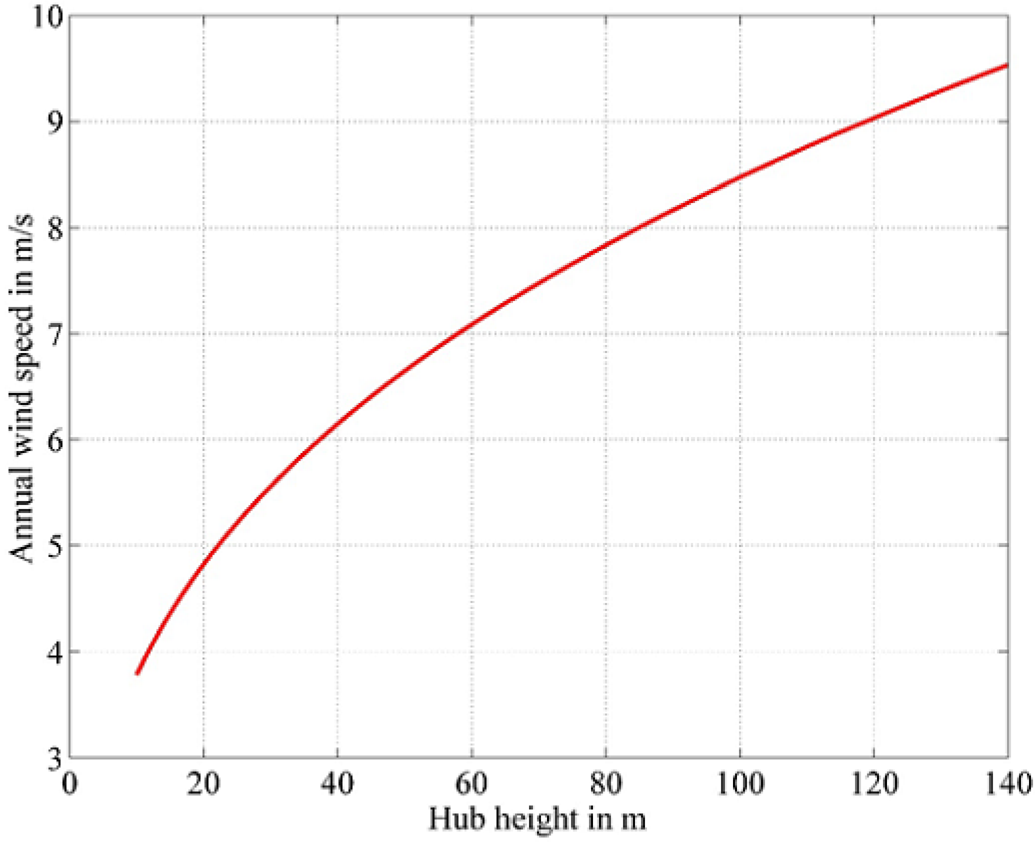
Appendix I: The Steam Trap Range.

Table 19I: The steam trap table range (http://www.spiraxsarco.com).

Steam trap operation	Thermodynamic	Mechanical		Thermostatic	
Steam trap types	**Thermodynamic** 	**Ball float** 	**Inverted bucket** 	**Balanced pressure** 	**Bimetallic** 
Main features	• Robust design giving excellent resistance to waterhammer and vibration • Inexpensive • Positive discharge with tight shut-off • Discharge condensate close to steam saturation temperature	• High capacity • Excellent air venting capabilities • Continuous discharge of condensate for maximum heat transfer • Will not back-up with condensate	• High capacity • Robust design • Near continuous discharge of condensate • Minimal back-up of condensate	• Utilises sensible heat in the condensate, reducing flash steam losses, which saves energy • Excellent air venting properties for quick start-up	
Typical applications	Mains drainage and all tracing applications. Some process applications with light loads such as small presses and cylinders	Temperature / pressure controlled applications with fluctuating loads	Temperature / pressure controlled applications with fluctuating loads	• Where condensate back-up can be tolerated or is required in order to remove excess enthalpy, e.g. non-critical tracing	
Size	DN8 – DN25 (¼"– 1")	DN15 – DN100 (½" – 4")	DN15 – DN50 (½" – 2")	DN8 – DN25 (¼"– 1")	DN8 – DN100 (¼"– 4")
Maximum body rating	PN250	PN100 and ASME Class 600	ASME 900	PN40 and ASME Class 300	PN420 and ASME Class 2500
Maximum operating pressure	250 bar g	80 bar g	110 bar g	32 bar g	150 bar g

Appendix J: Durban annual mean wind speeds at different hub heights

Characterisation of wind speed series and power in Durban - Scientific Figure on ResearchGate. Available from: https://www.researchgate.net/figure/Annual-mean-wind-speeds-at-different-hub-heights_fig4_319995530 [accessed 15 Sep, 2018]



Appendix K: Durban ambient temperatures by month (https://en.climate-data.org)

DURBAN WEATHER BY MONTH // WEATHER AVERAGES

	January	February	March	April	May	June	July	August	September	October	November	December
Avg. Temperature (°C)	24.1	24.5	23.7	21.8	19.4	17.2	16.8	17.9	19.2	20.5	21.8	23.4
Min. Temperature (°C)	20.6	20.9	19.9	17.5	14.4	11.6	11.3	12.9	15	16.8	18.2	19.8
Max. Temperature (°C)	27.7	28.1	27.6	26.1	24.4	22.9	22.4	22.9	23.5	24.2	25.4	27
Avg. Temperature (°F)	75.4	76.1	74.7	71.2	66.9	63.0	62.2	64.2	66.6	68.9	71.2	74.1
Min. Temperature (°F)	69.1	69.6	67.8	63.5	57.9	52.9	52.3	55.2	59.0	62.2	64.8	67.6
Max. Temperature (°F)	81.9	82.6	81.7	79.0	75.9	73.2	72.3	73.2	74.3	75.6	77.7	80.6
Precipitation / Rainfall (mm)	124	113	125	71	56	30	31	46	64	95	110	110

Appendix L: Typical heat losses from insulated pipes (W/m)

Table 20: Typical heat losses from insulated pipes (https://www.spiraxsarco.com/learn-about-steam/steam-distribution/air-venting-heat-losses-and-a-summary-of-various-pipe-related-standards)

Process diameter (mm)	Insulation thickness	Product/ambient temperature difference (°C)						
		25	75	100	125	150	175	200
DN100	50	14	43	58	71	86	100	115
	100	9	26	36	45	54	62	71
DN150	50	20	59	77	97	116	136	155
	100	12	35	46	58	69	81	92
DN200	50	24	72	97	120	144	168	192
	100	14	41	55	70	84	98	112
DN250	50	29	87	116	145	174	202	231
	100	16	49	66	82	99	115	131
DN300	50	33	101	135	168	201	235	268
	100	18	56	75	94	113	131	151
DN400	50	41	123	164	206	246	288	329
	100	23	68	91	113	136	158	181
DN500	50	51	151	201	252	301	352	403
	100	28	82	109	136	163	191	217

Appendix M: Numerical model data heat loss calculations at the ambient temperature of 20°C and wind speed of 4m/s.

Table 21: Numerical model data heat loss calculations at the average ambient temperature of 20°C and wind speed of 4m/s.

The total heat loss of the steam pipeline based on the current conditions is thus obtained as

Pressure loss = 0.1 b/100m ref : https://www.spiraxsarco.com/learn-about-steam/steam-distribution/pipes-and-pipe-sizing at figure 10.2.7

$$\dot{q} = \frac{(t_1 - t_6)}{\frac{1}{2\pi r_{1L\,h_1}} + ln\left(\frac{r_2}{r_1}\right)/2\pi\lambda_1 L + ln\left(\frac{r_3}{r_2}\right)/2\pi\lambda_2 L + ln\left(\frac{r_4}{r_3}\right)/2\pi\lambda_3 L + \frac{1}{2\pi r_{4L\,h_o}}}$$

Input	NB150 Insulated Steam Pipeline Section									Condensate drain point at 100m	NB150 Insulated Steam Pipeline Section		Equivalent length of pipe fittings	NB150 Insulated Steam Pipeline Section							Condensate drain point at 200m	NB150 Insulated Steam Pipeline Section						Manmanifold uninsulated pipeline systems	Units
$P_{sat\,(abs)}$ =	15.450	15.440	15.430	15.420	15.410	15.400	15.390	15.380	15.370	15.360	15.350	15.340	15.331	15.331	15.321	15.311	15.301	15.291	15.281	15.271	15.261	15.251	15.241	15.241	15.231	15.221	15.211	15.201	bar
t_s=	200.1	200.1	200.0	200.0	200.0	199.9	199.9	199.8	199.8	199.7	199.7	199.6	199.6	199.6	199.5	199.5	199.4	199.4	199.3	199.3	199.2	199.2	199.1	199.1	199.1	199.0	199.0	198.9	°C
t_a=	20																												°C
h_i=	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	W/m^2K
h_o=	18.0378																												W/m^2K
k_c=	43																												W/mK
k_{ins}=	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06		0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06		0.06	0.06	0.06	0.06		W/mK
$k_{clad(Alum)}$=	237	237	237	237	237	237	237	237	237	237	237	237		237	237	237	237	237	237	237	237	237		237	237	237	237		W/mK
r_1=	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.10137	m
r_2=	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.10955	m
r_3=	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415		0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415		0.13415	0.13415	0.13415	0.13415		m
r_4=	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615		0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615		0.13615	0.13615	0.13615	0.13615		m
$\dot{Q}$ =	**132.56**	**132.53**	**132.49**	**132.46**	**132.43**	**132.39**	**132.36**	**132.32**	**132.28**	**132.25**	**132.21**	**132.18**	**1109.14**	**132.14**	**132.10**	**132.07**	**132.03**	**131.99**	**131.96**	**131.92**	**131.89**	**131.85**	**1106.41**	**131.81**	**131.78**	**131.74**	**131.71**	**1443.72**	**W/m**
L=	10	10	10	10	10	10	10	10	10	10	10	9.354	0.646	10	10	10	10	10	10	10	10	9.354	0.646	10	10	10	10	1.5	m
$\dot{Q}$ =	**1.33**	**1.33**	**1.32**	**1.32**	**1.32**	**1.32**	**1.32**	**1.32**	**1.32**	**1.32**	**1.32**	**1.24**	**0.72**	**1.32**	**1.32**	**1.32**	**1.32**	**1.32**	**1.32**	**1.32**	**1.32**	**1.23**	**0.71**	**1.32**	**1.32**	**1.32**	**1.32**	**2.17**	**kW**
t_{surf}=	28.6	28.6	28.6	28.6	28.6	28.6	28.6	28.6	28.6	28.6	28.6	28.6	91.9	28.6	28.6	28.6	28.6	28.6	28.6	28.5	28.5	28.5	91.7	28.5	28.5	28.5	28.5	113.6	°C
																				Theoritical Condensate running load	0.0071	kg cond/s	25.61	kg/h					
																				Measured Condensate running load	0.0203	kg cond/s	73.20	kg/h	2.86	x as much	**Total Heat loss**	**36.46**	**kW**

Appendix M: Numerical model data heat loss calculations at the ambient temperature of 20°C and wind speed of 5m/s.

Table 22: Numerical model data heat loss calculations at the average ambient temperature of 20°C and wind speed of 5m/s.

The total heat loss of the steam pipeline based on the current conditions is thus obtained as

Pressure loss = 0.1 b/100m ref : https://www.spiraxsarco.com/learn-about-steam/steam-distribution/pipes-and-pipe-sizing,figure 10.2.7

$$\dot{q} = \frac{(t_1 - t_6)}{\frac{1}{2\pi r_{1L\,h_1}} + \ln\left(\frac{r_2}{r_1}\right)/2\pi\lambda_1 L + \ln\left(\frac{r_3}{r_2}\right)/2\pi\lambda_2 L + \ln\left(\frac{r_4}{r_3}\right)/2\pi\lambda_3 L + \frac{1}{2\pi r_{4L\,h_o}}}$$

Input	NB150 Insulated Steam Pipeline Section									Condensate drain point at 100m	NB150 Insulated Steam Pipeline Section		Equivalent length of pipe fittings	NB150 Insulated Steam Pipeline Section							Condensate drain point at 200m	NB150 Insulated Steam Pipeline Section						Manmanifold uninsulated pipeline systems	Units
$P_{sat\,(abs)}$ =	15.450	15.440	15.430	15.420	15.410	15.400	15.390	15.380	15.370	15.360	15.350	15.340	15.331	15.331	15.321	15.311	15.301	15.291	15.281	15.271	15.261	15.251	15.241	15.241	15.231	15.221	15.211	15.201	bar
t_s=	200.1	200.1	200.0	200.0	200.0	199.9	199.9	199.8	199.8	199.7	199.7	199.6	199.6	199.6	199.5	199.5	199.4	199.4	199.3	199.3	199.2	199.2	199.1	199.1	199.1	199.0	199.0	198.9	°C
t_a=	20																												°C
h_i=	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	W/m^2K
h_o=	21.5872																												W/m^2K
k_c=	43																												W/mK
k_{ins}=	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06		0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06		0.06	0.06	0.06	0.06		W/mK
$k_{clad(Alum)}$=	237	237	237	237	237	237	237	237	237	237	237	237		237	237	237	237	237	237	237	237	237		237	237	237	237		W/mK
r_1=	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.10137	m
r_2=	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.10955	m
r_3=	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415		0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415		0.13415	0.13415	0.13415	0.13415		m
r_4=	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615		0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615		0.13615	0.13615	0.13615	0.13615		m
$\dot{Q}$ =	**133.61**	**133.57**	**133.54**	**133.51**	**133.47**	**133.44**	**133.40**	**133.37**	**133.33**	**133.29**	**133.26**	**133.22**	**1241.33**	**133.19**	**133.15**	**133.11**	**133.07**	**133.04**	**133.00**	**132.96**	**132.93**	**132.89**	**1238.26**	**132.86**	**132.82**	**132.78**	**132.75**	**1616.44**	**W/m**
L=	10	10	10	10	10	10	10	10	10	10	10	9.354	0.646	10	10	10	10	10	10	10	10	9.354	0.646	10	10	10	10	1.5	m
$\dot{Q}$ =	**1.34**	**1.34**	**1.34**	**1.34**	**1.33**	**1.33**	**1.33**	**1.33**	**1.33**	**1.33**	**1.33**	**1.25**	**0.80**	**1.33**	**1.33**	**1.33**	**1.33**	**1.33**	**1.33**	**1.33**	**1.33**	**1.24**	**0.80**	**1.33**	**1.33**	**1.33**	**1.33**	**2.42**	**kW**
t_{surf}=	**27.2**	**27.2**	**27.2**	**27.2**	**27.2**	**27.2**	**27.2**	**27.2**	**27.2**	**27.2**	**27.2**	**27.2**	**87.2**	**27.2**	**27.2**	**27.2**	**27.2**	**27.2**	**27.2**	**27.2**	**27.2**	**27.2**	**87.1**	**27.2**	**27.2**	**27.2**	**27.2**	**107.5**	**°C**
																				Theoritical Condensate running load	0.0072	kg cond/s	25.96	kg/h					
																				Measured Condensate running load	0.0203	kg cond/s	73.20	kg/h	2.82	x as much	**Total Heat loss**	**37.15**	**kW**

Appendix M: Numerical model data heat loss calculations at the ambient temperature of 20°C and wind speed of 6m/s.

Table 23: Numerical model data heat loss calculations at the average ambient temperature of 20°C and wind speed of 6m/s.

The total heat loss of the steam pipeline based on the current conditions is thus obtained as

Pressure loss = 0.1 b/100m ref : https://www.spiraxsarco.com/learn-about-steam/steam-distribution/pipes-and-pipe-sizing,figure 10.2.7

$$\dot{q} = \frac{(t_1 - t_6)}{\frac{1}{2\pi r_{1L\,h_1}} + ln\left(\frac{r_2}{r_1}\right)/2\pi\lambda_1 L + ln\left(\frac{r_3}{r_2}\right)/2\pi\lambda_2 L + ln\left(\frac{r_4}{r_3}\right)/2\pi\lambda_3 L + \frac{1}{2\pi r_{4L\,h_o}}}$$

Input	NB150 Insulated Steam Pipeline Section									Condensate drain point at 100m	NB150 Insulated Steam Pipeline Section		Equivalent length of pipe fittings	NB150 Insulated Steam Pipeline Section							Condensate drain point at 200m	NB150 Insulated Steam Pipeline Section						Manmanifold uninsulated pipeline systems	Units
$P_{sat\,(abs)}$ =	15.450	15.440	15.430	15.420	15.410	15.400	15.390	15.380	15.370	15.360	15.350	15.340	15.331	15.331	15.321	15.311	15.301	15.291	15.281	15.271	15.261	15.251	15.241	15.241	15.231	15.221	15.211	15.201	bar
t_s=	200.1	200.1	200.0	200.0	200.0	199.9	199.9	199.8	199.8	199.7	199.7	199.6	199.6	199.6	199.5	199.5	199.4	199.4	199.3	199.3	199.2	199.2	199.1	199.1	199.1	199.0	199.0	198.9	°C
t_a=	20																												°C
h_i=	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	W/m^2K
h_o=	24.9999																												W/m^2K
k_c=	43																												W/mK
k_{ins}=	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06		0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06		0.06	0.06	0.06	0.06		W/mK
$k_{clad(Alum)}$=	237	237	237	237	237	237	237	237	237	237	237	237		237	237	237	237	237	237	237	237	237		237	237	237	237		W/mK
r_1=	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.10137	m
r_2=	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.10955	m
r_3=	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415		0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415		0.13415	0.13415	0.13415	0.13415		m
r_4=	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615		0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615		0.13615	0.13615	0.13615	0.13615		m
$\dot{Q}$ =	**134.34**	**134.31**	**134.28**	**134.24**	**134.21**	**134.18**	**134.14**	**134.10**	**134.07**	**134.03**	**133.99**	**133.95**	**1353.21**	**133.92**	**133.88**	**133.85**	**133.81**	**133.77**	**133.73**	**133.70**	**133.66**	**133.62**	**1349.87**	**133.59**	**133.55**	**133.51**	**133.48**	**1762.74**	**W/m**
L=	10	10	10	10	10	10	10	10	10	10	10	9.354	0.646	10	10	10	10	10	10	10	10	9.354	0.646	10	10	10	10	1.5	m
$\dot{Q}$ =	**1.34**	**1.34**	**1.34**	**1.34**	**1.34**	**1.34**	**1.34**	**1.34**	**1.34**	**1.34**	**1.34**	**1.25**	**0.87**	**1.34**	**1.34**	**1.34**	**1.34**	**1.34**	**1.34**	**1.34**	**1.34**	**1.25**	**0.87**	**1.34**	**1.34**	**1.34**	**1.33**	**2.64**	**kW**
t_{surf}=	26.3	26.3	26.3	26.3	26.3	26.3	26.3	26.3	26.3	26.3	26.3	26.3	83.3	26.3	26.3	26.3	26.3	26.3	26.3	26.3	26.2	26.2	83.1	26.2	26.2	26.2	26.2	102.4	°C
																				Theoritical Condensate running load	0.0073	kg cond/s	26.23	kg/h					
																				Measured Condensate running load	0.0203	kg cond/s	73.20	kg/h	2.79	x as much	**Total Heat loss**	**37.70**	**kW**

Appendix M: Numerical model data heat loss calculations at the ambient temperature of 20°C and wind speed of 7m/s.

Table 24: Numerical model data heat loss calculations at the average ambient temperature of 20°C and wind speed of 7m/s.

The total heat loss of the steam pipeline based on the current conditions is thus obtained as

Pressure loss = 0.1 b/100m ref : https://www.spiraxsarco.com/learn-about-steam/steam-distribution/pipes-and-pipe-sizing,figure 10.2.7

$$\dot{q} = \frac{(t_1 - t_6)}{\frac{1}{2\pi r_{1L} h_1} + ln\left(\frac{r_2}{r_1}\right)/2\pi\lambda_1 L + ln\left(\frac{r_3}{r_2}\right)/2\pi\lambda_2 L + ln\left(\frac{r_4}{r_3}\right)/2\pi\lambda_3 L + \frac{1}{2\pi r_{4L} h_o}}$$

Input	NB150 Insulated Steam Pipeline Section									Condensate drain point at 100m	NB150 Insulated Steam Pipeline Section		Equivalent length of pipe fittings	NB150 Insulated Steam Pipeline Section							Condensate drain point at 200m	NB150 Insulated Steam Pipeline Section						Manmanifold uninsulated pipeline systems	Units
$P_{sat\,(abs)}$ =	15.450	15.440	15.430	15.420	15.410	15.400	15.390	15.380	15.370	15.360	15.350	15.340	15.331	15.331	15.321	15.311	15.301	15.291	15.281	15.271	15.261	15.251	15.241	15.241	15.231	15.221	15.211	15.201	bar
t_s=	200.1	200.1	200.0	200.0	200.0	199.9	199.9	199.8	199.8	199.7	199.7	199.6	199.6	199.6	199.5	199.5	199.4	199.4	199.3	199.3	199.2	199.2	199.1	199.1	199.1	199.0	199.0	198.9	°C
t_a=	20																												°C
h_i=	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	W/m^2K
h_o=	28.3029																												W/m^2K
k_c=	43																												W/mK
k_{ins}=	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06		0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06		0.06	0.06	0.06	0.06		W/mK
$k_{clad(Alum)}$=	237	237	237	237	237	237	237	237	237	237	237	237		237	237	237	237	237	237	237	237	237		237	237	237	237		W/mK
r_1=	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.10137	m
r_2=	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.10955	m
r_3=	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415		0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415		0.13415	0.13415	0.13415	0.13415		m
r_4=	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615		0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615		0.13615	0.13615	0.13615	0.13615		m
$\dot{Q}$ =	**134.89**	**134.86**	**134.82**	**134.79**	**134.76**	**134.72**	**134.69**	**134.65**	**134.61**	**134.58**	**134.54**	**134.50**	**1449.66**	**134.47**	**134.43**	**134.39**	**134.36**	**134.32**	**134.28**	**134.24**	**134.21**	**134.17**	**1446.08**	**134.13**	**134.10**	**134.06**	**134.02**	**1888.94**	**W/m**
L=	10	10	10	10	10	10	10	10	10	10	10	9.354	0.646	10	10	10	10	10	10	10	10	9.354	0.646	10	10	10	10	1.5	m
$\dot{Q}$ =	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.26**	**0.94**	**1.34**	**1.34**	**1.34**	**1.34**	**1.34**	**1.34**	**1.34**	**1.34**	**1.26**	**0.93**	**1.34**	**1.34**	**1.34**	**1.34**	**2.83**	**kW**
t_{surf}=	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	25.6	79.9	25.6	25.6	25.6	25.5	25.5	25.5	25.5	25.5	25.5	79.7	25.5	25.5	25.5	25.5	98.0	°C
																				Theoritical Condensate running load	0.0073	kg cond/s	26.44	kg/h					
																				Measured Condensate running load	0.0203	kg cond/s	73.20	kg/h	2.77	x as much	**Total Heat loss**	**38.15**	**kW**

Appendix M: Numerical model data heat loss calculations at the ambient temperature of 20°C and wind speed of 8m/s.

Table 25: Numerical model data heat loss calculations at the average ambient temperature of 20°C and wind speed of 8m/s.

The total heat loss of the steam pipeline based on the current conditions is thus obtained as

Pressure loss = 0.1 b/100m ref : https://www.spiraxsarco.com/learn-about-steam/steam-distribution/pipes-and-pipe-sizing,figure 10.2.7

$$\dot{q} = \frac{(t_1 - t_6)}{\frac{1}{2\pi r_{1L\,h_1}} + \ln\left(\frac{r_2}{r_1}\right)/2\pi\lambda_1 L + \ln\left(\frac{r_3}{r_2}\right)/2\pi\lambda_2 L + \ln\left(\frac{r_4}{r_3}\right)/2\pi\lambda_3 L + \frac{1}{2\pi r_{4L\,h_o}}}$$

Input	NB150 Insulated Steam Pipeline Section									Condensate drain point at 100m	NB150 Insulated Steam Pipeline Section		Equivalent length of pipe fittings	NB150 Insulated Steam Pipeline Section							Condensate drain point at 200m	NB150 Insulated Steam Pipeline Section						Manmanifold uninsulated pipeline systems	Units
$P_{sat\,(abs)}$ =	15.450	15.440	15.430	15.420	15.410	15.400	15.390	15.380	15.370	15.360	15.350	15.340	15.331	15.331	15.321	15.311	15.301	15.291	15.281	15.271	15.261	15.251	15.241	15.241	15.231	15.221	15.211	15.201	bar
t_s=	200.1	200.1	200.0	200.0	200.0	199.9	199.9	199.8	199.8	199.7	199.7	199.6	199.6	199.6	199.5	199.5	199.4	199.4	199.3	199.3	199.2	199.2	199.1	199.1	199.1	199.0	199.0	198.9	°C
t_a=	20																												°C
h_i=	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	W/m²K
h_o=	31.5147																												W/m²K
k_c=	43																												W/mK
k_{ins}=	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06		0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06		0.06	0.06	0.06	0.06		W/mK
$k_{clad(Alum)}$=	237	237	237	237	237	237	237	237	237	237	237	237		237	237	237	237	237	237	237	237	237		237	237	237	237		W/mK
r_1=	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.10137	m
r_2=	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.10955	m
r_3=	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415		0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415		0.13415	0.13415	0.13415	0.13415		m
r_4=	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615		0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615		0.13615	0.13615	0.13615	0.13615		m
$\dot{Q}$ =	**135.32**	**135.28**	**135.25**	**135.22**	**135.18**	**135.15**	**135.11**	**135.08**	**135.04**	**135.00**	**134.96**	**134.93**	**1534.00**	**134.89**	**134.85**	**134.82**	**134.78**	**134.74**	**134.71**	**134.67**	**134.63**	**134.59**	**1530.21**	**134.56**	**134.52**	**134.48**	**134.45**	**1999.36**	**W/m**
L=	10	10	10	10	10	10	10	10	10	10	10	9.354	0.646	10	10	10	10	10	10	10	10	9.354	0.646	10	10	10	10	1.5	m
$\dot{Q}$ =	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.26**	**0.99**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.26**	**0.99**	**1.35**	**1.35**	**1.34**	**1.34**	**3.00**	**kW**
t_{surf}=	25.0	25.0	25.0	25.0	25.0	25.0	25.0	25.0	25.0	25.0	25.0	25.0	76.9	25.0	25.0	25.0	25.0	25.0	25.0	25.0	25.0	25.0	76.8	25.0	25.0	25.0	25.0	94.2	°C
																				Theoritical Condensate running load	0.0074	kg cond/s	26.62	kg/h					
																				Measured Condensate running load	0.0203	kg cond/s	73.20	kg/h	2.75	x as much	**Total Heat loss**	**38.53**	**kW**

Appendix M: Numerical model data heat loss calculations at the ambient temperature of 20°C and wind speed of 9m/s.

Table 26: Numerical model data heat loss calculations at the average ambient temperature of 20°C and wind speed of 9m/s.

The total heat loss of the steam pipeline based on the current conditions is thus obtained as

Pressure loss = 0.1 b/100m ref : https://www.spiraxsarco.com/learn-about-steam/steam-distribution/pipes-and-pipe-sizing,figure 10.2.7

$$\dot{q} = \frac{(t_1 - t_6)}{\frac{1}{2\pi r_{1L} h_1} + \ln\left(\frac{r_2}{r_1}\right)/2\pi\lambda_1 L + \ln\left(\frac{r_3}{r_2}\right)/2\pi\lambda_2 L + \ln\left(\frac{r_4}{r_3}\right)/2\pi\lambda_3 L + \frac{1}{2\pi r_{4L} h_o}}$$

Input	NB150 Insulated Steam Pipeline Section									Condensate drain point at 100m	NB150 Insulated Steam Pipeline Section		Equivalent length of pipe fittings	NB150 Insulated Steam Pipeline Section							Condensate drain point at 200m	NB150 Insulated Steam Pipeline Section						Manmanifold uninsulated pipeline systems	Units
$P_{sat\,(abs)}$ =	15.450	15.440	15.430	15.420	15.410	15.400	15.390	15.380	15.370	15.360	15.350	15.340	15.331	15.331	15.321	15.311	15.301	15.291	15.281	15.271	15.261	15.251	15.241	15.241	15.231	15.221	15.211	15.201	bar
t_s=	200.1	200.1	200.0	200.0	200.0	199.9	199.9	199.8	199.8	199.7	199.7	199.6	199.6	199.6	199.5	199.5	199.4	199.4	199.3	199.3	199.2	199.2	199.1	199.1	199.1	199.0	199.0	198.9	°C
t_a=	20																												°C
h_i=	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	W/m^2K
h_o=	34.6491																												W/m^2K
k_c=	43																												W/mK
k_{ins}=	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06		0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06		0.06	0.06	0.06	0.06		W/mK
$k_{clad(Alum)}$=	237	237	237	237	237	237	237	237	237	237	237	237		237	237	237	237	237	237	237	237	237		237	237	237	237		W/mK
r_1=	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.10137	m
r_2=	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.10955	m
r_3=	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415		0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415		0.13415	0.13415	0.13415	0.13415		m
r_4=	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615		0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615		0.13615	0.13615	0.13615	0.13615		m
$\dot{Q}$ =	**135.66**	**135.63**	**135.59**	**135.56**	**135.53**	**135.49**	**135.45**	**135.42**	**135.38**	**135.34**	**135.31**	**135.27**	**1608.60**	**135.23**	**135.20**	**135.16**	**135.12**	**135.08**	**135.05**	**135.01**	**134.97**	**134.93**	**1604.63**	**134.90**	**134.86**	**134.82**	**134.79**	**2097.08**	**W/m**
L=	10	10	10	10	10	10	10	10	10	10	10	9.354	0.646	10	10	10	10	10	10	10	10	9.354	0.646	10	10	10	10	1.5	m
$\dot{Q}$ =	**1.36**	**1.36**	**1.36**	**1.36**	**1.36**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.27**	**1.04**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.26**	**1.04**	**1.35**	**1.35**	**1.35**	**1.35**	**3.15**	**kW**
t_{surf}=	24.6	24.6	24.6	24.6	24.6	24.6	24.6	24.6	24.6	24.6	24.6	24.6	74.3	24.6	24.6	24.6	24.6	24.6	24.6	24.6	24.6	24.6	74.1	24.6	24.5	24.5	24.5	90.8	°C
																				Theoritical Condensate running load	0.0074	kg cond/s	26.77	kg/h					
																				Measured Condensate running load	0.0203	kg cond/s	73.20	kg/h	2.73	x as much	**Total Heat loss**	**38.85**	**kW**

Appendix M: Numerical model data heat loss calculations at the ambient temperature of 20°C and wind speed of 10m/s.

Table 27: Numerical model data heat loss calculations at the average ambient temperature of 20°C and wind speed of 10m/s.

The total heat loss of the steam pipeline based on the current conditions is thus obtained as

Pressure loss = 0.1 b/100m ref : https://www.spiraxsarco.com/learn-about-steam/steam-distribution/pipes-and-pipe-sizing,figure 10.2.7

$$\dot{q} = \frac{(t_1 - t_6)}{\frac{1}{2\pi r_{1L\,h_1}} + ln\left(\frac{r_2}{r_1}\right)/2\pi\lambda_1 L + ln\left(\frac{r_3}{r_2}\right)/2\pi\lambda_2 L + ln\left(\frac{r_4}{r_3}\right)/2\pi\lambda_3 L + \frac{1}{2\pi r_{4L\,h_o}}}$$

Input	NB150 Insulated Steam Pipeline Section								Condensate drain point at 100m	NB150 Insulated Steam Pipeline Section		Equivalent length of pipe fittings	NB150 Insulated Steam Pipeline Section							Condensate drain point at 200m	NB150 Insulated Steam Pipeline Section						Manmanifold uninsulated pipeline systems	Units
$P_{sat\,(abs)}$ =	15.450	15.440	15.430	15.420	15.410	15.400	15.390	15.380	15.370	15.360	15.350	15.340	15.331	15.331	15.321	15.311	15.301	15.291	15.281	15.271	15.261	15.251	15.241	15.241	15.231	15.221	15.211	15.201	bar
t_s=	200.1	200.1	200.0	200.0	200.0	199.9	199.9	199.8	199.8	199.7	199.7	199.6	199.6	199.6	199.5	199.5	199.4	199.4	199.3	199.3	199.2	199.2	199.1	199.1	199.1	199.0	199.0	198.9	°C
t_a=	20																												°C
h_i=	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	36.4235	W/m^2K
h_o=	37.7161																												W/m^2K
k_c=	43																												W/mK
k_{ins}=	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06		0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06	0.06		0.06	0.06	0.06	0.06		W/mK
$k_{clad(Alum)}$=	237	237	237	237	237	237	237	237	237	237	237	237		237	237	237	237	237	237	237	237	237		237	237	237	237		W/mK
r_1=	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.07704	0.10137	m
r_2=	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.08415	0.10955	m
r_3=	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415		0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415	0.13415		0.13415	0.13415	0.13415	0.13415		m
r_4=	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615		0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615	0.13615		0.13615	0.13615	0.13615	0.13615		m
$\dot{Q}$ =	**135.94**	**135.91**	**135.87**	**135.84**	**135.81**	**135.77**	**135.73**	**135.70**	**135.66**	**135.62**	**135.59**	**135.55**	**1675.21**	**135.51**	**135.48**	**135.44**	**135.40**	**135.36**	**135.33**	**135.29**	**135.25**	**135.21**	**1671.08**	**135.18**	**135.14**	**135.10**	**135.07**	**2184.38**	**W/m**
L=	10	10	10	10	10	10	10	10	10	10	10	9.354	0.646	10	10	10	10	10	10	10	10	9.354	0.646	10	10	10	10	1.5	m
$\dot{Q}$ =	**1.36**	**1.36**	**1.36**	**1.36**	**1.36**	**1.36**	**1.36**	**1.36**	**1.36**	**1.36**	**1.36**	**1.27**	**1.08**	**1.36**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.35**	**1.26**	**1.08**	**1.35**	**1.35**	**1.35**	**1.35**	**3.28**	**kW**
t_{surf}=	24.2	24.2	24.2	24.2	24.2	24.2	24.2	24.2	24.2	24.2	24.2	24.2	71.9	24.2	24.2	24.2	24.2	24.2	24.2	24.2	24.2	24.2	71.8	24.2	24.2	24.2	24.2	87.7	°C
																				Theoritical Condensate running load	0.0075	kg cond/s	26.91	kg/h					
																				Measured Condensate running load	0.0203	kg cond/s	73.20	kg/h	2.72	x as much	**Total Heat loss**	**39.14**	**kW**

Appendix N: Cost of steam pipeline inspection and design

GLOBAL INTEGRITY
SOLUTIONS CONSULTING (PTY) LTD

Physical Address
Unit 3 Ficus Building
Fairway Green
3 Aubrey Road
Kloof, 3610
Mobile: 071 1350 662

ATTENTION: Mr. Velaphi Phoswa 15 August 2019

REFERENCE Nr.: GS1266Q

COMPANY: Isegen SA Pty Ltd

QUOTATION: **PROFESSIONAL FEE FOR SERVICES RELATED TO CONDENSATE STEAM TRAP SYSTEM DESIGN AND INSPECTION**

1. INCLUSION IN THE SCOPE OF WORK

1.1 Kick-off meeting
1.2 Co-ordination of stakeholder meetings and project facilitation.
1.3 Assessment of the current existing 6" pipeline on the entire 250m length to ascertain the integrity of the pipeline and current thicknesses.
1.4 Conduct a visual inspection and thickness survey of the 6" pipeline on the 250m length. The entire length of the pipeline must be fully exposed to enable an inspection to be completed.
1.5 Detailed mechanical design calculations and drawings (2D and 3D) will be completed for the intended steam trap system.
1.6 Generating mechanical drawings indicating the general arrangement of all critical components and prominent parts of the steam trap system in ensuring that compliance to ASME B31.3 and or B31.3 as well as the relevant SANS 347 category is achieved.
1.7 Design Approval by the AIA (Approved Inspection Authority)
1.8 Conducting analysis to ascertain the thermal effects on the system pressure build-up due to piping hydraulic characteristics and applicable interconnecting piping configurations.
1.9 Performing design calculations to determine the recommended set Pressures for Safety Relief Valves and applicable venting capacities.
1.10 Prepare a detailed design report accompanied by mechanical calculations, mechanical drawings, and 3D drawings.
1.11 Submission of final design package for client acceptance and approval.

2. INFORMATION REQUIRED

2.1 Current 6" pipeline drawings, pressures, and transported medium.
2.2 Information on the pumping rates and location of pump sets including the discharge pipe routes and destinations.
2.3 Full access to the site where the intended system will be installed in order to extract all relevant dimensions and applicable tank geometric features.
2.4 Technical support from plant process personnel in relation to information that might be required in relation to plant operation.

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Unit 3 Ficus Building
Fairway Green
3 Aubrey Road
Kloof, 3610
Mobile: 071 1350 662

3. EXCLUSIONS
   - 3.1 Obtaining or any involvement in concessions.
   - 3.2 Metallurgical involvement (Inputs or recommendations).
   - 3.3 Weld map.
   - 3.4 Any meetings/site visits not listed above.
   - 3.5 Any QA/QC involvement.
   - 3.6 Rework due to any changes in scope.
   - 3.7 Any item, discipline, or responsibility not specifically in this Quotation.

4. COST AND INVOICES SCHEDULE
   - 4.1 Client to be invoiced 100% of the total Fee on submission of the full design package.
   - 4.2 All invoices are STRICTLY payable in 30days from the invoice date. 15% is charged on overdue payments.
   - 4.3 All pricing for the works to be as articulated in the pricing schedule in APPENDIX-A.
   - 4.4 All prices are VAT exclusive at 15%.

5. DELIVERY SCHEDULE

51. Design package ready as 'ISSUED FOR CONSTRUCTION' is envisaged to be completed and issued within fourteen days (14days) from the commencement of feasibility studies. The schedule is subject to negotiations with the Client.

Bheki Ngcobo
(Managing Director)
Cell:071 135 0662

GLOBAL INTEGRITY
SOLUTIONS CONSULTING (PTY) LTD

Physical Address
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Fairway Green
3 Aubrey Road
Kloof, 3610
Mobile: 071 1350 662

APPENDIX – A (Cost Schedule)

			DESIGN, DRAUGHTING, INSPECTION SERVICES FOR THE STEAM TRAP SYSTEM TO BE INSTALLED.		
	COST DESCRIPTION	TOTAL	RATE	Q TY	PRICE (VAT Excl.)
D. 1.1	Engineering Services on Tanks	Hours	Per Hour	Q TY	DESIGN COST
1.1	Site Investigations and Meetings	8	R 650.00	1	R 5,200.00
1.2	Generate 3D, Mechanical and General Arrangement Drawings	8	R 650.00	4	R 20,800.00
1.3	Compile Steam System Calculations and analysis	16	R 650.00	4	R 41,600.00
1.4	Compile Detailed Engineering Reports	16	R 650.00	4	R 41,600.00
1.5	Review and Approval by PrEng and AIA	16	R 1,050.00	4	R 67,200.00
					R 176,400.00
D. 1.2	INSPECTION AND NDT	Hours	Per Hour		INSPECTION COST
2.1	250m 6" Pipeline Visual Inspection	24	R 650.00	1	R 15,600.00
2.2	6" pipeline Ultrasonic Thickness Testing	40	R 650.00	1	R 26,000.00
					R 41,600.00
D. 1.3	Mobilization and Logistical Costs	Hours	Per Hour		MOBILIZATION COSTS
3.1	Team Travelling Time	14	R 950.00	1	R 13,300.00
3.2	Travelling and Milleage	1520	R 6.50	1	R 9,880.00
					R 23,180.00
			SUBTOTAL		R 241,180.00
			VAT @15%		R 36,177.00
			TOTAL FEE		R 277,357.00

Appendix O: Cost of steam pipeline upgrade

Vat No.: 4940255526 clint@kinlochseng.co.za admin@kinlochseng.co.za www.kinlochsengineering.co.za Kinloch's Engineering cc 117 Old Main Road Unit C ,New Germany Pinetown Tel No.: 031 701 2900 Fax No.: 086 616 9486	**Copy Quotation** Date 02/06/2020 Page 1 Document No QU101048
ISEGEN SOUTH AFRICAN (PTY)LTD 284 Refinery Dr, Isipingo Beach Isipingo 4115	Deliver to 284 Refinery Dr, Isipingo Beach Isipingo 4115

Account	Your Reference	Tax Exempt	Tax Reference	Sales Code	Expiry	
IS001	AWAITING ORDER NUMBER	N			02/06/2020	Exclusive

Code	Description	Quantity	Unit	Unit Price	Disc%	Tax	Nett Price
	SCOPE OF WORKS - Fabrication and Installation of 4x Steam Trap pockets set as per diagram						
LAB001	- Labour (Remove Old and Installation of New)	4.00		7,800.00		15.00%	R31,200.00
MAT	- Steam Equipment	4.00		18,790.00		15.00%	R75,160.00
MAT	- Pipe Fittings and Consumambles	4.00		1,850.00		15.00%	R7,400.00
MAT	- Guage	4.00		2,250.00		15.00%	R9,000.00
	(Subject to Remeasure)						
	EXCLUSIONS - Scaffolding - Pressure testing - X-Rays - Any 3rd Party Inspections						

PLEASE NOTE:
Quotation Valid for 30 Days

Received in good order

Signed____________________ Date____________________

Sub Total		R122,760.00
Discount @	0.00%	R0.00
Amount Excl Tax		R122,760.00
Tax		R18,414.00
Total		R141,174.00

Appendix P: Cost of new mineral wool and cladding and installation

Luleko
Contract Management Solutions (Pty) Ltd

75 Cosmos View
Trichard
3661
2729871047
phathisizwe@luleko.co.za

COPY QUOTATION

Date:	20/06/2020
Page:	1
Document No:	LLK2206

DELIVERY COMPANY DETAILS AND ADDRESS
ISEGEN SA (PTY)LTD
284 Refinery Road
Isiphingo Beach
4115

Delivery to:
ISEGEN SA (PTY)LTD
284 Refinery Road
Isiphingo Beach
4115

CODE	DESCRIPTION	QUANTITY	UNIT	DISC%	TAX	NET PRICE
1	Lagging material	1	38400		15%	38400
2	Cladding material	1	68730		15%	68730
3	Bends	6	1200		15%	7200
4	Tees	5	750		15%	3750
5	Flange boxes	6	3600		15%	21600
6	Labour	1	100000		15%	100000

Scope of work
Instalation of mineral wool insulation and aluminium lagging into a 6" steam pipeline at Isegen SA (Pty) Ltd.

Please Note:
Quotation Valid for 30 days

Received in good order

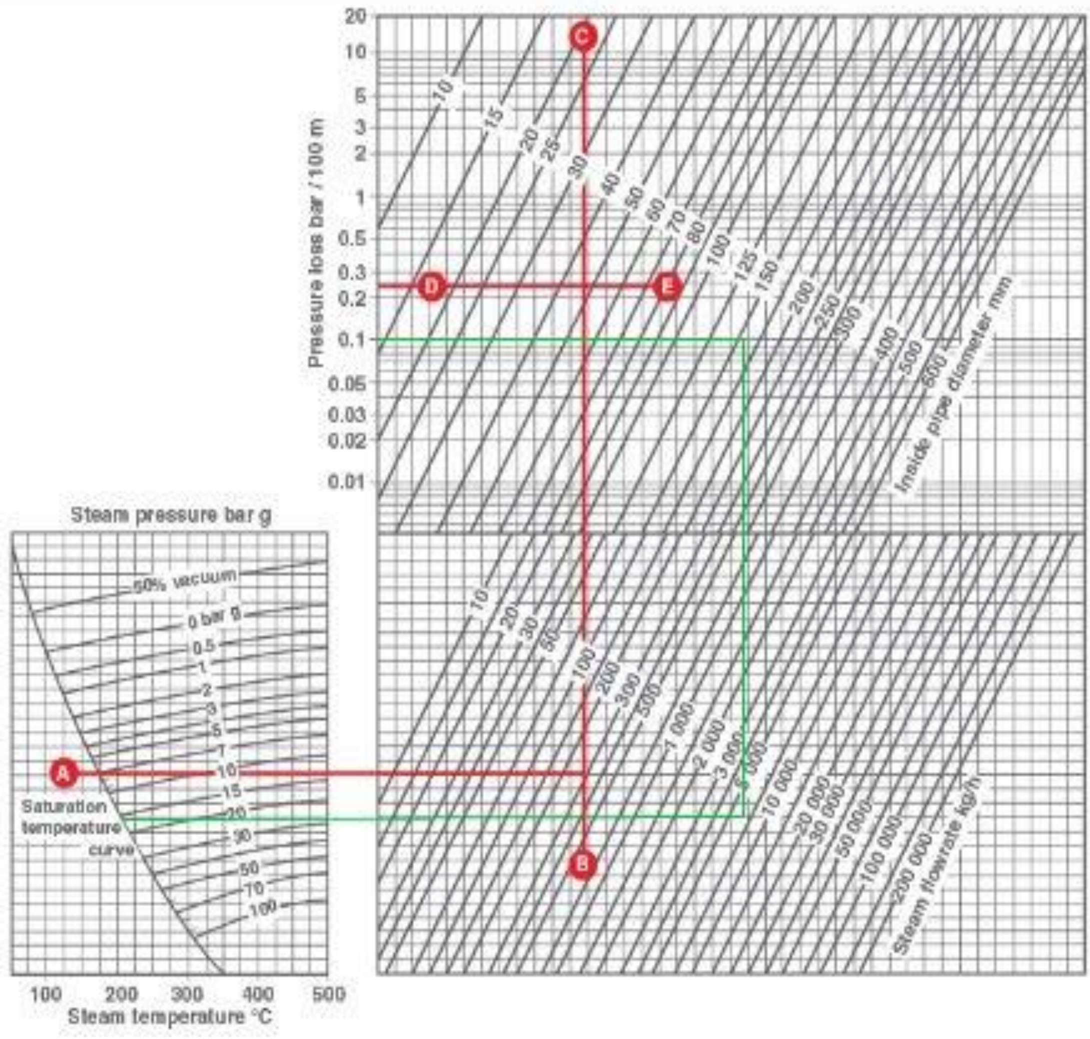
Signed...........................

Sub-Total	239680
Discount @	
Amount Excl VAT	
VAT	35952

TOTAL ZAR 275,632.00

Appendix Q: The steam pipeline sizing chart

Graph B: https://www.spiraxsarco.com/steam-distribution/pipes-and-pipe-sizing



Appendix R: The letter of Acknowledgement (Change of Management).

Letter of Acknowledgement: <u>The investigation of the effect of atmospheric conditions on the temperature drop across heat treatment systems.</u>

This letter confirms that Isegen SA (Pty) Ltd have been accepted the idea of the above-mentioned topic to be implemented in future on our site. The research study has been an eye-opener in terms of

- Energy Saving
- Fuel gas saving
- Steam consumption
- Cost reduction
- Process update

Change Management Plan Approval

The undersigned acknowledge of the (**Investigation of the effect of atmospheric conditions on the temperature drop across heat treatment systems**) Change Management Plan and agree with the approach it presents. Changes to this Change Management Plan will be coordinated with and approved by the undersigned of the following designated representatives.

Signature:		Date:	02/11/18.
Print Name:	AMITH SIEBALAK		
Title:	TECHNICAL MANAGER (MR)		
Role:	TECHNICAL MANAGER		
Signature:		Date:	2/11/18.
Print Name:	A. M. McKENZIE		
Title:	ENGINEERING LEADER.		
Role:	ENGINEER		
Signature:		Date:	2/11/2018
Print Name:	E. C. NOEL		
Title:	INSTRUMENTATION SUPERVISOR		
Role:	MAINTENANCE DEPARTMENT		

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