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Heat Recovery in a Milk Powder Spray-Drying Process

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by

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ABSTRACT

Energy used for processing in food and beverages factories is one of the most single expensive aspects of the entire process. Organisations depend on using recovered energy where possible to curb the cost of generating new energy. Heat recovery, therefore, becomes the area of focus to ensure that the total cost of energy is reduced. Furthermore, expanding organisations are most likely to depend on recovered energy sources for process optimisation and expansion. It has been hypothesised that there is a significant premium for recovering high-temperature heat energy and exhausting it to the atmosphere at the lowest temperature possible.

The aim of this study is to investigate the factors affecting the optimization of production of milk powder in a spray-drying process, with an argument that the heat energy in the exhaust air of the final stage of the drying process can be used to optimize the production capacity and energy demand of the process. The aim focuses specifically on ascertaining if the heat energy is recoverable, evaluating the amount of energy that is lost to the atmosphere, exploring the most economical way of using recovered energy, determining the expected optimization as a result of using recovered energy and evaluating the cost associated with recovering heat energy.

A qualitative research approach was used to understand the underlying problems and challenges in exhaust heat recovery. Based on the aforementioned challenges, a quantitative method was used to ascertain possible solutions to exhaust heat recovery. The findings from the primary research concluded that there is a significant amount of energy lost to the atmosphere which is recoverable. The results revealed that approximately 5 300 MJ of energy exit to the atmosphere on an hourly basis during production. Both the primary and secondary research revealed that the major challenge facing heat recovery is matching the recovered heat source to the heat sink in the process.

The study recommends that the most simple and economical way to use the recovered heat in spray-drying processes is to pre-heat the incoming fresh air. The study also recommends that the heat recovery systems should be well-designed and energy-saving-oriented to achieve the desired results. The capital cost of adopting an exhaust heat recovery system is high; however, the return on investment renders the investment lucrative.

DECLARATION

I, Asanda Ndimande, do hereby declare that this dissertation is a result of my own independent investigation and research and that this has not been submitted in part for any other degree or at any other University.

A.M. Ndimande

Date

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NOMENCLATURE

Q	$Heat\ Energy$	kJ/s
$\dot{m}$	$Mass\ flow\ rate$	kg/s
c_p	$Specific\ Heat\ Capacity$	kJ/kgK
ΔT	$Temperature\ difference$	K
η	$Efficiency$	$\%$
T, t	$Temperature$	℃
L, l	$Length$	m
u, h	$Heat\ Transfer\ Coefficient$	W/m^2 ℃
k	$Thermal\ Conductivity$	W/m ℃
r	$Radius$	m
A	$Cross - Sectional\ Area$	m^2
Δr	$Distance\ Difference$	m
q	$Heat\ Transfer\ Rate$	W
ε	$Material\ Emissivity$	
σ	$Stefan - Boltzmann\ Constant$	
C	$Carbon$	
O	$Oxygen$	
H	$Hydrogen$	
S	$Sulphur$	

N_2	$Nitrogen$	
CO_2	$Carbon\ Dioxide$	
H_2O	$Water$	
SO_2	$Sulphur\ Dioxide$	
CV	$Calorific\ Value$	kJ/kg
M, m	$Mass$	kg
V	$Voltage$	V
I	$Current$	A
R_{elec}	$Electrical\ Resistance$	Ohm
R_{th}	$Thermal\ Resistance$	K/W
ρ	$Density$	kg/m^3
μ	$Viscosity$	kg/ms
V	$Velocity$	m/s
Pr	$Prandtl\ Number$	
Re	$Reynold's\ Number$	
Nu	$Nusselt\ Number$	
D	$Diameter$	m
P	$Pressure$	kPa
RH	$Relative\ Humidity$	$\%$
V	$Volume$	m^3
n	$Number\ of\ Moles$	mol

R	$Ideal\ Gas\ Constant$	$kJ/mol\ K$
M	$Molar\ Mass$	kg/mol
ω	$Absolute\ Humidity$	kg/kg
H	$Enthalpy$	kJ
h	$Specific\ Enthalpy$	kJ/kg
r	$Discounted\ Rate$	$\%$
C	$Cash\ Flow$	R
T	$Time\ Period$	$years$

LIST OF ACRONYMS

AHU - Air Handling Unit

COP - Coefficient of Performance

FMCG - Fast Moving Consumer Goods

HFO - Heavy Fuel Oil

HVAC - Heating Ventilation and Air Conditioning

MFO - Marine Fuel Oil

NPV - Net Present Value

ORC - Organic Rankine Cycle

RH - Relative Humidity

SA - South Africa

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CHAPTER ONE – INTRODUCTION

1.1 Introduction

This chapter provides a brief overview of the subject matter organisation for the study, as well as the background to the problem that formed the foundation for this study. The research objectives are stated in accordance with the research questions that the objectives seek to answer. In this chapter, an overview of the format of the entire dissertation is briefly discussed and the content of the following chapter is stated.

1.2 Background to the Problem

Nestle is one of the leading Fast Moving Consumer Goods (FMCG) manufacturer in the world. Founded more than 150 years ago, it is the world's largest food and beverage company; represented in 191 countries around the world. Nestle employees are approximated at 328 000 globally. The entire operations consist of 418 factories around the globe with a single purpose of enhancing the quality of life and contributing to a healthier future. Offering a wide portfolio of products and services for people and their pets; Nestle boasts with more than 2 000 brands; ranging from global icons and local favourites.

The development by Henri Nestle of a life-saving infant cereal product laid the foundations for a company that has become the world leader in nutrition, health and wellness. At the heart of its wide portfolio, is Nestle South Africa (SA) that came into existence about 100 years ago. Focusing on milk powder processing and other secondary products, the on-going activities of the organisation have resulted in the company having a substantial portfolio for milk powder processing in SA. The milk powder processing operations component consists of four factories in total; strategically located to reap the benefits of low-cost supply and demand chain across SA.

The four milk powder processing operations have an installed capacity in excess of 150 000 tons per annum; servicing a diverse range of income earners and age groups in SA. With an installed processing capacity of approximately 150 000 tons per annum, Nestle SA has only been able to produce about 70% of installed processing capacity, annually, over the past five years. Failure to produce to installed capacity, together with growing consumer buying power has led to a loss of key businesses to competitors. This has originated from a number of factors, including both equipment related and human resource related issues. While the company has revised its human resource policies to assist with the training and development as well as the retention of employees; the ability to recover and reuse processing energy for efficiency and process optimization has not been tapped into. A thorough investigation into the possibility of recovering and reusing exhaust energy for process and energy demand optimization is necessary in order to improve production capacity in the future.

1.3 Problem Statement

The production capacity of the powder spray-drying process at Nestlé Harrismith cannot be increased any further as there is no excess energy available to allow for higher production capacities. Energy is generally the most expensive aspect of the milk powder factory. Heat energy is used in different stages of the process to raise the temperatures of the constituents of production to required levels and is a driving force of production. While the factory is unable to increase its energy capacity, there is a significant amount of exhaust air at high temperatures at the final stage of the process which is discharged to the atmosphere; which could be used as an energy source.

The energy-rich exhaust air discharged into the atmosphere causes powder particles to fall back down to the ground thus polluting the environment. The purpose of the study is to recover energy of exhaust air and identify cost-effective ways to use it to optimize productivity and energy demand of the process. This study will be based on identifying an efficient way of recovering heat from a milk powder spray-drying process. The significance of this exercise is that it will help salvage the energy that is lost to the atmosphere and use it as a substitute heating medium to reduce the overall factory energy generation.

1.4 Aim of the Study

The aim of the research is to assess and evaluate the benefit of recovering heat from the exhaust air of a powder spray-drying process with a hypothesis that heat energy in exhaust air can be used to optimize production and energy demand of the process.

1.5 Objectives of the Study

1.5.1 To determine the lost energy

An energy evaluation study will be conducted on the final stage of the drying process to evaluate the amount of energy that is lost to the atmosphere in the different phases of the process associated with the drying stage.

1.5.2 To evaluate overall energy consumption

Practical evaluations of steam usage in the entire process will be conducted. The emphasis will be on the quantity of steam used and temperature differentials to ascertain the overall steam energy consumed by the process.

1.5.3 To select the appropriate heat transfer method

Theoretical comparisons on viable heat sink techniques and methods of heat transfer technologies will be conducted to select an appropriate method that will best offer the desired results.

1.5.4 To examine the potential of optimization

Calculations on potential optimization of production capacity or energy demand of the process will be conducted to examine the increase in production capacity or decrease in energy demand as a result of using recovered heat energy from the exhaust.

1.5.5 To evaluate the cost of exhaust recovery

Calculations on the cost of recovering exhaust heat and using it for the process will be done to evaluate the monetary investments and benefits associated with the heat recovery option.

1.6 Research Questions

- How much energy is lost to the atmosphere?
- How much energy is consumed by the entire drying process?
- Which heat transfer methods are appropriate for the exhaust heat recovery?
- What impact will exhaust heat recovery have on production capacity and overall energy demand?
- What is the cost associated with the recovery of exhaust energy?

1.7 Significance of the Study

The research will contribute to energy saving initiatives worldwide as well as the reduction of greenhouse gases through the reduction of environmental pollution. The results of the study will be used by the organisation for optimization purposes in similar or related factories.

1.8 Format of the Study

The study consists of nine chapters structured as follows:

Chapter 1: Introduction

This chapter provides a brief overview of the organisation included in the study as well as the background to the problem underlying the study. It outlines the aim and the importance of the study. It explains the research objectives, formulates the research questions, and discusses the format of the study.

Chapter 2: Literature Review

Chapter 2 provides a comprehensive summary of the existing literature that was consulted. It focuses on the broader knowledge of previous studies that relate to the problems of interest in this study.

Chapter 3: Spray-Drying Process Overview

A complete detail of the spray-drying process related to the organisation in question as well as other modern organisations is discussed in this chapter with an intention of giving a holistic overview of the process.

Chapter 4: Lost Energy Evaluation

An energy evaluation study that will be conducted on the final stage of the drying process to evaluate the amount of energy that is lost to the atmosphere is detailed in this chapter.

Chapter 5: Overall Energy Consumption

Practical evaluation of steam usage in the entire drying process with emphasis on the quantity of steam used and temperature differentials to ascertain the overall energy consumed by the process will be outlined in this chapter.

Chapter 6: Heat Transfer Evaluation

This chapter discusses the various heat transfer technologies and heat sink medium that are viable and cost-effective for the recovery of exhaust heat energy.

Chapter 7: Production Optimization

This chapter discusses the potential optimization of production capacity or energy demand as a result of using recovered energy.

Chapter 8: Financial Analysis

The overall cost-benefit of the envisaged study will be discussed in this chapter with a focus on outlining the cost of recovering exhaust heat and the cost benefit as a result of optimization.

Chapter 9: Conclusions

This chapter concludes the study with a summary of research findings aimed at solving the research problem. The scope for future research areas is discussed and the conclusion for the entire study made.

1.9 Conclusion

Considering the turbulent environment in which the milk powder spray-drying processing organisations operate; it cannot be overlooked that there are benefits of having properly designed energy recovery systems aimed at optimization and future expansion for ensuring the survival of the business. In any organisation, with all factors remaining the same, recovered energy reduces the cost of new energy generation. Energy recovery is essential to providing a spare capacity for productivity when needed and thus ensuring the potential for optimization.

This chapter provided an overview of the entire study. The next chapter explores the literature that is relevant to this study.

CHAPTER TWO – LITERATURE REVIEW

2.1 Introduction

This chapter details the current literature on heat recovery initiatives. The purposive of which is to present an evaluation of the theoretical-based knowledge relevant to the research problem in order to present a link between the existing theory and the problem included in this study.

2.2 Heat Energy Recovery

In physics, energy is generally defined as an ability to do work. Span (2014) states that there are many different forms of energy – heat, light, sound, electrical, kinetic, potential, etc. and each form of energy may be transformed into another. Heat energy is related to the temperature of the matter; and for a given material and mass, the higher the temperature, the greater the heat energy (Incropera & De Witt, 2014). Heat energy flows from bodies of high temperature to bodies of low temperature and is measured in Joules (J) or kilojoules (kJ).

Increasing world population and industrialization are the two major factors that drive energy demands globally (Aneke, 2012). Fossil fuels, wind, solar, geothermal, etc. are amongst the different sources of energy. According to (Anderson, 2014), fossil fuels are most widely used due to their level of commercialization and technological advancements in their extraction. It is estimated that 80% of the global total energy is produced by burning fossil fuels (IEA, 2015). This indicates that fossil fuels have, and continue to play a major role in energy generation globally.

There has been an ongoing global concern on fully depending on fossil fuels for energy generation (Aneke, 2012). This is due to their exhaustible nature and detrimental effects on the environment. These global concerns call for the reduction of fossil fuel used and this can be achieved by energy saving and recycling (IEA, 2015). According to (Ahmad & Shafie, 2011) more than 90% of the energy related to greenhouse gases emission is as a result of fossil fuel exhaust emissions globally. The increase in the concentration of greenhouse gases emissions has caused a notable rise in temperature in the earth's atmosphere known as global warming (Singh, 2014).

Energy-intensive industries such as food, beverages, chemicals, paper and metal processing industries generate exhaust heat. According to (EUROSTAT, 2013) due to a large number of food and drinks processing industries globally, there is a greater potential for heat recovery in comparison to other industries. All boilers which burn fuel to produce heat create a large number of exhaust gases expelled outside at temperatures above 200^0 C (Brucker, 2015). According to (Anderson, 2014) condensing boilers have a built-in flue gas heat recovery system which forces some of the flue gases to condense into water, releasing latent heat which is recycled by the boiler.

Roughly one-third of the energy consumed by industries is discharged as thermal losses directly to the atmosphere (William, 2015). According to (William, 2015), these discharges are the result of process inefficiencies and the inability of the existing process to recover and use the waste energy. It is generally acknowledged that energy losses from the exhaust system are approximately 30% of the total losses of energy supplied to the internal combustion engine (Walsh, 2011). If some of this lost heat could be recovered and converted into useful heat, this could supplement the vehicles' energy demand and improve the overall efficiency of the system (Miller, Hendricks, Wang & Peterson, 2015).

In defining waste heat, (Allen & Backtrom, 2012) denote that it is the heat that is generated in a process by way of fuel consumption or chemical reaction, and is then dumped into the environment even though it could still be re-used for some useful and economic purposes. The essential aspect of heat energy is not the amount but its quantity or value (Span, 2014). A large quantity of hot flue gases is generated from boilers, kilns, ovens and furnaces. If some of this waste heat could be recovered, a considerable amount of primary fuel could be saved (Bai & Xu, 2011). Bai & Xu (2011) further assert that the energy lost in waste gases cannot be fully recovered. The strategy of heat recovery depends in part on the temperature of waste heat gases and the economics involved.

Waste heat can be rejected at virtually any temperature, usually, the higher the temperature at which heat is rejected, the higher the quality or value, and more cost-effective is the heat recovery (Becker, 1989). According to (Miller, et al., 2015), the key to waste heat recovery studies, is the necessity to use recovered heat energy. Typical examples of uses of waste heat in industries include pre-heating of combustion air in boilers, pre-heating of boiler feed-water, electricity generation, and provision of energy for heating, ventilation, and air conditioning (HVAC) systems. An idea of the use to which the waste heat will be put helps in selecting the best waste heat recovery technique for any given process (Aneke, 2012). In many developed countries, waste heat is used to further scientific and technological activities for the development of research and education programmes (Bai & Xu, 2011).

The rises in the price of energy and government environmental regulations are other driving forces for energy recovery. Regardless of the economic and environmental benefits of waste heat recovery, it is also thermodynamically important to recover waste heat from process industries (Aneke, 2012). The new concept of thermo-dynamical importance relates to entropy generation minimization (Brucker, 2015). According to the second law of thermodynamics, the entropy of an isolated system can never decrease (Kotas, 1985). For systems which interact with the environment, the increase in entropy gives rise to loss of performance (Bai & Xu, 2011). A loss in performance occurs as the system generates entropy into the environment as a result of heat loss; and it can be reduced by reducing any energy losses in the system (Kotas, 1985).

There are many ways to recover waste heat from the process. These include direct heat recovery, cascade system of recovery, absorption chillers and Rankine Cycle systems. A cascade system of heat recovery is considered the best practice that ensures that the maximum amount of energy is recovered at the highest potential (Smith, 2005). In energy cascading systems, the recovered heat is consumed efficiently when energy is firstly used for high-temperature applications. The heat rejected from high-temperature applications can then be used for low-temperature applications. This ensures efficient utilization of recovered heat, and, is subsequently economical for waste heat recovery system design and maintenance. In process plants where heat integration is the highest priority, the methods of direct heat recovery and cascade system of recovery are likely adopted.

The stages in the design of heat energy recovery systems include data extraction, performance targets, process modification, network design, design evolution and process simulation (Brucker, 2015). According to (Brucker, 2015), process modification is vital in ensuring that the scope for heat recovery is improved. Heat integration plays an important role in ensuring effective and efficient waste heat energy recovery (Laval, 2016). The basic definition of heat process integration is the physical arrangement of production plants, process equipment, process sections or even surroundings aimed at maximizing energy efficiency. Applying the stages of heat recovery systematically ensures that maximum heat energy is efficiently recovered (Brucker, 2015).

In any heat recovery situation, it is essential to know the amount of sensible heat recoverable. The generic formula for the amount of sensible heat is given below:

$$Q = \dot{m} \times c_p \times \Delta T \tag{2.1}$$

Where,

$Q\ is\ the\ heat\ content\ in\ kJ/s$

$\dot{m}\ is\ the\ mass\ flow\ rate\ in\ kg/s$

$c_p\ is\ the\ specific\ heat\ of\ a\ substance\ in\ kJ/kgK$

$\Delta T\ is\ the\ temperature\ difference\ in\ K$

From the above equation, it can be clearly seen that the heat quantity for a specific substance is dependent on the mass flow of that substance and the temperature difference.

According to (Law, 2014) one of the most useful tools for waste heat recovery system design is pinch analysis. It is asserted by (Hindmarsh, 1983), that pinch technology was introduced by Linnhoff and Vredeveld to represent a new set of thermodynamically based methods that guarantee minimum energy levels in the design of heat exchangers. The technology has emerged as an unconventional development in process design and energy consumption. Pinch technology helps to create an overall view of the heat demand of a process and highlight opportunities for heat recovery between hot and cold streams (Law, 2014).

The term pinch analysis is often used to represent the application of the tools and algorithms for studying industrial process energy utilization. The steps involved in pinch analysis include identification of hot and cold streams in the process, thermal data extraction for a process, selection of initial minimum temperature difference value, estimating energy costs targets, estimating heat exchanger capital cost and designing heat exchanger network (Law, 2014). The application of pinch technology has resulted in significant improvements in the energy efficiency of industrial facilities globally. Continuous and batch processes can be analysed using pinch technology on an individual basis and plant-wide basis (Aneke, 2012).

According to (Linnhoff & Polley, 1988), the first law of thermodynamics provides the energy equation for calculating the enthalpy changes in the streams passing through a heat exchanger. Heat flows only in the direction of hot to cold, as per the second law of thermodynamics. In a heat exchanger unit, neither a hot stream can be cooled to a temperature below cold stream supply temperature, nor can a cold stream be heated to a temperature above the hot stream supply temperature (Linnhoff, 1989).

In practice, the hot stream can only be cooled to a temperature defined by the temperature approach of the heat exchanger (Law, 2014). The temperature approach is the minimum allowable temperature difference in the stream temperature profiles for the heat exchanger unit. The point at which the minimum temperature difference is observed in the process is referred to as the pinch point (Linnhoff & Polley, 1988). The pinch point defines the minimum driving force allowed in the heat exchange unit. The benefits of pinch analysis include an ability to modify process flow diagrams, set practical layout targets, debottlenecking the process, and ability to use low-grade waste heat effectively and efficiently (Linnhoff, 1989).

The objective of pinch analysis is to identify energy cost and heat exchanger network capital cost targets for a process (Linnhoff, 1989). The prime objective of pinch analysis is to achieve financial savings by better process heat integration; maximizing process-to-process heat recovery and reducing the external utility loads (Hindmarsh, 1983). According to (Law, 2014), when the process involves single hot and cold streams, it is easy to design an optimum recovery heat exchanger. When there are a number of hot and cold streams, there is a need to construct a composite curve by including the enthalpy changes of the streams in the respective temperature intervals (Law, 2014).

According to (Nimbalkar, 2015) before deciding on a method of waste heat recovery, issues such as temperature, mass or volume flow rate and availability should be considered. Presence of particulate matter in flue gases, corrosive compounds and combustible vapours should be analysed before embarking on the journey of choosing the heat recovery equipment (ITP, 2008). Thermal cycling is another important phenomenon to be considered for heat recovery equipment selection. Heat flow in some industrial processes varies drastically, creating mechanical and chemical stress in equipment.

There are several interrelated barriers that impede the installations of waste heat recovery systems in process organisations. According to (ITP, 2008), the cost of heat recovery equipment, auxiliary systems, and design services lead to long payback periods in some applications. Certain applications require advanced and more costly materials. Economies-of-scale as well as operational and maintenance efforts present limitations to effective heat recovery systems in process plants (Hugo, 2017). Lack of viable end-use presents the major barrier to the advancement of waste heat recovery (ITP, 2008). Many process plants do not have an on-site use for low-grade heat.

2.3 Heat Energy Efficiency

Energy assessments are useful in identifying where and how much energy is consumed by an existing process (Fuchs, 2012). Reduction in energy consumption will most likely result in the reduction of operational cost. According to (Koch, 2013), historical energy consumption data, metered consumption, energy system name-plate data, weather data and building configurations are essential data sources for a thermal energy consumption audit. Processing industries energy consumption account for a substantial share of the overall operating costs. Saving energy through energy efficiency improvements can cost less than generating, transmitting and distributing energy from power plants, and provides multiple economic and environmental benefits (Kempton, 2014).

Many organisations have not moved towards measuring energy consumption throughout their plants. According to (Aneke, 2012), the implementation of energy-efficient consumption in many factories is often very low. Energy efficiency can be defined as a ratio between an output of performance or energy, and an input of energy (EUC, 2006). Another term closely related to energy efficiency is that of eco-efficiency. Eco-efficiency is commonly used to measure the ecological sustainability. Fuchs (2012) states that eco-efficiency is the amount of waste related to energy production or use. Organisations that do not monitor their energy consumptions are most likely to contribute significant disadvantages to natural resources (EUC, 2006). Energy consumption measurements make it possible for organisations to measure the amount of emissions generated by their operations.

Forsstrom, Lahti & Wahlgren (2011); denote that for processes where the output is not measurable as energy, a measure has to be devised for the output that properly describes the need that causes the demand for the energy. EUC (2006) calls such a measure, a demand indicator. Indicators of energy efficiency in practical use often take the form of energy intensity (Forsstrom, et al., 2011).

$$Energy\ Intensity = \frac{Energy\ Input}{Demand\ Indicator} \tag{2.2}$$

Since energy intensity measures the rate of energy consumption per outcome, it answers to the basic question at the root of energy efficiency: how much energy consumption is needed to achieve the desired results (Forsstrom, et al., 2011). Thermodynamic indicators that look into energy consumption and economic indicators that measure energy in monetary terms are widely used indicators to calculate the energy intensity of the process (Patterson, 1996). Demand indicators have limitations, and they are thus used for general approximation of energy efficiency (EUC, 2006).

In studying the principles for evaluating energy efficiency potentials, (Patterson, 1996) found that the degradation of energy and evaluation of the temperature difference between the process and the environment are of prime importance. The maximum theoretical efficiency that can be achieved is defined by the Carnot cycle efficiency equation:

$$\eta_c = 1 - \frac{T_{min}}{T_{max}} \tag{2.3}$$

When maximizing the efficiency of the process, the temperature difference between the maximum temperature of the process (T_{max}) and the minimum temperature of the process (T_{min}) must be as high as possible (EUC, 2006). According to (Koch, 2013), energy efficiency potentials in systems exist in theoretical and technical spheres. Theoretical energy efficiency represents the potential that exists as per the Carnot cycle efficiency equation above, and technical energy efficiency represents the potential that can be reached using technology.

There are many interventions that industries which are using boilers for thermal energy generation could apply to improve efficiency. Coetzee (2016) suggests that a 22 ^{0}C reduction in flue gas temperature of the boiler generally yields 1% increase in boiler efficiency. According to (Haas, 1997) improving oxygen trim control is essential in improving boiler efficiency. It is asserted by (Coetzee, 2016) that 5% reduction in excess air or 1% reduction of residual oxygen in stack gas increases boiler efficiency by 1%. It is hypothesized that scale and sediment on the water tubes of the boiler are the biggest cause of reduced efficiencies in boiler operation. A 1 mm thick scale on the boiler water tube could increase fuel consumption by 5 to 8% (Haas, 1997). A 3 mm thick soot deposition on the heat transfer surface can cause an increase in fuel consumption of about 2.5% (Coetzee, 2016).

Coetzee (2016) states that industries who apply continuous energy management programmes on thermal utilities realise improved efficiencies. Establishing a boiler efficiency-maintenance programme should begin with an energy audit (Hugo, 2017). Use of variable speed drives for boiler combustion air fans, clean burners, inspection for proper temperatures, automated blow-down and optimized deaerator venting are some improvements that can be made part of the boiler efficiency-maintenance programme to increase boiler efficiency (Coetzee, 2016). A study on the part-load characteristics to determine the most efficient mode for operating multiple boilers is at the heart of improving boiler efficiency (Coetzee, 2016).

Steam systems that are designed and maintained for efficiency yield a significant monetary benefit to process plants. In many process plants, steam and condensate leaks are ignored. According to (BEE, 2014), a 3 mm diameter hole on a pipeline carrying 7 kg/cm^2 steam would waste 33 000 litres of fuel oil per year. Condensate return or re-using it in the process is very crucial for boiler efficiency. Coetzee (2016) states that 6 ^{0}C rise in feed water temperature by condensate recovery corresponds to a 1% saving in fuel consumption in a boiler. Air removal is also essential from indirect steam-using equipment because 0.25 mm thick air film offers the same resistance to heat transfer as a 330 mm thick copper wall (BEE, 2014). Ensuring that process temperatures are correctly controlled and maintaining the lowest acceptable process steam pressures have direct impacts on boiler efficiency (Hugo, 2017).

There is room for improvements in efficiencies of thermal utilities such as furnaces (BEE, 2014). Ensuring that the furnace combustion chamber is under slight positive pressure is essential in maintaining the efficiency in process plants where furnaces are used. Infiltration of air, control of carbon dioxide or carbon monoxide to optimum levels improves the efficiency of the furnace (Haas, 1997). Combustion control and instrumentation are essential in improving combustion in furnaces. According to (BEE, 2014), matching the load to furnace capacity, reduction in cycle times, temperature controllers and flame control are some of the basic operational measures for efficiency improvements in furnaces.

At the heart of heat energy efficiency improvements in a reticulation system, is the issue of thermal insulation. Insulations are defined as those materials or combinations of materials which retard the flow of heat energy by conserving energy through reducing heat loss or gain or facilitating temperature control of the process (Eastop & Croft, 1990). To reduce heat losses in a thermal system to a minimum, the system can be heat insulated (Botes, 2007). Thermal insulation has a direct positive impact on the energy consumption. By lagging the systems by means of some form of heat insulating material, heat losses can be controlled. Botes (2007) affirms that any surface which is hotter than its surroundings will lose heat. The heat loss depends on many factors, but the surface temperature and its size are dominant.

There are many different types of insulation materials and systems available, each having different thermal performances, handling properties, and other characteristics. It is vital that thermal properties are primarily considered in choosing insulation material (Yuen, 2004). Temperature limits, thermal conductivity, emissivity, thermal resistance and the overall conductance of heat flow through a system are some of the most common critical thermal properties to be considered for material selection (Eastop & Croft, 1990). According to (Botes, 2007), breaking load, chemical reaction, chemical resistance, coefficient of expansion and contraction, combustibility, compressive strength and density are the most important mechanical and chemical properties to be considered in selecting insulating material.

It is critical that damaged insulation is repaired and kept in a good state to ensure heat efficiency of the system. Coetzee (2016) asserts that a bare steam pipe of 150 mm diameter and 100 m length, carrying saturated steam at 8 kg/cm^2 would waste 25 000 litres of furnace oil in a year for an oil-fuelled boiler. It is further hypothesized that on open tanks, 70% heat losses can be reduced by floating a layer of 45 mm diameter plastic balls on the surface of approximately 90 ^{0}C hot condensate (BEE, 2014). Developments of Fourier’s equation have led to the ability to determine heat loss to the environment in pipelines through conduction; and selecting the appropriate material (Botes, 2007). Botes (2007) proposes the equation for calculating heat losses for lagged pipes as below:

$$Q_L = \frac{2 \times \pi \times l \times (t_s - t_e)}{\left(\frac{1}{u_1 r_1}\right) + (\frac{1}{u_2 r_3}) + \frac{1}{k_1}\ln(\frac{r_2}{r_1}) + \frac{1}{k_2}\ln(\frac{r_3}{r_2})} \tag{2.4}$$

Where,

$Q_L\ is\ the\ heat\ loss\ in\ J/s$

$l\ is\ the\ lagged\ pipe\ length\ in\ m$

$t_s\ is\ the\ temperature\ of\ a\ substance\ in\ °C$

$t_e\ is\ the\ temperature\ of\ the\ environment\ in\ °C$

u_1 $is\ the\ convective\ heat\ transfer\ coefficient\ of\ fluid\ in\ W/m^2\ °C$

u_2 $is\ the\ convective\ heat\ transfer\ coefficient\ of\ outside\ air\ in\ W/m^2\ °C$

k_1 $is\ the\ thermal\ conductivity\ of\ piping\ material\ in\ W/m\ °C$

k_2 $is\ the\ thermal\ conductivity\ of\ insulation\ material\ in\ W/m\ °C$

r_1 $is\ the\ inner\ radius\ of\ bare\ pipe\ in\ m$

r_2 $is\ the\ outer\ radius\ of\ bare\ pipe\ in\ m$

r_3 $is\ the\ outer\ radius\ of\ lagged\ pipe\ in\ m$

From the equation above, it can be concluded that insulation materials with lower thermal conductivity, k, together with a lower heat transfer coefficient, u, will result in less heat losses in pipelines.

In process industries, there are potentials for improving energy efficiency through electrical utilities. Electrical distribution systems can contribute to overall plant energy efficiency if they maintain a higher load factor (Greenwood, 2013). Properly sizing the load for optimum efficiency is important in the electrical system. High-efficiency motors offer about 4 to 5% reduction in energy consumption than the standard motors of the same size (Beggs, 2002). It is stated that provision of proper ventilation is important in maximizing the efficiency of operating motors. For every 10 ^{0}C increase in motor operating temperature over a recommended peak, the life of the motor is estimated to be reduced by half (BEE, 2014). Beggs (2002) says that if rewinding is not done properly, the energy consumption can be increased by 5 to 8%.

Compressed air systems are generally used in process plants and they have a huge contribution to energy efficiency. Reduction of air compressor discharge pressure to the lowest acceptable setting is considered important in improving the efficiency of the energy system. Reduction of 1 kg/cm^2 air pressure would result in a 9% reduction on input power and contributes about 10% to compressed air leakages reduction (BEE, 2014). Greenwood (2013) suggests that compressed air leak from 1 mm hole size at 7 kg/cm^2 pressure is equivalent to a loss of 0.5 kW of energy. Ensuring that compressor intake air is coolest as possible but not air-conditioned, will save a great amount of energy throughout the plant.

Every 5 ^{0}C reduction in intake air temperature would result in a 1% reduction in compressor energy consumption (Beggs, 2002).

Process plants can drive energy efficiency throughout the plant by focusing on each area of energy consumption (Hugo, 2017). For process plants that make use of chilled water, using the lowest temperature condenser water that the chillers can handle is important. A reduction of condensing temperature by 5.5 ^{0}C results in about 20 to 25% reduction in compressor power consumption (Beggs, 2002). Furthermore, 1 mm scale build up on condenser tubes can increase the energy consumption by 40% (BEE, 2014). Increasing the chilled water temperature set-point to match end-user requirements and optimizing condenser and refrigeration water flow rates are some of the basic operational interventions that can be employed for running chillers at as efficient levels as possible (Hugo, 2017). Areas such as HVAC, refrigeration, cooling towers, and buildings lighting contribute very small amounts to plant energy consumption; however, a combination of the small amounts of energy consumption can prove to be a huge area for improvement in overall energy efficiency of the plant (Beggs, 2002).

2.4 Heat Transfer

Incropera & De Witt (2014) define heat transfer as a study of the exchange of thermal energy through a body or between bodies which occur when there is a temperature difference. There are three types of heat transfer namely conduction, convection and radiation. According to (Riffat, 2016), conduction is the transfer of heat through solids or stationary fluids. Convection uses the movement of fluids to transfer heat. Radiation does not require a medium for transferring heat; this mode uses electromagnetic radiation emitted by an object for exchanging heat. The advantages and disadvantages of the three heat transfer methods are based on the use of a heat transfer method (Riffat, 2016).

Conduction is a process of heat transfer generated by lattice vibration and molecular collision within the object (Holman, 2009). In solids, conduction occurs by a combination of these two mechanisms; and in stationary fluids, heat is conducted primarily by molecular collision (Riffat, 2016). Conduction through molecular collision is more effective than through lattice vibration; this is why metals are generally better heat conductors than ceramic materials which do not have many free molecules (Holman, 2009). The equation for calculation of conduction heat transfer rate is known as Fourier's Law of Heat Conduction.

$$Q_c = -k \times A \times \frac{\Delta T}{\Delta r} \tag{2.5}$$

Where,

Q_c is the conductive heat transfer rate in W

k is the thermal conductivity of the material in $W/m\,°C$

A is the cross − sectional area in m^2

ΔT is the temperature difference of surfaces in $°C$

Δr is the distance separating the surfaces in m

The effectiveness by which heat is transferred through a material is measured by thermal conductivity (Incropera & De Witt, 2014). In heat transfer, positive heat conduction means that heat is flowing into the body in question, and negative heat conduction represents heat leaving the body.

Heat transfer has been applied in various research fields mainly to increase the heat transfer rate, decrease heat transfer rate and keep the temperature in a certain range (Nave, 2015). In industrial applications, the convection method of heat transfer is dominant for heating or cooling of substances (Riffat, 2016). In the convection method of heat transfer, the mechanism between the fluid and the surface may be natural or forced (Incropera & De Witt, 2014). Natural convection refers to a case where the fluid movement is created by the fluid itself. Forced convection uses external means of producing fluid movement. Conveying heat energy due to natural convection is limited; therefore, forced convection provides some immediate solution to process plants (Nave, 2015). The basic formula for calculating the convective heat transfer rate was proposed by Isaac Newton, known as the Newton's Law of Cooling (Nave, 2015).

$$Q_{conv} = -h \times A \times (T_2 - T_1) \tag{2.6}$$

Where,

Q_{conv} is the convective heat transfer rate in W

h is the convection heat transfer coefficient in $W/m^2\,°C$

A is the cross − sectional area in m^2

T_2 is the temperature of the surface in $°C$

T_1 is the temperature of a fluid in $°C$

The convection heat transfer coefficient measures how effectively fluids transfer heat by convection. It is determined by factors such as the fluid density, viscosity and velocity (Incropera & De Witt, 2014). According to (Nave, 2015), for a flowing fluid, convection works best in surfaces where there is a certain degree of roughness. The roughness elements contribute to increased heat transfer by increasing the level of turbulence (Nave, 2015).

For conduction and convection heat transfer, heat can only be transferred through a material medium; whereas, radiation heat transfer can exist in a vacuum (Ryan, 2013). Energy transfer through electromagnetic waves is called radiation, and thermal radiation is propagated by any matter with the temperature above zero degrees Kelvin (Fox & McDonald, 2009). Radiative heat transfer occurs when the emitted radiation strikes another body and is absorbed. According to (Yang, 2016), the electromagnetic spectrum classifies radiation according to wavelengths of the radiation, and radiation with shorter wavelengths are more energetic and contains more energy.

According to (Incropera & De Witt, 2014), a concept of the emissive power was given by Holman as the energy emitted by the body per unit area and per unit time. From this, the Stefan-Boltzmann Law of Thermal Radiation describes the amount of radiation emitted as a relationship between the emissive power and temperature. The equation is as follows:

$$Q_e = \varepsilon \times \sigma \times A \times T^4 \tag{2.7}$$

Where,

Q_e is the amount of radiation emitted in W

ε is the material emissivity ranging from $(0$ to $1)$

σ is the Stefan − Boltzmann constant equal to $5.67 \times 10^{-8}\ W/m^2K^4$

A is the radiating surface area in m^2

$T\ is\ the\ temperature\ of\ the\ body\ in\ K$

The emitted radiation strikes the second surface, where it is reflected, absorbed or transmitted (Ryan, 2013). The portion that contributes to the heating of the surface is the absorbed radiation.

2.5 Heat Transfer Fluids

According to (Lenert, Nam & Wang, 2013) heat transfer fluids refer to a gas or liquid that is used in a heat exchange arrangement specifically for the purpose of transmitting heat from one system to another. A coolant that flows through a device in order to prevent it from overheating is a heat transfer fluid (Lenert, et al., 2013). Heat transfer fluids serve the purpose of transporting heat from the supplier to the user. In a typical industrial application, heat transfer fluids are required to have a low freezing point, high operating temperatures, high thermal conductivity, high volumetric heat capacity, low viscosity, non-corrosive, low flash point, environmentally safe and cost-effective (Holman, 2009). In plant process applications, many constraints, both performance and practical, influence the choice of the heat transfer fluid.

Gaseous heat transfer fluids are suited for high-temperature applications (Incropera & De Witt, 2014). Exhaust gases can be recovered and thus render high thermal system efficiency. Nave (2015) states that gaseous heat transfer fluids, however, have low heat transfer coefficients and low densities. Air is considered by far as the most investigated gaseous heat transfer fluid. According to (Fox & McDonald, 2009), air is favoured because it is practically infinitely available, theoretically free of cost, will not freeze or boil, and is non-corrosive. The downside of air is that it has a low heat capacity and tends to leak out of ducts (Lenert, et al., 2013). Other gases such as helium or nitrogen could also be used as heat transfer fluids for their superior heat transfer and flow characteristics compared to air (Nave, 2015).

Oil and water are two common liquids predominately used in industrial applications as heat transfer fluids. Water is favoured due to high specific heat, very low viscosity, inexpensive, non-toxic and easy to pump (Riffat, 2016). Water, however, has a relatively low boiling point, can cause deposits if not rid of minerals, and can also be corrosive if the pH is not maintained (Lenert, et al., 2013). Oils have been the most common heat transfer fluid due to their high stability over a relatively wide temperature range (Riffat, 2016). Oil has higher viscosity and a lower specific heat than water. The disadvantages of oil are that it is degradable over time, the cost is very high, oils are highly flammable and require more energy to pump (Riffat, 2016).

Perry (2014) suggests that steam globally dominates as the heat transfer fluid in process plants. The cost of steam which includes fuel, water used, water treatment, pumping energy, combustion, maintenance and labour is the prime driver for researchers to look into alternative effective heat transfer fluids (Perry, 2014). Lenert, et al., (2013) conducted an experiment to compare different heat transfer fluids and concluded that molten salts perform better than oil-based fluids; however, issues of proper materials for piping and freeze protection need to be considered. The choice of proper heat transfer fluid is essential for the development of high-efficiency process plants, as it determines the thermal and hydraulic performance of the heat exchange equipment (Incropera & De Witt, 2014). The performance of heat transfer fluids is determined by the combination of the properties rather than by each of the individual properties (Perry, 2014).

In the recent years, the research on heat transfer enhancement through heat transfer fluids has been focusing on nanofluids. According to (Yang, Li & Nakayama., 2013), nanofluids are heat transfer fluids containing nanometre-sized particles in suspension. A possible effective way of improving the heat transfer performance of common fluids is to suspend various types of small solid particles, such as metallic and non-metallic particles, in conventional fluids, such as water and mineral oils to form colloidal solutions (Yang, et al., 2013). The enhancement of heat transfer using nanofluids has been used as one of the passive heat transfer techniques in several heat transfer applications (Hussein & Sharma, 2014). Nanofluids are considered to have great potential for heat transfer enhancement and are highly suited to application in processes like microelectronics, fuel cells, pharmaceutical processes, hybrid-powered engines, engine cooling, chillers and heat exchangers (Mahmood, 2015).

Shedid (2014) states that the poor thermal properties of convective heat transfer fluid act as the main barrier to the growth of energy efficient heat exchangers. The conventional method for increasing heat dissipation is to increase the area available for exchanging heat, however, this approach involves an undesirable increase in the size of a thermal management system (Yang, 2016). The innovative concept of nanofluids has been proposed as a prospect for these challenges. The specific heat of nanofluids typically increases or decreases with increasing particle loading depending on whether the nanoparticles have a higher or lower capacity relative to the base fluid (Yang, et al., 2013). Thermal conductivity and effective viscosity of nanofluids depend strongly on the volume fraction of particles in suspension (Mahmood, 2015).

Nanofluids cause drastic changes in the properties of the base fluids. The benefits of nanofluids include minimum pressure drop due to nanoparticles and higher thermal conductivity (Hussein & Sharma, 2014). It is hypothesized that successful employment of nanofluids leads to lighter and smaller heat exchangers. This results in significant energy and cost savings. According to (Shedid, 2014), heat transfer rate increases due to a large surface area of nanoparticles in the heat transfer fluid. Nanofluids are said to be suitable for rapid heating and cooling applications. The stability of nanofluids and their production costs are the major factors that hinder the commercialization of nanofluids (Mahmood, 2015). Experiments proved that the heat transfer rate of nanofluids is directly proportional to the Reynolds number of the nanofluids (Yang, et al., 2013). Spherical shaped nanoparticles were found to increase the heat transfer rate of nanofluids compared to other shapes (Shedid, 2014).

2.6 Heat Recovery in Spray-Drying

The manufacturing of milk powder is one of the highest energy consuming processes. It is hypothesized that the specific heat consumption of spray-drying is very high, approximately 4200 kJ are needed to remove 1 kg of water (Koninklijke, 2007). Atkins, Walmsley & Neale (2011) confirm that spray drying of milk powder is an energy-intensive process and there remains a significant opportunity to reduce energy consumption by applying process integration principles. Energy losses in spray-drying occur at several places, but most energy is lost with the exhaust drying air (Jansen, Steenbergen & Bergers, 1981). The ability to optimally integrate the drying process with the other processing steps has the potential to improve the overall efficiency of the entire process, especially when exhaust heat recovery is considered (Atkins, et al., 2011).

The warm and humid air leaving the spray dryer contains a small portion of potentially recoverable sensible heat and a large portion of latent heat that is impractical to recover by direct condensation due to the low dew point of this stream and due to the presence of milk powder particles that become sticky at high relative humidity values (Atkins, et al., 2011). Heat recovery of the energy is a possibility and can reduce significant energy losses.

According to (Kemp, 2012), integrating the dryer with the rest of the process can increase the viability of exhaust recovery. Hot utility use can be reduced by as much as 21% if suitable heat sinks are selected (Atkins, et al., 2011). The most simple and economical way to use the recovered heat is pre-heating of the fresh drying air (Goula & Adamopoulos, 2005). During heat recovery simulation researchers found that flow resistance of exhaust heat is dependent on powder and moisture content (Golman & Julklang, 2014).

There has been a limited success with heat recovery from milk spray dryer exhausts with recuperators (Atkins, et al., 2011). According to (Kemp, 2012) typical milk powder spray-dryers are not integrated whatsoever due to three main factors: economics, particle loading and fouling, and relatively low exhaust temperatures. The cost that can be cut by making use of exhaust air as a heating medium in dehumidifiers as opposed to using steam is the biggest motivator for an exhaust air heat recovery system (Hugo, 2017). Exhaust air recovery systems can be cost-effective if made simple (Bilic & Glavas, 2012). The set-back of installing a heat recovery system for exhaust air in spray-dryers lies in the fact that chemical cleaning is required to remove fouling (Shakeel, 2017). Pre-heating of fresh air with powder and moisture content of exhaust air results in serious fouling because of small particles of powder passing through (Zhong & Hua, 2003). Due to hygienic consequences, care must be taken in choosing the heat exchanger to be used (Jansen, et al., 1981).

2.7 Conclusion

This chapter reviewed the relevant literature to gain a better and in-depth understanding of the factors affecting waste heat recovery in industrial applications. From this chapter, it can be concluded that researchers have studied the possibilities of recovering low-grade waste heat for efficiency improvements and information on different technology application for improvement is readily available. The issues of the cost of the waste heat recovery system and areas of end-use were discovered to be the underlying challenges in the effectiveness of low-grade waste heat recovery systems adoption by processing industries.

The next chapter discusses the milk powder spray-drying process in order to outline the operational characteristics of spray-drying.

CHAPTER THREE – MILK POWDER SPRAY-DRYING PROCESS

3.1 Introduction

This chapter details an in-depth understanding of the milk powder spray drying process. The scope of this chapter includes the steps through which raw materials are transformed into a final product. In order to effectively conduct the experimental work on waste heat recovery in the milk powder spray-drying process, it is necessary to understand the required operational characteristics and process conditions of the entire process. Fundamental literature in relation to the technology used in the process is briefly discussed to form a link between the existing technology and the currently used techniques in the process. The most important scope of this chapter is to outline the heat sinks and sources throughout the process steps involved in the manufacturing of the final product.

3.2 Milk Powder Spray-Drying History

Spray-drying has its origin in the United States since the first patented design was registered there in 1872 (Padma, 2015). Samuel Percy is credited with the description of the principle of spray-drying and is considered to be the inventor of the spray-drying technology. According to (Anu, 2014), the first industrial spray dryer was installed in 1905 by the US company by the name of Merril Soul. Rapid developments of dairy industries due to increased production of milk brought significant improvement of milk spray dryers. The spray-drying process has been developed for use in a wide range of industrial applications where the production of a free-flowing powder is required. This method of dehydration has become the most successful in the production of powdered soaps, powdered milk, instant coffee, corn-starch, pharmaceuticals and flavours (Huang, 2013). Throughout the years, this perpetual process has exhibited a huge growth with innumerable innovations in terms of its operational; design and widely varied applications (Crosby & Marshall, 2013).

By definition, spray-drying is the transformation of feed from a fluid state into a dried particulate form by spraying the feed into a hot drying medium (Anu, 2014). In spray drying, concentrated milk is sprayed into the hot air, whereas foam-spray drying sends a jet of hot air into concentrated milk (Bilic & Glavas, 2012). The basic idea of spray drying is the production of highly dispersed powders from a fluid feed by evaporating the solvent. This is achieved by mixing heated air with a sprayed fluid of high surface-to-mass ratio droplets, ideally of equal size, within a drying vessel, causing the solvent to evaporate uniformly and quickly through direct contact (Padma, 2015).

Spray-drying stands out from other processes involving liquid drying by its ability to handle feedstock of varying nature; producing flowing powders of specific particle size, high productivity and versatile applications (Crosby & Marshall, 2013). Spray-drying advantages over roller drying are immediately evident in terms of the less destructive nature of thermal treatments involved (Anu, 2014). Almost all other methods of drying produce a mass of material requiring further processing. Spray-drying, on the other hand, offers a very flexible control over powder particle properties such as density, size, flow characteristics and moisture content (Hugo, 2017). One purpose of drying milk is to preserve it; milk powder has a far longer shelf life than liquid milk and does not need to be refrigerated.

The dairy industry is the major food sector utilizing the spray-drying technique in order to meet the ever-increasing demand for dried milk products (Bilic & Glavas, 2012). Milk powder manufacture is a simple process, able to be performed on a large scale. Production involves the gentle removal of water at the lowest possible cost under stringent hygiene conditions while retaining all the desirable natural properties of the milk such as colour, flavour, solubility and nutritional value (Crosby & Marshall, 2013). The process involves three main steps namely: Standardization, Concentration and Spray-Drying. Understanding the process steps, along with the hardware systems involved in it, will enable visualization of the operation at a glimpse of a reading.

3.3 Standardization

The first main step in milk powder spray-drying process is the process known as standardization. This process ensures that the resulting composition key parameters are standardized as per the requirements. The process of standardization ensures that the pH, fat content, protein content, vitamins and minerals are of the correct ratios as required. The process of standardization consists of four consecutive sub-steps: Tipping, Conveying, Dry mixing and Wet mixing.

3.3.1 Tipping

Different ingredients to be used for standardization come from suppliers in dehydrated powdered form and are tipped into different tipping hoppers for primary storage. Tipping stations are ergonomically designed to provide safe handling of sacks whilst controlling dust emissions (Gericke, 2015). The operator places the sack onto a rest platform, which is enclosed within an extractor cabinet. The contents are tipped into a hopper, and there is an extraction fan provided to ensure that airborne dust is carried away from the operator.

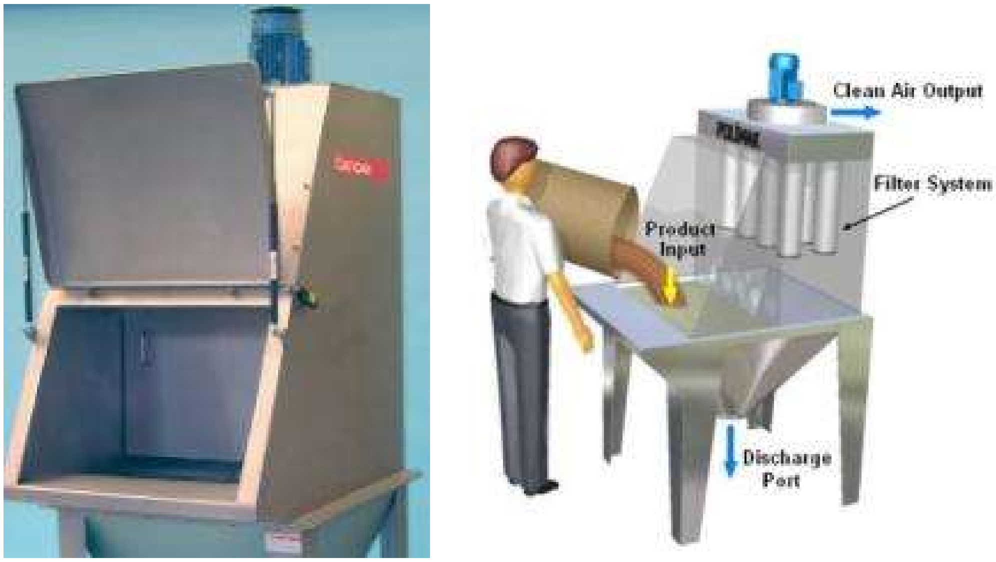


Figure 3.1: A typical tipping hopper and its operation **(Gericke, 2015)**.

The discharge port is aided with sifting devices, to remove oversized particles and extraneous material (GEA, 2016). Two types of sifters are usually used in powder handling, the rotary sifter and vibrating sifter. The rotary sifter consists of the rotor with flights which rotates around the sieve, and the vibrating sifter uses a vibrating platform to sift out oversized particles. The advantage of the rotary sifter over vibrating sifter is effective sifting and crushing of oversized particles. However, the rotary sifter presents the dangers of metal to metal contact as the flights can touch the sieve cage if misalignment occurs. This presents a huge quality issue in production, and as such, vibrating sifters are preferred as they eliminate this quality compromising scenario.

3.3.2 Conveying

In order to convey ingredients from the primary storage hopper to the main hopper, a pneumatic conveying system is used. Pneumatic conveying systems use an air stream to move dry bulk powders through a combination of horizontal and vertical pipelines in a powder processing system (PSG, 2018). According to (Buhler, 2013), pneumatic conveyors are characterised as pressure or vacuum systems; and either use a dilute phase or dense phase technology to transport powdered material. A dilute-phase conveying system uses high-volume and low-pressure air to keep powder suspended all the way through to its destination; and in a dense phase conveying systems, low-volume and high-pressure air is used to move material through a pipeline (PSG, 2018). Pneumatic conveying systems are favoured over mechanical conveying systems in that they offer lower maintenance, enhanced safety and better sanitation (Buhler, 2013).

Traditionally, pneumatic conveying systems include an air source, an in-feed device, conveying line and receiver (Strebel, 2017). It is an absolute necessity that the conveying pipeline is fed at the correct rate (PSG, 2018). If the feed rate is too high, the pipeline could block, and low feed rate means the pipeline is under-utilized (Buhler, 2013). The feed into the pipeline is controlled by the rotary airlock valve. A rotary valve is a compact device that serves as a lock to prevent air loss while simultaneously controlling the continuous discharge of particulate solids under gravity (Gericke, 2015). The vanes of the valve rotate to form pockets between them, and material enters the pockets through the inlet port and then exit through the outlet port (Buhler, 2013). The primary duty of the rotary valve is to maintain pressure between the ports, which helps material to move continuously. For powder industries, rotary valves also perform an important function of acting as a flame barrier and process isolator in the event of an explosion (PSG, 2018).

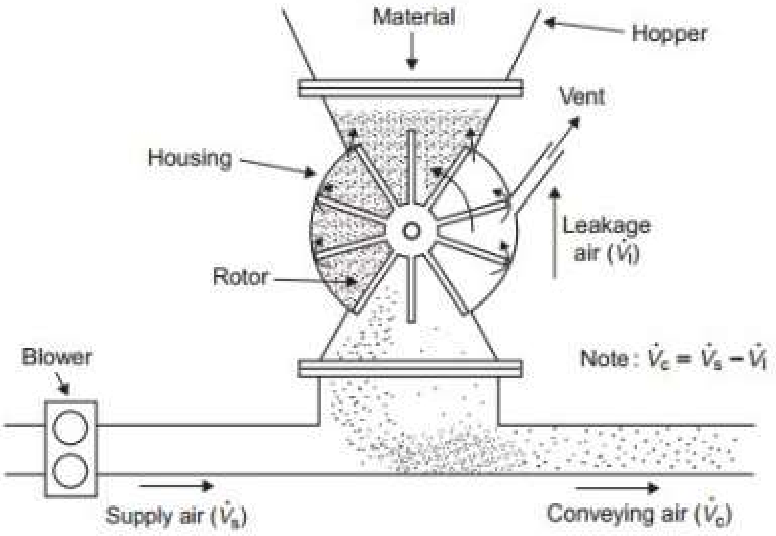


Figure 3.2: A typical pneumatic conveying system in which the rotary valve is used **(Aptech, 2018)**.

According to (Buhler, 2013), air pressure, velocity, temperature and quality are important considerations in a pneumatic conveying system. In milk powder processing plants, an air handling unit (AHU) is used to ensure that the temperature and quality of the air that is used for conveying are of the correct standard. The AHU takes air from outside, reconditions it by filtering, heating or cooling and then supplies it for the purpose of conveying powder. In the blow through AHU, the fan blows air through the filters and cooling or heating coils before discharging it, while in the draw through AHU; the fan pulls air through the filters and heating or cooling coils before discharging it to the outlet space (Gericke, 2015). For a milk powder process, dehumidification is the prime requirement. This is achieved in the cooling coil, where the air is cooled to reach the dew point temperature, and thus, condensation occurs to render the air with reduced moisture content (Hugo, 2017). The AHU is one of the energy intensive units in the plant, as it is continuously using steam even when production is halted.

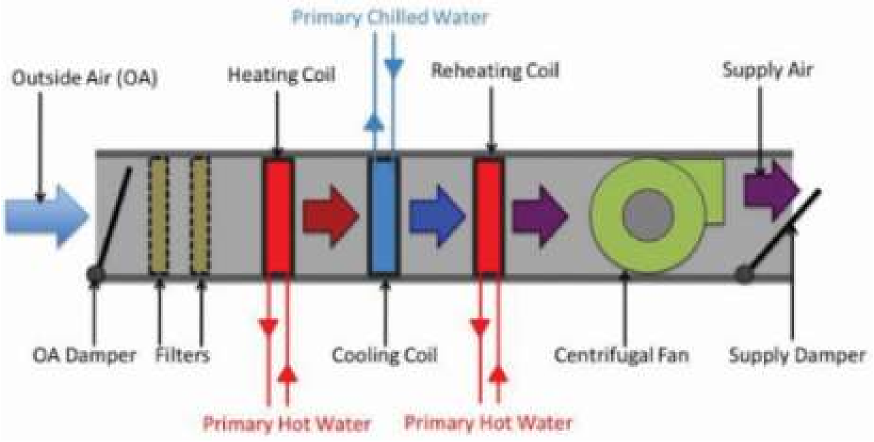


Figure 3.3: The flow of air through the AHU **(PSG, 2018)**.

3.3.3 Dry Mixing

When different powdered ingredients have been conveyed to the main storage hopper, it is necessary that they are homogeneously mixed to achieve a uniform blend. In a milk powder processing plant, dry powder ingredients are mixed by making use of a suitable dry powder stirrer. Dry mixing is less capital intensive and more energy efficient than wet mixing. Continuous mixing occurs as separate streams of ingredients are combined in a hopper, mixed, and the mixture removed continuously. The mixing process ensures that the ingredients are continuously blended until the nutrients are uniformly distributed throughout the batch. Dry mixing can be influenced by particle sizes, equipment configuration, mixing volume, mixing angle and rotational speed (Dali, 2016).

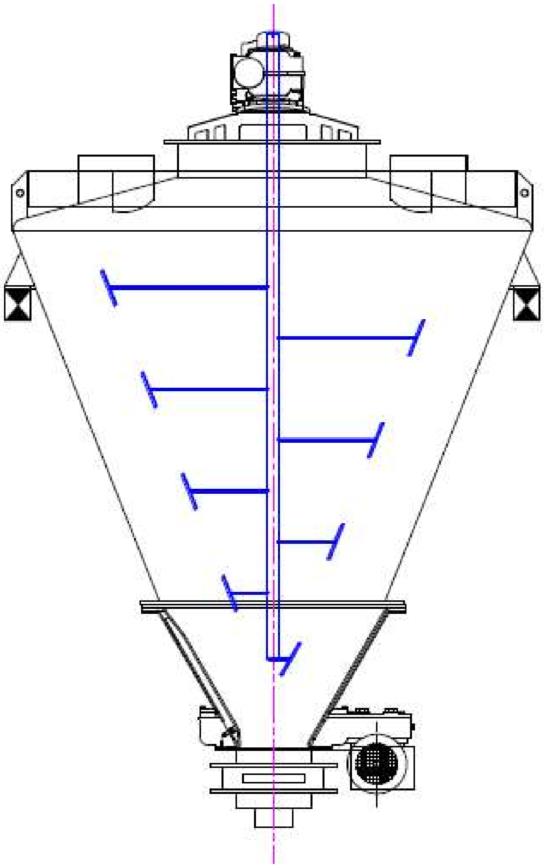


Figure 3.4: A dry mixing hopper with a stirrer.

In a milk powder processing application, of all the factors that affect the dry mixing process, none is as critical as the rate in which the different dry ingredients are fed into the dry mixing hopper (Coperion, 2017). To mitigate against the risk of inconsistent feeding, loss-in-weight feeders are used. Loss-in-weight feeders can be used for both small to high feed ranges and they work with extreme precision. In a loss-in-weight feeding system; the entire feeder, hopper and material are continuously weighed (Gericke, 2015). As the material is fed through the feeder, the weight of the system decreases, the feeder controls continuously check whether the material is flowing. The system electronically balances tare weight so the controller senses only the weight of the material in the supply hopper (Coperion, 2017).

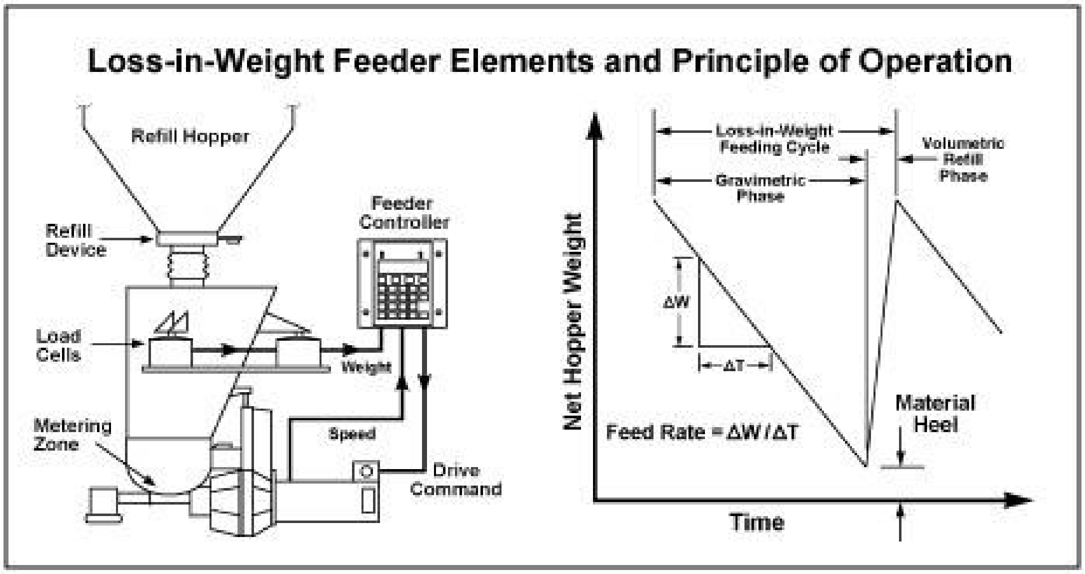


Figure 3.5: Elements of the loss-in-weight feeder **(Coperion, 2017)**.

3.3.4 Wet Mixing

In a dry mixing process, there is no heat treatment to destroy bacteria. If one or more ingredients in a dry-blended product are contaminated by even a low number of harmful bacteria, the bacteria will most likely be present in the final product. To prevent such harmful bacteria from being present in the final product, a combination of the heat treatment process and high shear mixing is adopted. In this process, water at ambient temperature is passed through a heat exchanger to increase its temperature to about 60 ^{0}C and then to a tank where mixing takes place. This is one of the energy-intensive processes in the milk powder spray-drying process, as steam is continuously used to achieve the desired outcomes. The tank is operated under vacuum, to create a pressure differential between the wet mixing tank and dry mixing hopper to allow for the powder to be drawn into the wet mixing tank.

In a wet mixing tank, a high shear mixer, comprised of the three-blade rotor turning at high speed within a stationary stator rotates as powder and hot water are continuously drawn into one end of the mixing head and expelled at high velocity (Banaszek, 2012). The purpose of the high shear wet mixing process is for emulsification, homogenization, particle size reduction, dispersion and standardization of milk (Charles, 2013). Fluids undergo shear when one area of the fluid travels with a different velocity relative to an adjacent area (Banaszek, 2012). The tip velocity of the fluid at the outer diameter of the rotor will be slightly higher than the velocity at the centre of the rotor. The resulting velocity difference creates hydraulic shear which promotes fast mixing by breaking down solid agglomerates (Charles, 2013).

Figure 3.6: A high shear mixer in the wet mixing tank.

3.4 Concentration

Transforming a liquid into a dry powder requires removal of practically all water. The second step in the milk powder spray-drying process is the process known as concentration. Concentration is the process of concentrating the milk of water-like viscosity to a concentrate (Vagn, 2010). The concentration of a liquid by the evaporation process is distinguished from drying in that the final product is still liquid. The concentration of a liquid by evaporation occurs under vacuum. Water is evaporated by means of indirect heating in the evaporator. The concentration process in the evaporator is determined by the product properties such as viscosity and heat stability. The process of concentration consists of three consecutive sub-steps: Pre-heating, Evaporation, and Homogenization.

3.4.1 Pre-heating

Pre-heating is the thermal treatment of the standardized milk before evaporation, in order to denature milk proteins to control the viscosity of the final product (Beutler & Espalieu, 2013). The process of pre-heating before evaporation is vital in managing the after-thickening tendency of milk (Vagn, 2010). According to (Shakeel, 2017), pre-heating avoids coagulation during evaporation, ensures microbiological safety in the finished product and also inactivates enzymes. The heat treatment before evaporation also affects the colour and taste development during the shelf life of milk (Beutler & Espalieu, 2013). Shakeel (2017) states that the pre-heating process needs to be closely monitored as the rate of deterioration of milk powders increases as the result of the Maillard reaction. The Maillard reaction occurs between the reducing sugars and amino acids when heated (Shakeel, 2017).

The pre-heating process is done in three steps to gradually increase the temperature. Pre-heating is usually the highest temperature step that is applied during the manufacture of milk powders. It is performed by indirect heating in the first two steps and direct injection of steam in the final stage. An increase of pasteurization temperature slows down natural increase of viscosity over time (Shakeel, 2017). Pasteurization at low temperature increases the after thickening during shelf life; while pasteurization at high temperature leads to a decrease of the after thickening (Beutler & Espalieu, 2013). Higher temperature pasteurization is preferred; however, issues of Maillard reaction add limitations to the temperature of pasteurization (Shakeel, 2017).

Standardized milk from the wet-mixing process with an outlet temperature of 65 ^{0}C is initially pre-heated in the first pasteuriser to 75 ^{0}C. The second stage involves indirect pre-heating the milk to about 87 ^{0}C. The level of pre-heating is the principal means of achieving the desired extent of denaturation of proteins in milk production so that the least heat damage takes place during evaporation and spray-drying stages (Beutler & Espalieu, 2013). The advantage of the indirect heating is that the product will not be mixed with the condensation of steam and neither will the product be diluted (Vagn, 2010). The disadvantage is that it takes a long time for the product to be heated if the temperature needed is high, resulting in a concentrate with high viscosity (Vagn, 2010).

The final step in the pre-heating of milk is the direct pasteurization to increase the temperature from 87 ^{0}C to 95 ^{0}C. According to (Beutler & Espalieu, 2013), direct pasteurization can be done either by direct injection of live steam using a tangential swirl heater, or direct infusion at a sufficient pressure. The method used in the process for this study is the direct infusion of steam. This method offers controlled and short residence time with no mechanical impact. The greatest advantage of direct pasteurization is that it takes a short time to reach the desired temperatures. However, steam needs to be of good quality for use in human consumption. The direct method is preferred over the indirect method as biofilms of thermophilic bacteria may develop with the indirect method of pre-heating (Vagn, 2010).

3.4.2 Evaporation

Evaporation is the step that concentrates the pre-heated standardized milk to increase the total solids by extraction of water. Achieving the correct total solids is one of the essential requirements to ensure that a shelf stable product is obtained. Multiple effect falling film evaporators are used for this process. These types of evaporators' offer a short holding time, the amount of product in the evaporator is reduced and the surface from which evaporation takes place is increased (Beutler & Espalieu, 2013). The liquid to be evaporated is evenly distributed to the inner surface of a tube, and it flows downwards forming a thin film from which evaporation will take place because of the heat applied by the steam (Vagn, 2010).

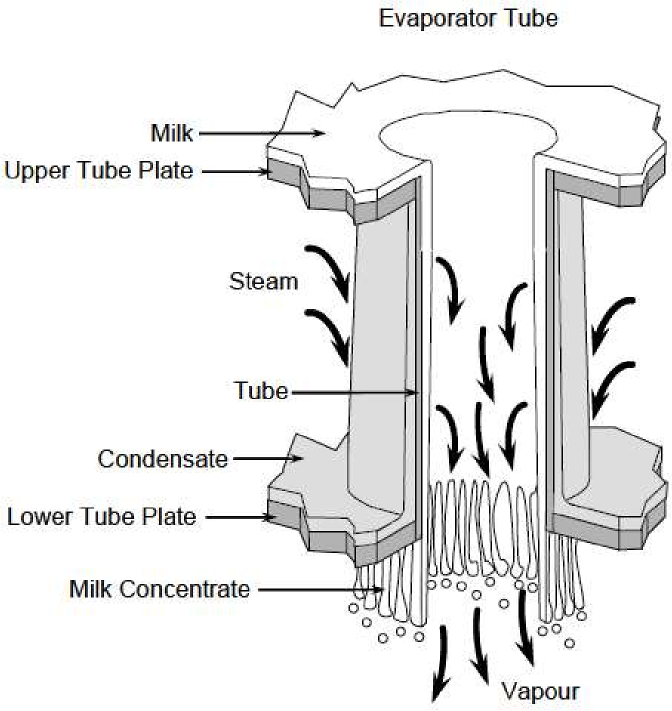


Figure 3.7: Evaporation mechanism in a falling film evaporator tube **(Vagn, 2010)**.

The steam condenses and flows downward on the outer surface of the tube. A number of tubes are built together side by side. At both the top and bottom ends, the tubes are fixed to tube plates and the entire bundle is enclosed by a jacket.

Figure 3.8: The evaporator calandria dome

The steam is introduced through the jacket. The space between the tubes forms a heating section. The inner side of the tubes is called the boiling section. The heating section and boiling section are collectively called the calandria. The concentrated liquid and the vapour leave the calandria at the bottom. The steam on the heating section condenses on the outer surface of the tubes and is collected as condensate at the bottom of the heating section part.

Figure 3.9: The bundle of tubes making up a calandria.

For any given feed of product, a part of the solvent is evaporated, resulting in a concentrated product. The evaporation ratio which is the measure for the evaporation intensity can be defined as the ratio between the amount of feed and concentrate (Beutler & Espalieu, 2013).

$$Evaporation\ Ratio = \frac{Feed}{Concentrate} \tag{3.1}$$

Since milk is a sensitive product due to protein, the boiling section is operated under vacuum to avoid denaturation of proteins (Vagn, 2010). The evaporation, therefore, takes place at a lower temperature than that corresponding to atmospheric pressure, which is necessary to maintain the product quality. In a multiple effect evaporator, the number of effects is determined according to economic criteria (Beutler & Espalieu, 2013).

There are two effects used in the evaporation process for this study. The principle of multiple effect evaporators is to save energy. As vapour from the first effect contains almost all the applied energy, it is used to evaporate water in the second effect. The boiling temperature for the second effect is lower than that of the first effect, thus working as a condenser for vapours from the first effect and the energy in the vapour is used as it condenses. In order to obtain a temperature difference in the second effect between the milk and vapour coming from the first effect, the boiling section of the second effect is operated at a higher vacuum corresponding to a lower boiling temperature (Beutler & Espalieu, 2013). Many effects can be added as possible, the limit is the lowest vacuum obtainable and that is dependent on the temperature of the cooling water condensing the vapour from the last effect whereby a vacuum is maintained (Vagn, 2010).

The design of evaporators has a significant impact on the energy consumption of the evaporator (Shakeel, 2017). Usually, the increased demand for multiple effect evaporators requiring a bigger heating surface area to obtain better consumption figures can be met by using more tubes. This results in less liquid getting into each tube, and the produced film becomes too thin (Vagn, 2010). At high solids contents, the viscosity increases and the film does not flow very well, thus resulting in the risk of burnt deposits. The design operating principle should be in line with the coverage coefficient:

$$Coverage\ coefficient = \frac{output\ at\ the\ lower\ end\ of\ the\ tubes}{Periphery\ of\ the\ tubes} \tag{3.2}$$

The trend has been to manufacture the calandria with longer tubes in order to obtain more heating surface and maintaining the coverage coefficient at the same level (Vagn, 2010). This approach has been adopted for the design of the evaporators that this study is based on. Furthermore, each calandria is split into two sections in both the first effect and the second effect. The product is pumped into the initial section of the first effect and then to the second section of the first effect. Having passed through the second section of the first effect, it is pumped into the second effect in the similar manner. This system is as cheap as the recirculation, but has the advantage of the divided calandria with no circulation necessary (Beutler & Espalieu, 2013).

The auxiliary equipment fitted onto the evaporators in relation to this study includes a thermo-compressor, separators, product distribution system, condenser and cooling towers. The use of a thermo-compressor is adopted to save energy by increasing the temperature and pressure level of the discharged vapour. The vapour from a low pressure is compressed to a high pressure by using steam of a higher pressure than that of the vapour. Thermo-compressors operate at very high steam flow velocities and have no moving parts, moreover, their construction is simple, the dimensions small, and the costs low (Vagn, 2010).

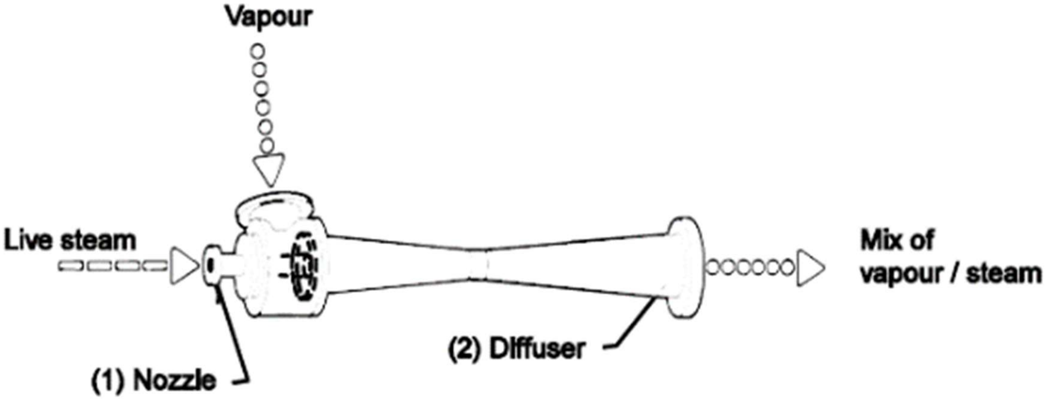


Figure 3.10: The thermo-compressor and its working principle **(Vagn, 2010)**.

In a live steam nozzle, the pressure of the inflowing steam is converted into a high velocity, and thus creates a jet which draws in part of the vapour from the separator of the evaporator. In the diffuser, a mixture of live steam and vapours is formed, and deceleration occurs which increases the pressure and temperature of the mixture. This mixture is used as heating steam for the evaporator. The best efficiency in the thermo-compressor is obtained when the temperature difference between the boiling section and the heating section is low (Love, 2014). A thermo-compressor which has been designed for a higher live steam pressure can draw a larger amount of vapour from the separator than one built for a lower pressure.

Since the vapours generated from evaporation in the first effect are used as heating media in the second effect, it is vital that product is separated. This is a vital requirement as it would contaminate the condensate and result in loss of good condensate that would otherwise be returned to the boilers. Product separation from vapours is achieved by using separators with a tangential inlet, connected to the calandria below the tubes. The majority of the concentrate is discharged from the bottom of the calandria, but some of the concentrates will be carried along with vapour as small droplets due to a high velocity of vapour. Special care is taken to design the separator to avoid product carry-over at lowest possible pressure drop, as a drop in the pressure is equal to drop in heating enthalpy in the following effect with an overall drop in the efficiency as a result (Vagn, 2010).

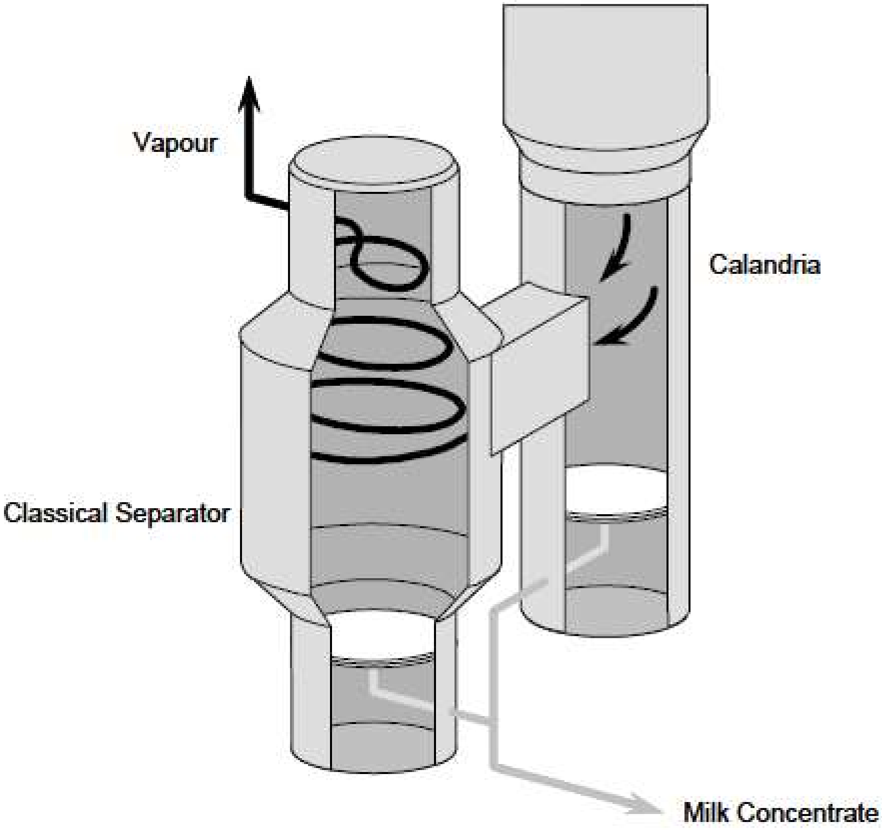


Figure 3.11: An arrangement of the separator with a tangential inlet and calandria **(GEA, 2016)**.

It is vital that the product is distributed evenly into all tubes in the calandria to ensure a good coverage ratio as required. This is achieved by a static distribution plate. The incoming product enters a distribution plate placed inside the cone (GEA, 2016). The cone is placed above the distributor bowl with a number of holes to maintain a certain level of product. The product flows through the holes in the plate by gravity. Each hole is placed just above the area between the tubes (Vagn, 2010). The product flows onto the tube plate and then over the edge down along the surface of each tube. The flash vapour also enters the tubes, and pushes the product against the inner surface of the tubes, giving it the initial velocity. This distribution system is much more flexible in respect of capacity, as an increase in the level in the distributor bowl as a result of increased capacity will make the product flow through the holes at a higher velocity, thus maintaining the level (Vagn, 2010).

Figure 3.12: The calandria distribution plate.

A surface condenser is installed and its working principle is similar to that of the tubular heat exchanger. The advantage of the surface condenser is that the cooling water and vapour condensate cannot mix. Surface condensers are preferred due to the cleaning process which involves the use of acids and chemicals. This ensures that acid vapours which could be evaporated remain separate from the cooling water. The cooling towers are used to cool the water used in the condenser. The flow of water goes from the cooling tower to the condenser from where it is pumped back to the cooling tower.

3.4.3 Homogenization

According to (Gósta & Ulrika, 2015), homogenization is the standard industrial process universally practised for stabilizing fat emulsion against gravity separation in milk processing. Homogenization primarily causes disruption of fat globules into much smaller ones and consequently diminishes creaming and tendency of coalescing (Kouroutsidis, 2016). All homogenized milk is produced by mechanical means as milk is forced through a small passage at high velocity (Kouroutsidis, 2016). The disintegration of the original fat globules is achieved by a combination of contributing factors such as turbulence and cavitation (Gósta & Ulrika, 2015). According to (Kouroutsidis, 2016), Fox and others studied fat content in homogenisation of milk and found that homogenization reduces fat globule size from an average of 3.5 µm in diameter to below 1 µm.

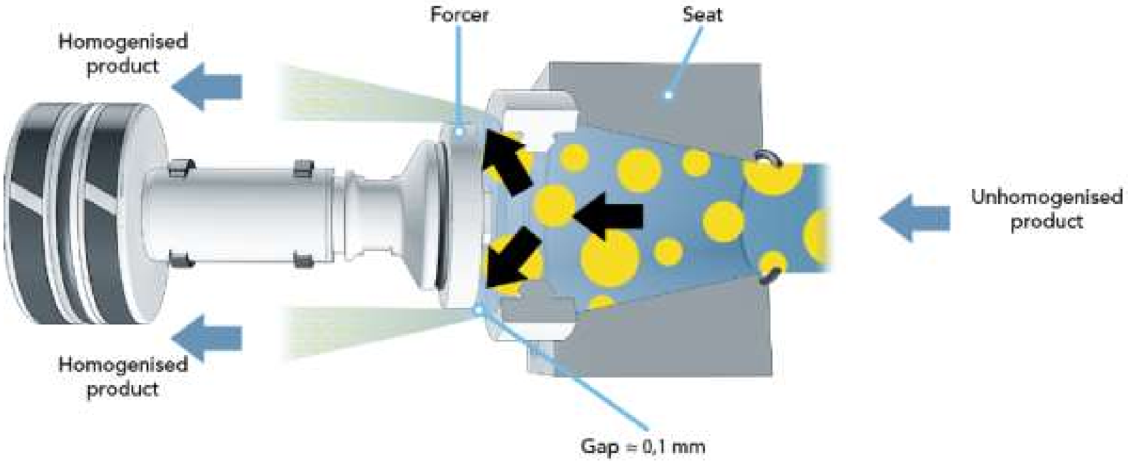


Figure 3.13: The working principle of homogenization **(Kouroutsidis, 2016)**.

Passing the liquid through the narrow gap causes its flow velocity to increase until the static pressure is so low that the liquid starts to boil. The maximum speed depends mainly on the inlet pressure. As the liquid exits the gap; the speed decreases and the pressure increases again; the liquid stops boiling and the steam bubbles implode. The physical state and concentration of the fat phase at the time of homogenization contribute materially to the size and dispersion of the ensuing fat globules (Kouroutsidis, 2016).

A homogenizer is essentially a piston pump with a homogenization device. The product enters the pump block and is pressurized by the piston pump. The pressure that is achieved is dependent on the back pressure obtained through the distance between the plunger and seat in the homogenization device. Since the piston pump is driven by an electric motor and it's a positive displacement pump, its capacity can only be adjusted by changing the speed of the motor or transmission from the motor to the gearbox. There is a two-stage homogenization device installed for the process that this study is based on. In a single stage homogenization device, the back pressure is created by the process and cannot be adjusted (Kouroutsidis, 2016). In the two-stage homogenization process, the back pressure is created by the second stage, and thus it can be chosen by the operator to achieve optimal homogenization efficiency (Kouroutsidis, 2016).

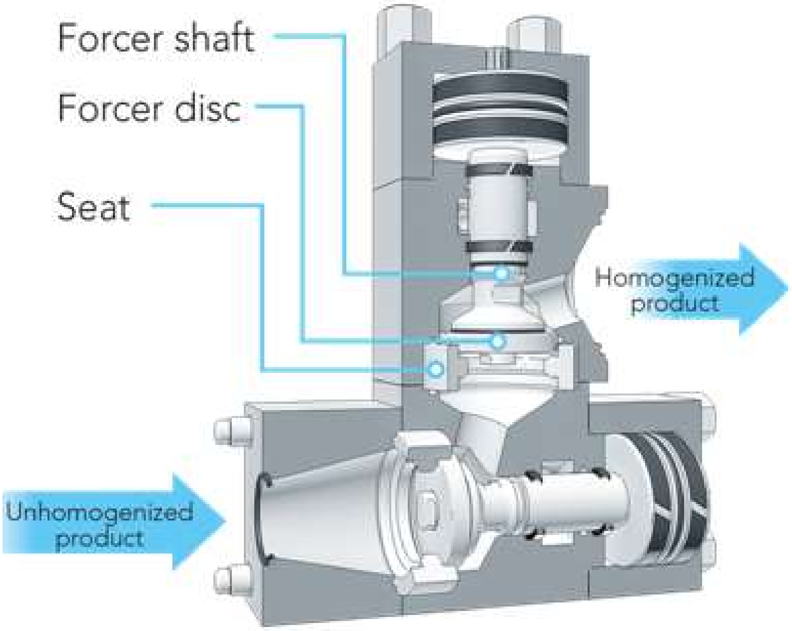


Figure 3.14: The two-stage homogenization device **(Kouroutsidis, 2016)**.

Concentrated milk from the evaporator enters the homogenizer at about 90 ^{0}C. Since the homogenization of cold milk is difficult as the fat is solidified, the temperature of concentrate is very important in homogenization. According to (Gósta & Ulrika, 2015), increasing the homogenization temperature decreases the viscosity of milk and improves the transport of membrane material to the fat globules. The advantages of homogenization include smaller fat globules leading to a lesser cream-line formation, more full-bodied flavour and white colour of milk (Gósta & Ulrika, 2015).

3.5 Spray-Drying

The final step in the milk powder processing is the process known as spray-drying. It is the transformation of concentrated and homogenized fluid feed into dried form by spraying the feed into a hot drying medium (Fu & Woo, 2012). Drying occurs by means of direct heating. Similar to the concentration process in the evaporator; the drying process is determined by the product properties such as viscosity and heat stability. Moreover, the flow of feed, as well as the homogeneity, has a major impact on the drying process. Spray-drying is a continuous process operation with a series of steps. A concentrated feed entering the spray dryer undergoes a series of continuous transformations before powder is discharged (Gohel & Parikh, 2009). The spray-drying process involves atomization, drying and separation.

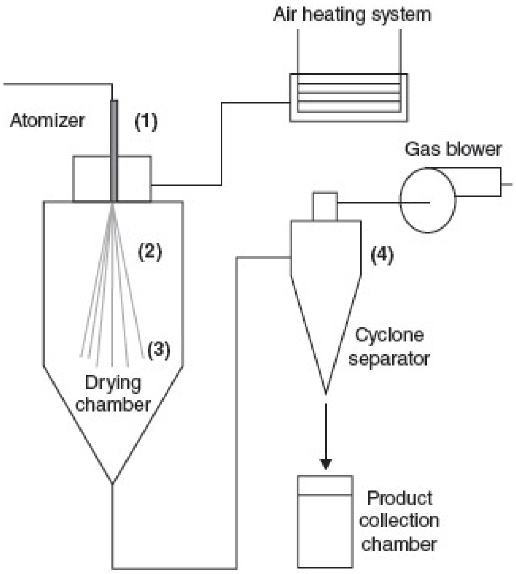


Figure 3.15: An overview of the drying chamber and related equipment **(Vagn, 2010)**.

In essence, the product is fed through an atomizer into the drying chamber. The drying air is supplied by the air heating system to the hot air dispenser, located at the inlet of the drying chamber. At the inlet of the drying chamber, hot air and atomized droplets of product meet and evaporation take place. Product falls to the bottom of the drying chamber and enters the conveying pipe. The particles with a small diameter remain entrained in the air and pass through a cyclone separator for separation. The separated fines leave the cyclones to enter the conveying system with the product. The air passes from the cyclone to the atmosphere via the gas blower.

3.5.1 Atomization

Atomization is defined as bringing fluid or solid substances into a state of minute division (Percy, 1987). The aim of atomizing the concentrate is to provide a very large surface, from which the evaporation can take place. The breaking up of bulk liquid into small droplets drives the rest of the spray drying process by reducing internal resistance to heat transfer (Handscomb, Kraft & Bayly, 2009). The smaller droplets, the bigger surface and thus a better thermal efficiency of the dryer is obtained. Atomization is the heart of spray-drying and is the first transformation process that the feed undergoes during spray-drying (Handscomb, et al., 2009).

After homogenization, the feed is pumped into the tank for intermediate storage. The temperature decreases during the intermediate storage. Atomization requires high feed temperatures and lower viscosity than that coming from the homogenizer. In order to achieve these objectives, a tubular heat exchanger is used to increase the temperature of the feed to about 70 ^{0}C and thus the viscosity decreases. Indirect heating is vital in this step to ensure that chances of contaminating product before drying are completely eliminated. The feed is then pumped by the high-pressure piston pump through the high-pressure pipe to the atomizing device which is located in the inlet of the drying chamber, where the hot air dispenser is also located.

Figure 3.16: The feed concentrate passing through an atomizer.

The ideal situation is to spray droplets of equal sizes which would result in the same drying time for all particles and equal moisture content of the final product. However, in practice, it is difficult to obtain completely homogeneous droplets from an atomizing device. Atomization devices influence the shape, velocity, structure, and size distribution of the droplets and in turn have an influence on the particle size and nature of the final product. The working principle of the atomizers is governed by the liquid disintegration phenomenon (Gohel & Parikh, 2009). According to (Fu & Woo, 2012), a liquid jet of constant radius initially, when falling vertically under gravity increases the length and reaches a critical value. At the critical value, the jet loses its cylindrical shape and decomposes into a stream of droplets that occur primarily due to a decrease in surface tension (Fu & Woo, 2012). This phenomenon is key to achieving the good results in the dryer.

Atomization is key to a successful utilization of the spray dryer and is directly responsible for many distinctive advantages offered by the spray-drying process (Percy, 1987). To comply with the requirements of high surface to mass ratio resulting in high evaporation rates in the drying chamber; and production of powder particles of desired shape, size and density; a pressure nozzle atomization device is used in the subject process. The basic function of the pressure nozzle is to convert the pressure energy supplied by the high-pressure piston pump through the pipe into kinetic energy in a form of a thin film (Beutler & Espalieu, 2013). The conversion of pressure energy to kinetic energy is facilitated by the discharge of liquid through the orifice. The feed emerging from the nozzle orifice as a high-speed film easily breaks into a spray of droplets.

The advantage of a pressure nozzle is that atomization can be duplicated. Pressure nozzles can be integrated into multiple arrangements to obtain an increased amount of flow rate and particle size flexibility (Vagn, 2010). There are five pressure nozzles used each with a design capacity in excess of 1500 kg/hr of concentrate liquid. This results in powder production rate of approximately 5000 kg/hr, depending on the concentration of feed. Pressure nozzles result in particles with less blocked air when compared to other types of atomizers (Fu & Woo, 2012). Other advantages of pressure nozzle atomizing devices include the production of powder with high bulk density, improved flow-ability, fewer deposits in the drying chamber, and ability to produce big particles (Vagn, 2010). The limitation of a high-pressure nozzle is that at high feed rates, the droplets become less homogenous and coarser; however, due to its number of advantages, it is practicable to use it over other types of atomizing devices.

3.5.2 Drying

The main objective of the drying process is to transform standardized concentrate from a liquid state to a powder state, as quickly as possible, and at a lowest possible temperature for minimal effects on functionalities of the product while maximizing energy efficiency (Beutler & Espalieu, 2013). The characteristic features of the dryer used in the subject matter process are that a tall stainless steel tower of 6.1 metres height with an opening at the top receives atomized feed from the atomizers and hot air from the hot air distribution chamber for the purposes of evaporating water and thus drying concentrated fluid.

Figure 3.17: The tower opening and air distribution chamber.

Figure 3.18: The 6.1-metre stainless steel tower.

At the tower opening, very fast air is fed at a speed of approximately 220 m/s at the temperature of about 350 ^{0}C. This primary air is blown straight downwards into the tower, with warm secondary air at approximately 55 ^{0}C around the building of the drying chamber. The air mass ratio between the primary air and the secondary air typically vary from 40% to 60 %; and this supports the fast drying compared to the closed top dryer (Beutler & Espalieu, 2013). Drying is established by the contact of the dry hot air with the concentrate droplets atomized to increase the surface area for heat and mass transfer. Spraying concentrate into a high turbulence zone between the primary and the secondary air greatly enhances the heat and mass transfer (Vagn, 2010). The drying mechanism is driven by the relative humidity and temperature of the surrounding air.

The quality of drying air is a very important feature in the spray-drying process. Since the drying air is in contact with the product, quality aspects of the air are a major importance. Air is filtered before being supplied to the tower to ensure cleaner operation. The highest degree of filtration is necessary to avoid foreign particles from finding their way into the product. Control and monitoring of relative humidity, air temperature, and correct condition of filtration are important. Filtered air passes through the indirect oil-fuelled air heater, where drying air and combustion gases have separate flow passages. Combustion gases pass through galvanized tubes that act as heat transfer surface for drying air. The combustion chamber is made from heat-resistant steel, with the end covers hinged to be easily open for cleaning of tubes. According to (Vagn, 2010), air heaters of this type range from 250 to 400 ^{0}C and have the efficiency in the regions of 85%.

The distribution of air is another vital aspect of a spray-drying process. The goal in the milk powder processing operation is to get the best mixture of incoming hot air and concentrate droplets in order to accelerate evaporation. This is achieved by a co-current air distribution arrangement. The air disperser should have the ability to guide the air and the atomized droplets in the right direction in order to avoid deposits in the drying chamber (Beutler & Espalieu, 2013). However, this is difficult to avoid, for this reason, softballs are used as knocking devices located outside the tower, around the circumference. These softballs are hinged on the ropes and move at different times, knocking the tower, thus creating a vibratory effect which removes deposits inside the drying chamber. It is important to ensure that the room where the drying chamber is located is over-pressurised and the temperature kept above the atmospheric temperature during and after production (Hugo, 2017).

3.5.3 Separation

After completion of drying, the moist air and course dried particles are separated at the cyclones. The drying air contains about 30% of small portions of powder when it leaves the chamber (Fu & Woo, 2012). It is necessary to clean the drying air by separating the powder particles to prevent pollution. Separation is never completely effective; however, it reduces the content of powder in air from 30% to about 5% depending on the type and stages of separation involved before discharging to the atmosphere. Cyclone separators are used in the process under investigation. The particles separated from the air are re-injected into continuous flow into the tower and participate in the agglomeration of the powder. The air is discharged into the atmosphere with a small amount of powder particles in comparison to before the separation process.

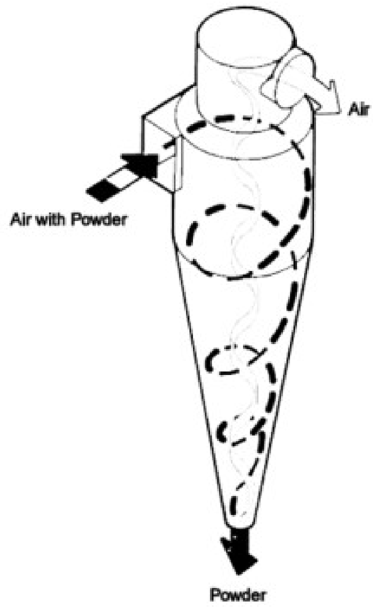


Figure 3.19: The working principle of the cyclone separator **(Vagn, 2010)**.

The operation of cyclones is based on a vortex motion. According to (Vagn, 2010), the centrifugal force acting on each particle causes the particle to move away from the cyclone axis towards the inner cyclone wall. The movement in the radial direction is the result of two opposing forces where the centrifugal force acts to move the particle to the wall; while the drag force of the air acts to carry the particles into the axis. As the centrifugal force dominates, separation takes place. Powder and air pass tangentially into the cyclone at equal velocities. They swirl in a spiral form down the base of the cyclone and thus separating the powder out to the cyclone wall. A softball knocking device, as explained previously, is used to cause the powder to fall down from the separator. The air spirals upwards along the centre axis of the cyclone and passes out at the top.

Application of centrifugal force equations has proved that high particles mass yield better separation effectiveness. Moreover, the shorter way the particle has to travel the better efficiency; and the closer the particle is to the wall the better efficiency, because the velocity is highest and the radial distance is short (Fu & Woo, 2012). However, time is required for the particles to travel to the cyclone wall, so a sufficient air residence time should be taken into consideration when designing a cyclone (Fu & Woo, 2012). The cyclone has some obvious advantages, such as high efficiency, if it is constructed properly, it is easily maintained as there are no moving parts, and, furthermore, it is easy to clean, if the construction is with a fully welded centre cyclone (Gunter, 2017).

The particles residence times in a spray dryer are short. In order to improve the drying economy, the drying is divided into two steps. The first step is done in a spray drying chamber as discussed above. The evaporation of moisture from a particle becomes more difficult in the core and requires more time. The second drying step is done in a fluid bed. The fluid bed drying technology is suitable as the residence time in the fluid bed is so long that the moisture from the core of the particle can be evaporated. In a fluid bed, the drying air is introduced to the powder through a special perforated plate at the bottom. The fluid bed used is the vibrating type, and thus transports powder for packing purposes while at the same time dries the powder.

3.6 Conclusion

This chapter detailed in-depth operational characteristics of the milk powder spray-drying process. From this chapter, it can be concluded that researchers have focused on each stage of the spray-drying process to maximize efficiency. Operational characteristics and process conditions revealed that there are opportunities for energy efficiency improvements in the process. The heat sinks and sources throughout the processes steps involved in the process were discussed.

The next chapter is the energy evaluation study in the process in order to ascertain the amount of energy that is lost.

CHAPTER FOUR – ENERGY LOSSES IN DRYING

4.1 Introduction

This chapter evaluates the energy losses associated with the powder drying stage. The scope of this chapter includes energy losses that could be recovered in the combustion of fuel for heating drying air, reticulation of drying air, drying chamber and fluid bed. Evaluating the amount of energy losses in the drying stage is necessary to determine the significance of the losses that can be recovered. A mathematical method of analysing heat energy was used to determine the energy losses in the drying system. Literature related to the subject under evaluation is briefly discussed where necessary.

4.2 Air Heater Losses

In the combustion chamber, heavy fuel oil (HFO) also known as marine fuel oil (MFO) combines with atmospheric air to burn and generate flue gases that are necessary for use as a heating medium in increasing the temperature of drying air in the air heater. The flue gases used as heating medium in the air heater exit the stack to the atmosphere at high temperatures. This section determines the amount of energy lost in the combustion of fuel for increasing the temperature of drying air. In order to evaluate the energy losses in the combustion system, a mass and energy balance is employed. Mass and energy balance is essential in solving for compositions and flow rates of process streams, identifying chemical properties and integrating materials to solve for energy inputs and outputs (Henda, 2015).

Referring to Table A.1 in appendices; the chemical composition of HFO are Carbon, Hydrogen and Sulphur. It is Oxygen that is responsible for the combustibility of these elements, and it is assumed that the elements burn completely.

The molecular balance of Carbon and Oxygen reaction is given by:

$$C + O_2 = CO_2$$

$$12 + 32 = 44$$

For a single molecule of Carbon:

$$1 + 2.667 = 3.667$$

From the equation above, it can be concluded that a single molecule of Carbon requires 2.667 molecules of Oxygen to burn and produces 3.667 molecules of Carbon dioxide.

Similarly, the molecular balance of Hydrogen and Oxygen reaction is given by:

$$2H_2 + O_2 = 2H_2O$$

$$4 + 32 = 36$$

For a single molecule of Hydrogen:

$$1 + 8 = 9$$

From the equation above, it can be deduced that a single molecule of Hydrogen requires 8 molecules of Oxygen to burn and produces 9 molecules of water.

Finally, the molecular balance of Sulphur and Oxygen reaction is given by:

$$S + O_2 = SO_2$$

$$32 + 32 = 64$$

For a single molecule of Sulphur:

$$1 + 1 = 2$$

From the equation above, it can be deduced that a single molecule of Sulphur requires a single molecule of Oxygen to burn and produces 2 molecules of Sulphur dioxide.

The chemical composition of HFO is given as 85.1% Carbon, 10.9% Hydrogen and 4% Sulphur. The amount of Oxygen required must be sufficient to burn the molecules of the elements presented in the chemical composition of HFO. Working on a conventional principle that air contains 23% Oxygen and 77% Nitrogen by mass; for the complete combustion of 1kg of HFO, the sufficient amount of Oxygen required for combustion can be ascertained by calculating the mass of theoretical air required.

The mass of theoretical air is given by:

$$M_{at} = \frac{100}{23} \times \{(2.667 \times C) + (8 \times H_2) + S\}$$

$$= \frac{100}{23} \times \{(2.667 \times 0.851) + (8 \times 0.109) + 0.04\}$$

$$= 13.832 \ kg/kg\ fuel$$

The stack emissions tests results as depicted in Table B.2 of appendices indicate that on different days of tests, the excess air in the flue gases varies. This is mainly due to different conditions of operation from which the tests are conducted. The average excess air percentage of fuel can be calculated: $Average\ Excess\ Air\ \% = 21\%$

Since the emissions tests show that an average of 21% of theoretical air is used as excess air for combustion per mass of fuel. The total mass of air initially drawn in for combustion can be calculated on the basis of 1 kg of fuel.

$$M_{tot} = M_{at} + M_{exc}$$

$$M_{tot} = M_{at} + 21\% \times M_{at}$$

$$M_{tot} = 13.832 + (0.21 \times 13.832)$$

$$M_{tot} = 16.737 \ kg/kg\ fuel$$

The Excess Air is given by:

$$M_{exc} = 13.832 \ \times 21\%$$

$$= 2.905 \ kg/kg\ fuel$$

The products of combustion which result in flue gases are given by:

CO_2	⇨	3.667 x 0.851	= 3.1206
H_2O	⇨	9 x 0.109	= 0.981
SO_2	⇨	2 x 0.04	= 0.08
Excess O_2	⇨	2.905 x 0.23	= 0.6682
Excess N_2	⇨	16.737 x 0.77	= 12.8875
Total Flues			$= 17.737\ kg/kg\ fuel$

Table 4.1: The chemical composition of flue gases to the atmosphere.

From the results above, it can be seen that for 1 kg of oil; 16.737 kg of air is required for complete combustion and the emitted flue gases in the combustion chamber would equate to 17.737 kg. The ratio of oil mass to air is 1:16.737 and that of oil mass to flues is 1:17.737. The diagram below depicts the material movement in the combustion chamber and air heat exchanger.

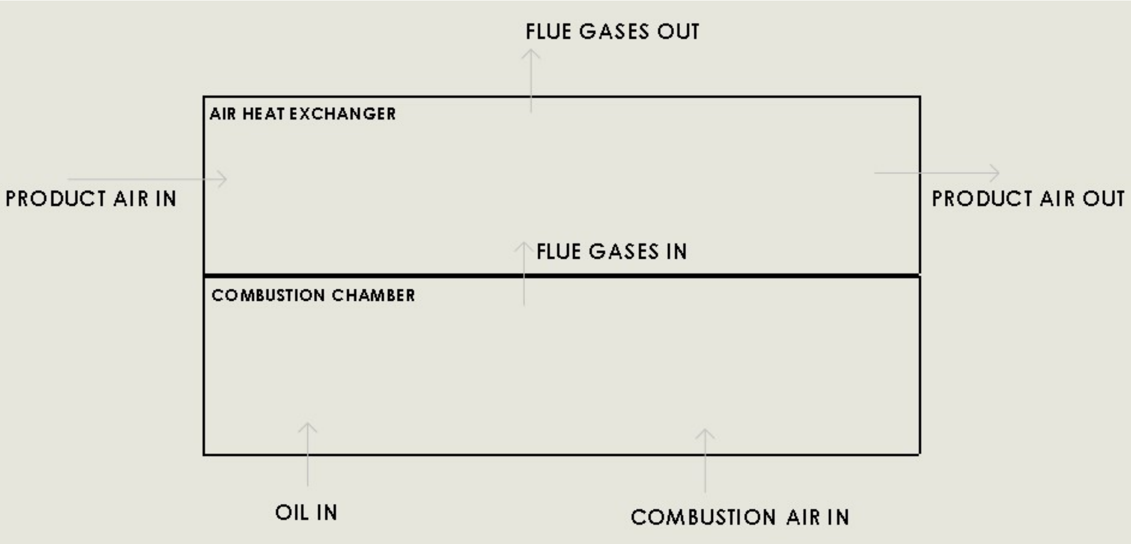


Figure 4.20: The material movement in the combustion chamber and heat exchanger.

The overall average efficiency for the system is deduced from the stack emissions tests results in Table B.2 on appendices. The stack emissions tests results and the daily records of air heater performance in Table B.3 of appendices give the following average values:

- The flow rate of oil in = 270 l/hr
- The temperature of the product air out = 355 ^{0}C
- The temperature of the flue gases out = 265 ^{0}C

The mass flow rate of oil is calculated based on the given specific gravity of 0.97 for the HFO oil as shown in Table A.1 on appendices:

$$\dot{m}_{oil} = 270 \times 0.97$$

$$= 261.9\ kg/hr$$

From the oil mass to air and flues ratio:

$$\dot{m}_{ca} = 4383.420\ kg/hr$$

$$\dot{m}_{fl} = 4645.320\ kg/hr$$

A system for ascertaining temperatures and relative humidity was developed by (Shakeel, 2017) for Harrismith area and depicts that the average temperature of the atmospheric air in the area is 14.4 ^{0}C. The graph depicted in Table C.4 on appendices shows that the calculation was developed from different temperatures throughout the day and night, and the average of the 24 hours results recorded. For the basis of these calculations, the average atmospheric temperature shall be used as 15 ^{0}C. The air at this temperature is by convention assumed to have the specific heat capacity of 1.006 kJ/kgK (Botes, 2007).

From the overall efficiency of 83% for the system as shown in Table A.1 of appendices; the mass flow of product air for drying purposes is calculated based on the fuel properties given on the same table in appendices:

$$\eta_o = \frac{Energy\ Out}{Energy\ In}$$

$$\eta_o = \frac{\dot{m}_{pa} \times c_{p_{pa}} \times (T_{pa\ out} - T_{pa\ in})}{\dot{m}_{oil} \times CV_{oil}}$$

$$\dot{m}_{pa} = \frac{0.83 \times 261.9 \times 39.57 \times 10^3}{1.006 \times (355 - 15)}$$

$$= 25\,147.96\ kg/hr$$

This is the amount of product air that is passed through the drying chamber for removal of moisture from the concentrated product on an hourly basis.

Coskan, Oktay & Ilten (2009) developed the method of determining the specific heat capacity of flues gases at various temperatures and fuel chemical composition. According to (Coskun, et al., 2009) the specific heat capacities of the flue gas differ depending on the chemical composition of fuels, excess air ratio and temperature of flues. The model developed estimates accurately the specific heat capacity by entering the chemical composition of the fuel, excess air and flue gas temperature. The approach can be applied to all carbon-based fuels and is suitable for practical estimation of flue gas specific heat provided that the fuel chemical composition is given (Coskun, et al., 2009).

In general applications, heat exchanger energy balance calculations depend on the enthalpy values of the heat source passing through them, usually, flue gases. The enthalpy of flue gases is the function of the specific heat capacity and temperature (Coskun, et al., 2009). Different fuels used for industrial applications were used in the experiments to determine the specific heat of flue gases and the results were presented in the graph.

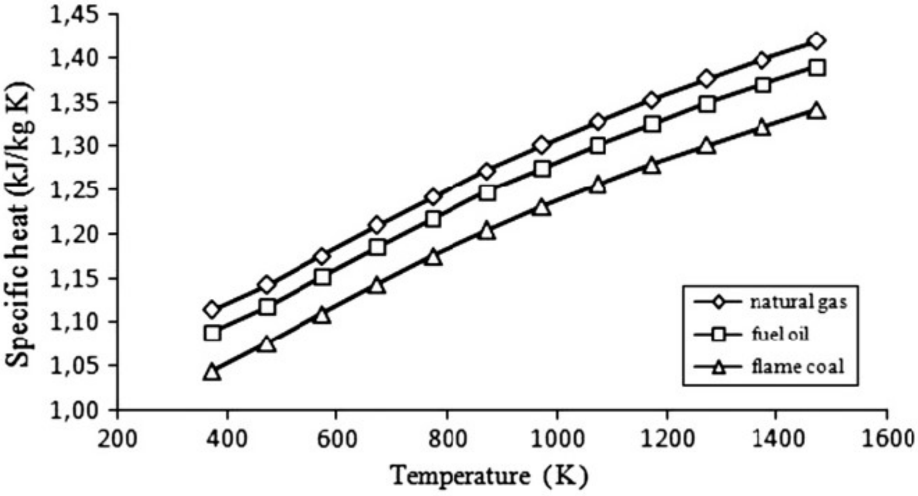


Figure 4.21: The relationships of flue gas temperatures to specific heat capacity for different fuels **(Coskun, et al., 2009)**.

The specific heat capacity of flue gas for each different composition has a similar trend but different values. Coskun, et al., (2009) affirm that chemical composition of fuel, excess air amount and gas temperature directly affect the flue gas specific heat and these parameters have effects on combustion and enthalpy values which greatly affect cost optimization. The flue gas temperature at the outlet of the air heat exchanger is 265 ^{0}C and this value correlates to the specific heat capacity of approximately 1.14 kJ/kgK for fuel oil. While different grades of fuel oil are used by different industries, the approximation of the specific heat capacity at any given temperature would not differ vastly for different grades of fuel oil.

The energy leaving the heat exchanger as flue gases to the atmosphere can now be calculated from equation (2.1) as:

$$Q_{fl\,out} = \dot{m}_{fl} \times c_{p_{fl}} \times \left(T_{fl\,out} - T_{atm}\right)$$

$$= 4645.320 \times 1.14 \times (265 - 15)$$

$$= 1\,323.92 \; MJ/hr$$

The above is the amount of energy that exits the air heater stack every hour during production. A greater amount of this energy can be recovered and re-used for pre-heating of incoming air to the air heater to improve the efficiency and cost factor on the air heater.

4.3 Reticulation Losses

In determining the energy losses that occur during the distribution of hot air from the air heater to the drying chamber, a suitable energy transfer equation needs to be applied. The heat flow passing through a pipe with insulation can be calculated if the thermal resistance of each material forming the insulation layer is known. For cylinders, the thermal resistance varies with the thickness of insulation, thermal conductivity, pipe diameter and temperature of fluid passing through the pipe.

4.3.1 Heat Losses Equation

From the basics of Ohm's Law for electricity, the voltage drives the current of a certain magnitude. The amount of current that flows for a given voltage is inversely proportional to the electrical resistance. This is presented by the equation below, better known as Ohm's Law.

$$V = I \times R_{elec} \tag{4.1}$$

In thermodynamics, the temperature difference drives the heat flow of a certain magnitude. The amount of heat flow for a given temperature difference is inversely proportional to the thermal resistance.

$$\Delta T = Q \times R_{th} \tag{4.2}$$

Figure 4.22: The relationship between electrical and thermal conduction.

In electrical conductors, the resistance depends on the material properties, and, similarly in thermal conductors, the material properties determine the resistance to heat flow. Re-arranging equation (4.2) above, concludes that the thermal resistance equation for thermodynamics can be written as:

$$R_{th} = \frac{(T_2 - T_1)}{Q} \tag{4.3}$$

Fourier's Law of heat conduction affirms that the effectiveness of heat transfer through the material is dependent on the thermal conductivity. From equation (2.5):

$$Q_c = -k \times A \times \frac{\Delta T}{\Delta r} \tag{4.4}$$

Re-arranging:

$$Q_c \times \Delta r = -k \times A \times \Delta T \tag{4.5}$$

By convention, the area of the cylinder is given by: $2\pi r L$

Therefore equation (4.5) can be re-written as:

$$Q_c \times \Delta r = -k \times 2\pi r L \times \Delta T \tag{4.6}$$

Re-arranging:

$$\frac{Q_c \times \Delta r}{r} = -2\pi kL \times \ \Delta T \tag{4.7}$$

Integrating:

$$Q_c \int_{r_1}^{r_2} \frac{\Delta r}{r} = -2\pi kL \ \int_{T_1}^{T_2} \Delta T \tag{4.8}$$

$$Q_c \times \ln\left(\frac{r_2}{r_1}\right) = 2\pi kL \times (T_2 - T_1) \tag{4.9}$$

Solving equation (4.9):

$$\frac{(T_2 - T_1)}{Q_c} = \frac{\ln\left(\frac{r_2}{r_1}\right)}{2\pi kL} \tag{4.10}$$

Substituting equation (4.3) into equation (4.10); we get:

$$R_{th_c} = \sum \frac{\ln\left(\frac{r_2}{r_1}\right)}{2\pi kL} \tag{4.11}$$

The conductive thermal resistance equation for cylindrical pipes is given by equation (4.11) above. The equations derived above are only for thermal resistance for heat flow through conduction. In order to incorporate the thermal resistance for heat flow through convection, the Newton Law of Cooling equation can be re-arranged.

Re-arranging equation (2.6):

$$\frac{1}{2\pi rLh} = \frac{(T_2 - T_1)}{Q_{conv}} \tag{4.12}$$

Substituting equation (4.12) into equation (4.3); we get:

$$R_{th_{conv}} = \sum \frac{1}{2\pi rLh} \tag{4.13}$$

The convective thermal resistance equation for cylindrical pipes is given by equation (4.13) above. Since heat losses through the pipe are a combination of both conduction and convection, the thermal resistances of both cases can be added to determine the total inner heat losses through a pipe.

$$Q_{L_{tot}} = \frac{(T_2 - T_1)}{\sum R_{th_c} + \sum R_{th_{conv}}} \tag{4.14}$$

Substituting thermal resistances:

$$Q_{L_{tot}} = \frac{(T_2 - T_1)}{\sum \frac{\ln\left(\frac{r_2}{r_1}\right)}{2\pi kL} + \sum \frac{1}{2\pi rLh}} \tag{4.15}$$

Simplifying and re-arranging for total losses:

$$Q_{L_{tot}} = \frac{2 \times \pi \times L \times (T_2 - T_1)}{\sum \frac{1}{k} ln\left(\frac{r_2}{r_1}\right) + \sum \frac{1}{r_1 h}} \tag{4.16}$$

From equation (4.16) above, the subscripts 1 and 2 in the denominator denote the internal and external dimensions of a bare pipe; and the internal and external dimensions of the insulation layer of the pipe referencing from the centre of the pipe.

4.3.2 Heat Transfer Coefficient of Hot Air

The dimensions and material data of the pipe used for the distribution of hot air from the air heater to the drying chamber are depicted in Figure D.3 on appendices.

The 70 m long pipe has the inner bare pipe radius of 0.4 m; outer bare pipe radius of 0.404 m and the insulated pipe outer radius is 0.554 m. It is used to convey product drying air which leaves the air heater at a temperature of 355 ^{0}C and enters the drying chamber at the temperature of 340 ^{0}C. According to (CSS, 2015); at these temperatures, the thermal conductivity of 3Cr12 stainless steel is approximately 36 W/mK.

From Table E.8 on appendices, the thermal conductivity of insulation material was determined by interpolation as 0.0905 W/mK.

To determine the heat transfer coefficient of hot air conveyed through the pipe, the properties of the fluid at film temperature need to be determined. The film temperature is given by the average of the sum of air temperature at the inlet of pipe and air temperature at the exit from the pipe.

$$T_f = \frac{T_{in} + T_{out}}{2}$$

$$= \left(\frac{355 + 340}{2}\right) + 273$$

$$= 620.5\ K$$

At the above film temperature, the properties of hot air as shown on Table C.5 are as follows:

- $\rho = 0.5697\ kg/m^3$
- $c_p = 1.0559\ kJ/kgK$
- $k = 0.04781\ W/mK$
- $\mu_f = 3.083\ \times 10^{-5}\ kg/ms$
- $\mu_1 = 3.1072\ \times 10^{-5}\ kg/ms$

There are three important dimensionless parameters that are essential in determining the heat transfer coefficient of fluids namely: the Prandtl number, Reynolds number and Nusselt number. The Prandtl number is the approximation of the ratio of diffusivity of momentum to thermal diffusivity, and physically relates the relative thickness of the hydrodynamic layer and thermal boundary layer (Kunes, 2012). The Reynolds number represents the ratio of the inertia to the viscous forces and the Nusselt number relates data for the convective heat transfer coefficient to the thermal conductivity of the fluid under the same conditions (Kunes, 2012). For a mass flow rate of 25 147.96 kg/hr of hot air calculated above; the velocity of air in the pipe can be calculated:

$$\dot{m}_{air} = \rho \times V \times A$$

$$V = \frac{25\ 147.96 \times 4}{\pi \times (0.8)^2 \times 0.5697} \times \frac{1}{3600}$$

$$V = 24.39\ m/s$$

The Prandtl number is given by:

$$\mathrm{Pr} = \frac{c_p \times \mu_f}{k}$$

$$= \frac{1.0559 \times 10^3 \ \times 3.083 \times 10^{-5}}{0.04781}$$

$$= 0.681$$

The flow of hot air is that of forced convection through the cylindrical pipe as it is assisted by centrifugal fans to travel along the pipe. The applicable formula for Reynold's number must be used to determine whether the flow is laminar or turbulent.

$$Re = \frac{D \ \times \ V \ \times \ \rho}{\mu_f}$$

$$= \frac{0.8 \ \times 24.39 \ \times 0.5697}{3.083 \times 10^{-5}}$$

$$= 360\ 557$$

Since Reynold's number is above 6000; the flow is fully turbulent.

Kunes (2012) suggests the below equation for the Nusselt number of fully turbulent flows that are of forced convection through a pipe.

$$Nu = 0.023 \times Re^{0.8} \times Pr^{0.33} \times \left(\frac{\mu_f}{\mu_1}\right)^{0.14}$$

$$= 0.023 \times (360557)^{0.8} \times 0.681^{0.33} \times \left(\frac{3.083}{3.1072}\right)^{0.14}$$

$$= 564.64$$

Finally, the heat transfer coefficient of hot air inside the ducting can be determined as follows:

$$Nu = \frac{h_1 \ \times D}{k}$$

$$h_1 = \frac{Nu \ \times k}{D}$$

$$= \frac{564.64 \times 0.04781}{0.8}$$

$$= 33.74\ W/m^2K$$

4.3.3 Heat Transfer Coefficient of Atmospheric Air

The outer convective heat transfer is the function of the heat transfer coefficient of air outside the ducting. The heat transfer coefficient of atmospheric air is dependent on the relative velocity between the object surface and the air. The heat transfer coefficient of atmospheric air differs depending on the velocity of air, but there is not much contribution due to temperature. Air velocities between 1 m/s and 5 m/s are usually classified as free convective flows in heat transfer problems (Yunus & Afshin, 2017). According to (Yunus & Afshin, 2017), the practicably convective heat transfer coefficient of air at these speeds is between 5 and 7 W/m^2K. This approach can be applied to all atmospheric air velocities whose speeds are within an acceptable range for practical estimation of free convective heat transfer coefficient of air.

According to (Meteoblue, 2017), the average speed of the wind for the Harrismith area is in the region of 12.5 km/hr throughout the year, which is approximately 3.5 m/s. For the purpose of these calculations, the heat transfer coefficient of atmospheric air outside the ducting will be used as 6 W/m^2K.

4.3.4 Heat Losses Calculations

The total energy losses through the reticulation system can be determined from equation (2.4) as follows:

$$Q_L = \frac{2 \times \pi \times l \times (t_s - t_e)}{\left(\frac{1}{u_1 r_1}\right) + \left(\frac{1}{u_2 r_3}\right) + \frac{1}{k_1}\ln\left(\frac{r_2}{r_1}\right) + \frac{1}{k_2}\ln\left(\frac{r_3}{r_2}\right)}$$

$$= \frac{2 \times \pi \times 70 \times (355 - 15)}{\left(\frac{1}{33.74 \times 0.4}\right) + \left(\frac{1}{6 \times 0.554}\right) + \frac{1}{36}\ln\left(\frac{0.404}{0.4}\right) + \frac{1}{0.0905}\ln\left(\frac{0.554}{0.404}\right)}$$

$$= \frac{149\ 539.81}{74.096 \times 10^{-3} + 300.84 \times 10^{-3} + 276.398 \times 10^{-6} + 3.48895}$$

$$= 38\ 699.15\ J/s$$

$$= 139.32\ MJ/hr$$

The above is for energy lost through conduction and convection as a result of conveying hot air over a long distance. Other losses that occur due to radiation were not considered for the purpose of this study as they do not contribute any significant amount of recoverable energy. The surface temperature of insulation was measured and found to be ranging between 25 ^{0}C and 22 ^{0}C. Preliminary calculations showed that the radiation losses are insignificant in this temperature range and thus were omitted in this study.

The energy losses through reticulation amount to 1.34% of the total energy used on hourly basis.

4.4 Drying Chamber Losses

In order to ascertain the energy that is lost through the exhaust of the drying chamber, a material and energy balance around the system boundary is essential. In order to perform a total water balance in the dryer, psychrometric chart and table of water properties of liquid and saturated vapour at varying temperatures are used to determine the amount of moisture contained in the incoming air used for drying purposes. The psychrometric chart is the graphical presentation of air properties under various conditions (Shelton, 2008). Understanding the chart allows one to use it as an aid in the design and management of drying systems, ventilation systems and dehumidification systems. The versatility of the psychrometric chart lies in the fact that by knowing only two properties of air, the rest of the properties can be determined.

4.4.1 Moisture Content in Air

Atmospheric air contains nitrogen, oxygen, carbon dioxide, water vapour, other gases, and miscellaneous contaminants such as dust (Shelton, 2008). Dry air exists when all contaminants and water vapour have been removed from the atmospheric air, and moist or humid air is a mixture of dry air and water vapour. Air temperature is vital in the designing of drying systems. Air temperature is a measure of the heat content of the air. In psychrometric charts, it can be represented as a dry bulb, wet bulb or dew point temperature (MAC, 2015). Dry bulb temperature is the air temperature determined by an ordinary thermometer, wet bulb temperature reflects the cooling effect of evaporating water; and dew point temperature is the temperature below which moisture will condense out of the air (MAC, 2015).

Humidity is another vital parameter in the design or management of drying systems. Relative humidity (RH) is a measure of how much moisture or water vapour is present compared to how much moisture the air is capable of containing at that temperature (Shelton, 2008). It can be expressed mathematically as:

$$\%RH = \frac{P_{wv}}{P_{wv_s}} \tag{4.18}$$

There are three main processes that play a significant role in the treating of drying air (Hugo, 2017). A psychrometric process that involves the increase or decrease in the temperature of air without changing its humidity ratio is known as sensible heating or cooling. Humidity ratio is the actual mass of water vapour present in humid air to the mass of the dry air. When the process involves the increase of dry bulb temperature and the humidity ratio of the air simultaneously, it is known as heating and humidifying. A psychrometric process that involves the removal of water from the air as the air temperature falls below the dew point temperature is known as cooling and dehumidification.

The kinetic theory of gases states that the different molecules in a mixture of gases are so far apart that they act independently and do not react with each other. Dalton's law of partial pressures expands on this concept by stating that the total pressure of a mixture of gases is equal to the sum of the partial pressures of the component gases (Schmidt, 2013). Dalton's law of partial pressures is used to determine the individual pressures of each gas in a mixture of gases. The pressure exerted by an individual gas in a mixture is known as partial pressure. Assuming that humid air has the mixture of water vapour and air as ideal gases, we can use the ideal gas law to solve problems involving gases in a humid mixture.

Idea gas law of water vapour:

$$P_{wv}V = n_{wv}R_0T \tag{4.19}$$

The number of moles is given by:

$$n = \frac{m}{M} \tag{4.20}$$

Substituting equation (4.20) into (4.19):

$$P_{wv}V = \frac{m_{wv}}{M_{wv}} R_0 T \tag{4.21}$$

Re-arranging:

$$\frac{P_{wv}M_{wv}}{R_0 T} = \frac{m_{wv}}{V} \tag{4.22}$$

Equation (4.22) can be written in terms of density as:

$$\rho_{wv} = \frac{M_{wv}P_{wv}}{R_0 T} \tag{4.23}$$

Similarly, for the Idea Gas Law of dry air, the density will be given by:

$$\rho_{da} = \frac{M_{da}P_{da}}{R_0 T} \tag{4.24}$$

The pressure of moist air is the sum of the partial pressures of dry air and the water vapour.

$$P_{ma} = P_{da} + P_{wv} \tag{4.25}$$

The absolute humidity of air is defined as the ratio of the mass of water vapour to the mass of dry air. Denoting that the volume occupation of dry air and water vapour is the same, the absolute humidity can then be expressed in terms of density or mass as:

$$Absolute\ humidity\ (\omega) = \frac{\rho_{wv}}{\rho_{da}} = \frac{m_{wv}}{m_{da}} \tag{4.26}$$

Substituting equation (4.23) and equation (4.24) into equation (4.26) gives:

$$\omega = \frac{M_{wv}P_{wv}}{M_{da}P_{da}} \tag{4.27}$$

The absolute humidity equation can be written as:

$$\omega = \frac{M_{wv}P_{wv}}{M_{da}(P_{ma} - P_{wv})} \tag{4.28}$$

The molar mass of dry air and water vapour or water depends on the composition components in the air or water. In calculations, the standard conventional approximations used for the molar mass of air are 29 g/mol and that of water is 18 g/mol (MAC, 2015). The absolute humidity equation can be finally written as:

$$\omega = \frac{18}{29} \times \frac{P_{wv}}{(P_{ma} - P_{wv})} \tag{4.29}$$

The pressure of humid air is that of the atmosphere used as 101 kPa, and the pressure of water vapour can be obtained from calculation using equation (4.18) if the saturation pressure at that particular temperature is known. The moisture content is a dimensionless number, but usually, based on equation (4.27) it can be expressed as: kg_{H_2O}/kg_{air}.

4.4.2 Enthalpy of Moist Air

Another important parameter in the design and management of drying systems is the specific enthalpy of air that is used in the system for drying purposes.

For a moist air, the enthalpy is the sum of the enthalpies of the dry air and the water vapour.

$$H_{ma} = H_{da} + H_{wv} \tag{4.30}$$

In humidification or dehumidification systems, the flow of dry air tends to remain constant as the moist air passes through the humidifier or dehumidifier. For this reason, for convenience, the specific enthalpy of moist air is defined per unit mass of dry air.

$$h_{ma} = \frac{H_{ma}}{m_{da}} \tag{4.31}$$

Equation (4.30) can be rewritten in terms of specific enthalpy as:

$$h_{ma} = h_{da} + \frac{m_{wv}}{m_{da}} h_{wv} \tag{4.32}$$

Finally, equation (4.32) can be written as:

$$h_{ma} = h_{da} + w h_{wv} \tag{4.33}$$

According to (Shelton, 2008), for ideal gases, the specific enthalpy is a function of temperature only, provided that the gases are at constant and similar pressure. The specific enthalpy of moist air can, therefore, be written by defining the specific heat capacity of the mixture as below (Shelton, 2008):

$$c_{p_{ma}} = c_{p_{da}} + \omega c_{p_{wv}} \tag{4.34}$$

For the purpose of these calculations, the specific heat capacity of dry air is used as 1.006 kJ/kgK and that of water vapour is 1.88 kJ/kgK (MAC, 2015).

4.4.3 Energy Losses in the Secondary Dryer

Considering the secondary dryer as depicted in Figure G.5 on appendices:

Overall Mass Balance:

$$m_{SDP} + m_{AD1} + m_{AD2} + m_{AC} = m_{EXH2} + m_{FP}$$

$$m_{SDP} = 17520 + 4670 - 3016 - 8568 - 4863$$

$$= 5\,743\ kg/hr$$

The above is the mass flow of semi-dry milk powder from the primary dryer that has been partially dried as it has not passed through the secondary or final dryer.

The secondary dryer receives the product at a rate of 5 743 kg/hr with a moisture content of 7.1%. The final product leaves the secondary dryer at the rate of 4 670 kg/hr with a moisture content of 2.8% as per the data depicted in Table F.9 on appendices. Some percentage of milk powder leaves the dryer through the exhaust as losses.

Considering the after-dryer first air (AD 1) at 75 ^{0}C, the saturation vapour pressure at this temperature can be seen from the psychrometric chart or tables of steam properties and is equivalent to: 38.58 kPa. AD1 air is at 20% RH, and thus the partial pressure of water vapour can be calculated using equation (4.18):

$$\%RH = \frac{P_{wv}}{P_{wv_s}}$$

$$20\% = \frac{P_{wv}}{38.58}$$

$$P_{wv_{AD1}} = 7.716\ kPa$$

Using equation (4.30) to calculate the absolute humidity for AD 1:

$$\omega_{AD1} = \frac{18}{29} \times \frac{P_{wv}}{(P_{ma} - P_{wv})}$$

$$= \frac{18}{29} \times \frac{7.716}{(101 - 7.716)}$$

$$= 0.05134 \ kg_{H_2O}/kg_{air}$$

The specific heat capacity of AD 1 air can now be calculated from equation (4.34) since the absolute humidity is now known.

$$c_{p_{ma}} = c_{p_{da}} + \omega c_{p_{wv}}$$

$$= 1.006 + 0.05134 \times 1.88$$

$$= 1.1025 \ kJ/kgK$$

Applying the similar method in calculating the absolute humidity and specific heat capacity of after-dryer second air (AD 2) air and after-cooler (AC) air will give:

$$\omega_{AD2} = 0.02 \ \ kg_{H_2O}/kg_{air}$$

$$c_{p_{AD2}} = 1.0436 \ kJ/kgK$$

$$\omega_{AC} = 0.00154 \ \ kg_{H_2O}/kg_{air}$$

$$c_{p_{AC}} = 1.0088 \, kJ/kgK$$

To calculate the water content in the air exiting the secondary dryer to the atmosphere, a water balance in the secondary dryer is applied.

The amount of water vapour leaving the secondary chamber is calculated as:

$$Water\ Out = m_{EXH2}\omega_{EXH2} + m_{FP}\omega_{FP}$$

$$= 17520 \times w_{EXH2} + 4670 \times 2.8\%$$

$$= 17520 \times w_{EXH2} + 130.76$$

The amount of water vapour entering the secondary chamber is calculated as:

$$Water\ Vapour\ In = m_{SDP}\omega_{SDP} + m_{AD1}\omega_{AD1} + m_{AD2}\omega_{AD2} + m_{AC}\omega_{AC}$$

$$= 5743 \times 7.1\% + 3016 \times 0.05134 + 8568 \times 0.02 + 4863 \times 0.00154$$

$$= 741.44\ kg/hr$$

Finally, the water vapour balance:

$$Water\ Vapour\ In = Water\ VapourOut$$

$$741.44 = 17520 \times \omega_{EXH2} + 130.76$$

$$\omega_{EXH} = 0.0349 \quad kg_{H_2O}/kg_{air}$$

The principle of powder separation from moisture leaving the secondary dryer is that all powder particles must be retained in the system. However, this is never the case as some small particles of powder leave with air to the atmosphere, and some escape the drying chamber to the floor during operation. The exhaust leaving the secondary dryer consists of air, water vapour and some powder particles. Powder particles contribute to atmospheric pollution but have an insignificant amount of energy contained in them relative to the amount of energy contained in the moist air.

The specific heat capacity of the secondary exhaust is given by:

$$c_{p_{EXH2}} = c_{p_{da}} + \omega_{EXH2}c_{p_{wv}}$$

$$= 1.006 + (0.0349 \times 1.88)$$

$$= 1.0716\ kJ/kgK$$

The total energy leaving the secondary exhaust is calculated from equation (2.1):

$$Q_{EXH2} = \dot{m}_{EXH2} \times c_{p_{EXH2}} \times (T_{EXH2} - T_{atm})$$

$$= 17520 \times 1.0716 \times (65 - 15)$$

$$= 938.72\ MJ/hr$$

4.4.4 Energy Losses in the Primary Dryer

The product drying air that enters the drying chamber at 340 ^{0}C is initially drawn from the atmosphere at 15 ^{0}C with a relative humidity of 20%. It is heated up in the air heating system described above to the required temperature of 340 ^{0}C without addition or removal of moisture in the air. The humidity ratio of the initially drawn air remains the accurate measure of the water content in the air regardless of the change in temperature of the air, provided that the temperature of the air does not fall below the dew point temperature. In this scenario, the process is termed sensible heating as it increases the temperature without changing the humidity ratio of the air. Sensible heating increases the temperature of the air and thus improves the water holding capacity of the air.

The amount of water that can be contained by air at higher temperatures is much higher in comparisons to the amount of water that can be held by air at lower temperatures. It is for this reason that the air entering the dryer needs to be at the highest temperature possible to ensure that drying takes place effectively. From calculations based on 20% RH and 15 ^{0}C of atmospheric air, it can be concluded that the air entering the dryer has the humidity ratio of 0.0021 kg_{H_2O}/kg_{air}.

In the primary dryer, as depicted in Figure G.5 on appendices, the mass of exhaust air leaving the primary dryer can be calculated from the mass balance of the materials entering and leaving the dryer.

Overall Mass Balance:

$$m_{CP} + m_{PA} + m_{DP} = m_{EXH1} + m_{SDP}$$

$$m_{EXH1} = 8070 + 25147.96 + 1118.6 - 5742$$

$$= 28\,593.56 \ kg/hr$$

The moisture content in the exhaust air of the primary dryer can be obtained from the water balance of all moisture that enters and leaves the primary dryer.

The amount of water vapour entering the primary dryer is calculated as:

$$Water\ Vapour\ In = m_{CP}\omega_{CP} + m_{DP}\omega_{DP} + m_{PA}\omega_{PA}$$

$$= 8070 \times 41\% + 1118.6 \times 2.4\% \ + 25147.96 \times 0.0021$$

$$= 3388.357 \ kg/hr$$

The amount of water vapour leaving the primary dryer is calculated as:

$$Water\ Vapour\ Out = m_{EXH1}\omega_{EXH1} + m_{SDP}\omega_{SDP}$$

$$= 28\,539.56 \times \omega_{EXH2} + 5743 \times 7.1\%$$

Finally, the water vapour balance in the primary dryer:

$$Water\ Vapour\ In = Water\ Vapour\ Out$$

$$3388.357 = \ 28\,539.56 \times \omega_{EXH2} + 5743 \times 7.1\%$$

$$\omega_{EXH1} = 0.10072 \quad kg_{H_2O}/kg_{air}$$

The specific heat capacity of the primary exhaust is given by:

$$c_{p_{EXH1}} = \ c_{p_{da}} + \omega_{EXH2} c_{p_{wv}}$$

$$= 1.006 + (0.10072 \times 1.88)$$

$$= 1.1954 \ kJ/kgK$$

The total energy leaving as the primary exhaust is calculated from equation (2.1):

$$Q_{EXH1} = \dot{m}_{EXH1} \times c_{p_{EXH1}} \times (T_{EXH1} - T_{atm})$$

$$= 28593.56 \times 1.1954 \times (98.4 - 15)$$

$$= 2\,850.67 \ MJ/hr$$

4.5 Conclusions

The evaluation of energy losses associated with the drying stage of powder processing was carried out using a mathematical evaluation method. Energy is one of the highest factors driving production cost; and, the idea is to ensure that no energy is lost but rather all energy is used in manufacturing the product. Energy losses analysis enables identification of high quality and quantity of energy that can be used as a heating medium if recovered; and thus reduces cost and maximizes efficiency.

The total energy losses equate to the 2 850.67 MJ/hr in the primary dryer, plus the 938.72 MJ/hr lost in the secondary dryer. On hourly basis, the total energy lost is 3 789.39 MJ/hr. This amount correlates to the energy losses of 36.6% of the total energy input into the drying system. Reduction of energy losses throughout the factory will result in reduced fuel costs and improved process efficiency.

The next chapter evaluates the overall steam consumption in the process in order to assess the efficiency of steam utilization in the factory.

CHAPTER FIVE – EVALUATION OF STEAM CONSUMPTION

5.1 Introduction

This chapter evaluates the overall steam consumption of the milk powder spray-drying process. An overview of the method used to investigate the overall steam consumption is outlined in this chapter. The significance of the analysis used to determine steam consumption is highlighted. The scope of this chapter includes all heat sinks that make use of steam energy as a heat source throughout the process for operation. Evaluating the total steam energy consumption for the process is essential in identifying areas in which steam energy losses can be reduced, system design shortfalls and deficiencies, and alternative methods of heat transfer that can be explored as a substitute for steam energy usage.

5.2 Methods of Estimating Steam Consumption

The steam system's optimum operation largely depends on whether the steam consumption rate has been accurately established and the system optimally designed and erected. A properly designed steam system will consist of correctly sized pipes and ancillaries such as valves, steam traps, and others carefully selected and installed to give the best possible results desired. In any heating process, the warm-up component decreases as the product temperature rises, and the differential temperature across the heating coil reduces. However, the heat loss component increases as the product and vessel temperatures rise (Coetzee, 2016). The total energy demand at any time is the sum of the heat gaining and heat losing components. The heating element, when sized on the sum of the mean values of both these components, should normally be able to satisfy the overall heat demand of the application (Coetzee, 2016).

There are three main methods used for determination of steam consumption in a system: calculation, thermal rating and measurement (Coetzee, 2016). The calculation method makes it possible to estimate the steam consumption by analysing the heat output of an item using heat transfer equations. Although heat transfer requires one to take into account the losses occurring in the system which present unknown variables, the results acquired by this method are usually accurate in most cases. The thermal rating is provided by manufacturers to express the probable heat output in kW from which the steam consumption in kg/h is dependent on the recommended steam pressure. Measurement of steam consumption can either be done directly by using a flowmeter or measuring condensate.

5.3 Measurement of Steam Consumption

Steam consumption in non-flow type applications where the heat sink is of a fixed mass and a single batch within the vessel require the process heat-up time to be determined for the process. Flow type applications where a heated fluid constantly flows over the heat transfer surface require the flow-rate of the heat sink to be known. Steam consumption can be determined directly by use of a steam flowmeter. Direct measurement by use of the flowmeter is only useful in an operational plant. It is used to monitor the results of energy saving initiatives, indicate an increase in steam usage in the item of plant, and also to compare the efficiencies of one plant to another.

Steam consumption can also be estimated by measuring the mass of condensate over a period of time in an indirect heat exchanger. The method is considered to be as accurate as the calculation method and can be used in both non-flow and flow type applications (Coetzee, 2016). In a direct steam injection application, this method cannot be used as there is no condensate to be collected. Usually, when the flow rate of the heat sink is known, the process is simulated and the output mixture of both the heat source and heat sink is collected and measured to establish the flow rate of steam in the plant. It is important that the test is conducted with the condensate discharging into an atmospheric system and 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. It is always advisable to run three tests under similar conditions and average the results in order to get a reliable answer (Coetzee, 2016).

The following indicates the steps undertaken in measuring steam consumed by each plant item in the subject matter process.

Step 1:

A scale was first checked if within the calibration and verification date; and it was zeroed before the commencement of weighing. This ensures that the errors that can be introduced by the scale are eliminated in the weighing process.

Figure 5.23: A scale verification for the purpose of weighing condensate.

Step 2:

The weight of the empty container was weighed and recorded.

Figure 5.24: Weight of the empty container.

Step 3:

It is important that the condensate is allowed to flow for a few seconds without taking measurements to ensure consistency of flow.

Figure 5.25: The condensate collection step.

Step 4:

In order to ensure consistency of results, where a steam control valve is provided; the position of the valve was also recorded, and all readings taken at the same position or as close as possible to the initial reading position.

Figure 5.26: The position of the steam control valve.

Step 5:

Condensate was collected at equal times and more than three readings were collected to ensure consistency of results.

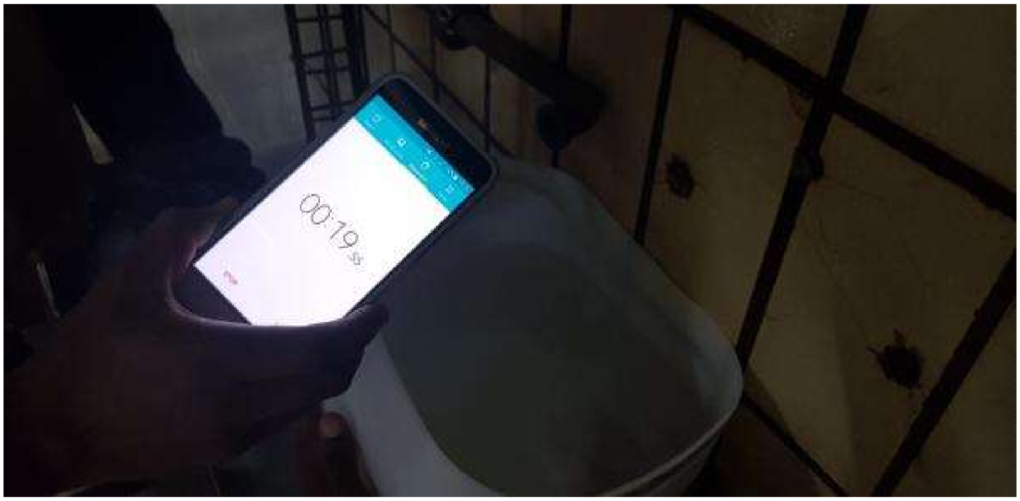


Figure 5.27: The collection of condensate and time readings.

Step 6:

The collected condensate was weighed and recorded, subtracting the weight of the empty container as per the second step. All collected condensate in each plant item was weighed and the average reading was taken.

Figure 5.28: The weight of the collected condensate.

Furthermore, the temperatures of the steam inlet to plant items and pressure were recorded. For the measuring of steam temperatures, an infrared thermometer was used to take the readings. Infrared thermometers are very good due to their high accuracy but present a challenge in measuring shiny or reflective objects. Materials do not emit the same amount of infrared energy even when at the same temperature. In order to take reliable readings, a non-reflective tape was wrapped around the pipe from which measurements were taken and it was evident that the readings are more accurate and close to those depicted in the steam table at specific pressures.

Figure 5.29: Readings of temperature measurements.

The table below depicts the measured amount of steam consumed by the process. The readings were taken at a particular time of production and may vary depending on the factors involved on any particular day of production. The heat source flow-rate was determined by the condensate measurement method, and the heat sink flow-rate was determined by the direct reading of the water flowmeter to equipment. Adding the hourly flow-rate of steam for the process gives a value of $steam\ consumption = 7\ 507\ kg/h$

	STEAM CONSUMPTION								
	Heat Source (Steam)					Heat Sink (Various)			
#	**Equipment/Line**	**Guage Pressure (Bar)**	**Measured Temperature (deg. Cels)**	**Control Valve Status (%)**	**Flowrate (kg/h)**	**Type**	**Measured Inlet Temperature (deg. Cels)**	**Measured Outlet Temperature (deg. Cels)**	**Flowrate (kg/h)**
1	Hot Water Supply	4.2	139.2	45	2440	Water	20	84.3	17325
2	Circulation for Oil Pipes	4.2	139.7	10	141	Water	37.3	59.6	3079
3	Wet Mixing Heater	4.2	144.1	35.6	589	Water	20	66.8	6457
4	Hydration Heater	4.2	136.3	36.5	476	Water	41	73	7226
5	Rework Heater	4.2	141	53.6	477	Water	36	65	8820
6	Adria AHU	4.2	124.9	100	780	Air	8	28	Unknown
7	Thermo Compressor	4.5	142.1	36.74	526	Milk	78	55	11500
8	Condenser	4.5	139	54.36	532	Water	28	58	9535
9	Evaporator Preheater	4.5	144.6	57	446	Water	35	85	4049
10	Pasteurizer	4.5	138.9	27.4	432	Water	35	82	4480
11	Secondary Air 2	4.5	126.5	60.5	668	Air	15	68	Unknown

Table 5.2: Plant items steam consumption in the process.

5.5 Conclusion

The direct method of steam consumption was carried out to investigate the amount of steam consumed by plant equipment during the milk powder spray-drying processing. Steam is usually the most expensive form of energy in process plants and the method of generating it contributes a large portion to environmental pollution by most process plants. Steam consumption analysis per plant equipment enables focused improvements in each item of equipment to ensure the steam system is working at its optimum efficiency and the cost of generating steam is minimized. Moreover, consumption analysis per plant equipment allows for alternative forms of energy to be considered for each equipment. Reduction of steam consumption in the factory will result in reduced fuel costs, reduced chemicals costs, reduced emissions surcharges, reduced maintenance cost and improved process efficiency.

The next chapter compares the different technologies available for heat transfer; in order to ascertain the most effective technology to be employed in the recovery of lost heat for the subject matter process.

CHAPTER SIX – EVALUATION OF HEAT TRANSFER TECHNOLOGIES

6.1 Introduction

This chapter compares the various available technologies used in heat recovery systems with the intent for providing a method for selecting a particular technology. From a wide range of technologies available for waste heat recovery, the three categories that will be discussed include heat exchangers, heat pumps and power generation. The selection method between the categories will be outlined through the review of the literature.

6.2 Heat Transfer Technologies

6.2.1 Heat Exchangers

There is a wide range of technologies available for low-grade waste heat recovery in process plants. The selection of the most suitable technology is dependent on the process plant, cost, technical viability and other factors. In most industries, heat transfer occurs in heat exchangers. Mikkonen (2013) states that counter flow heat exchangers are ideal for heating. In counter flow heat exchangers, heat source and heat sink, flow in opposite directions and thus can achieve a higher efficiency. Multiple-pass heat exchangers are preferred as they achieve more effectiveness (Kan & Jiajun, 2015). Due to the low specific heat of air, the use of air for heating purposes is limited (Rotor Industry, 2011). Larger heat loads require large volumes of air resulting in large ducting and fans, leading to a higher energy required for transportation (Chen, 2010). Due to a high moisture content of exhaust air, condensation may take place if the dew point is reached (Jansen, et al., 1981).

In gas-to-gas heat transfer, a variety of heat exchangers are designed to facilitate waste heat recovery (Law, 2014). The gas-to-gas heat exchanger is often referred to as an air handling unit (AHU). In an AHU heat is exchanged between the source and the sink across a series of heat conductive metal plates in a counter-flow configuration (Span, 2014). According to (Aneke, 2012), the unit is usually configured for collection of condensation in the recovery of latent heat from humid air sources. This heat exchanger has been proven to provide economical and environmentally beneficial solutions (Law, 2014).

The shell-and-tube heat exchanger is by far the most commonly used heat exchanger unit across process industries globally (Incropera & De Witt, 2014). Shell-and-tube plate heat exchangers are capable of handling high pressures and temperatures (Laval, 2016). Industrial applications of temperatures above 260 ^{0}C and pressures above 50 MPa usually make use of this technology (BEE, 2014). Shell-and-tube heat exchangers can facilitate heat transfer between two gaseous or two liquid streams (Laval, 2016). According to (Smith, 2005), fouling can be tolerated on the tube side owing to the ease of accessing the tubes for cleaning.

The plate-and-shell heat exchanger may be considered a mixture of the plate heat exchanger and the shell-and-tube heat exchanger (Incropera & De Witt, 2014). This type of technology has an outer shell enclosing a pair of welded circular plates (Law, 2014). The principle of the design is to combine a high heat transfer coefficient of the plate heat exchanger with the rigid design of a shell-and-tube heat exchanger (Law, 2014). The cooling medium generally flows on the shell side between the plate pairs, while the heat source flows between the welded plate pairs (Incropera & De Witt, 2014). It generally withstands corrosive fluids and can tolerate high operating temperatures and pressures (Span, 2014).

A welded plate heat exchanger consists of the series of thin plates joined together by welds (Allen & Backtrom, 2012). The hot stream and cold stream of gases flow in a counter-flow arrangement. Welded plate heat exchangers are not commonly used in gas-to-gas heat exchange application (Miller, et al., 2015). Since the unit is fully welded, it cannot be open for cleaning and thus does not tolerate fouling (Allen & Backtrom, 2012). Similar to the shell-and-tube heat exchanger, this type of unit can tolerate very large operating pressures and temperatures (Laval, 2016). When the heat source is exceedingly high and fouling is not a concern, welded plate heat exchangers are preferred over shell-and-tube heat exchangers.

A heat pipe heat exchanger is comprised of a number of heat pipes separated by a splitter plate (Walsh, 2011). The source and sink flow on opposite sides of the splitter plate and heat transfer is facilitated by the heat pipes (Ahmad & Shafie, 2011). The heat pipes have a very high effective thermal conductivity achieved by the simultaneous evaporation and condensation of the working fluid (Anderson, 2014). The advantage of this type of heat exchanger in gas-to-gas applications is that the heat pipes structure helps in overcoming low convective heat transfer coefficients associated with gases as heat transfer fluids (Anderson, 2014). Heat pipes allow for high rates of heat transfer and very low approach temperatures (Ahmad & Shafie, 2011).

The printed circuit heat exchanger is a highly compact unit comprised of a number of plates containing chemically engraved flow channels linked by diffusion bonding (Incropera & De Witt, 2014). The efficiency of this heat transfer technology is in the ranges of 95% (Aneke, 2012). According to (Incropera & De Witt, 2014), the operating temperatures for this unit can be around 900 ^{0}C and pressures in excess of 60 MPa. The printed circuit heat exchanger can be used for evaporative or condensing duties (Aneke, 2012). The manufacturing procedure consists of the complex multiple stages and renders the unit extremely costly (Incropera & De Witt, 2014). Process industries usually go for cheaper options, and thus the use of this unit is not common.

A rotary regenerator, also known as a heat-wheel is a widely used heat transfer technology in waste heat recovery (Singh, 2014). In a heat-wheel, the air in two adjacent ducts flows through a rotating matrix spanning the ducts which achieve heat transfer (Aneke, 2012). Span (2014) says that the rotary regenerator is often chosen as an alternative to the gas-to-gas plate heat exchanger due to the high efficiency of the unit in the ranges of 95%. Gas entrainment between the ducts during rotation of the matrix often results in an inherent problem of cross-contamination in this type of technology (Singh, 2014). A heat-wheel can tolerate light fouling due to the inclusion of the purge section (Span, 2014). It is reported that variations of rotary regenerators are offered by manufacturers to cater for the needs of the heat recovery process.

A run-around-coil heat exchanger is also used for heat exchange purposes where water or brine solution is the heat transfer fluid (Brucker, 2015). It is comprised of two separate coiled heat exchangers which are connected by pipework (William, 2015). There are two distinct advantages of a run-around-coil heat exchanger. Brucker (2015) affirms that the probability of cross-contamination between the two streams is close to impossible. The second advantage is that the waste heat may be transported over large distances (Brucker, 2015). The size of the pump and heat losses over the distance determine the maximum distance. According to (William, 2015), the maximum overall heat exchanger efficiency that can be achieved is around 60%.

The gasketted-plate heat exchanger is commonly used for a variety of liquid-to-liquid heat transfer applications (Bai & Xu, 2011). The unit consists of a series of thin corrugated plates packed together using gaskets. The hot and cold fluids flow in counter-current configuration in the adjacent flow channels created by the plate structure (Laval, 2016). The corrugated plates create a large heat transfer area for this type of technology. This type of heat exchanger can be easily opened for maintenance and cleaning, and thus becomes suitable for applications where fouling can occur (Smith, 2005). Gasketted-plate heat exchangers are available off-the-shelf at very low costs (Laval, 2016). Since this unit uses gaskets, there are huge pressure and temperature limitations related to them (Bai & Xu, 2011).

The brazed plate heat exchanger operates in the same mode as the gasketted-plate unit discussed above. The unit flow regime is identical to that of the gasketted-plate heat exchanger (Bai & Xu, 2011). The plate structure of the brazed plate heat exchanger is held together by brazing as opposed to gaskets. According to (Laval, 2016) the most common brazing used in these types of heat exchangers is copper brazing. The advantage of the copper brazing in these types of heat exchangers is that it allows for high temperature and pressure limits (Smith, 2005). Brazed heat exchangers cannot tolerate fouling or solid particles in the heat source or sink, but other advantages such as the compact size and high efficiency remain (Smith, 2005). The brazed plate heat exchanger is more costly than that the gasketted-plate heat exchanger due to costs associated with copper brazing compared to gaskets (Laval, 2016).

The finned-tube heat exchanger is an extended surface heat exchanger designed specifically for duties where one of the fluids has a significantly smaller film heat transfer coefficient than the other (Ahmad & Shafie, 2011). It is commonly utilised in gas-to-liquid heat transfer as gas film coefficients are typically around a tenth of the value for liquids (Walsh, 2011). The finned-tube heat exchanger is commonly packaged in a shell-and-tube type configuration, and hence shares similarly high temperature and pressure limitations (Miller, et al., 2015). The unit can also tolerate fouling due to the ability to remove the tube bundle for cleaning while protective coatings can be applied to the tubes to withstand any corrosion particularly in duties such as economisers (Bai & Xu, 2011). This unit is viewed as the industry standard in extended surface heat transfer; therefore, the finned-tube heat exchanger is seen as an ideal unit for gas-to-liquid duties where an extended surface is necessary due to its operational flexibility (Bai & Xu, 2011).

The spiral plate heat exchanger is configured as two elongated plate channels rolled around a central core (Ahmad & Shafie, 2011). The heat source and sink flow in counter-current configuration in adjacent plates. The smooth and curved channels of the unit tend to reduce fouling (Aneke, 2012). Fouling is further reduced as any local fouling will result in a reduction in the channel cross-sectional area which in turn increases the fluid velocity, creating a scouring effect to clean the channel (Law, 2014). The spiral heat exchanger is generally considered for use when dealing with highly fouled or slurry fluids. Heavily fouling fluids can be accommodated on both sides of this heat exchanger (Aneke, 2012).

For complex dynamics fluids, the scraped surface heat exchanger has been specifically developed for heat transfer purposes (Law, 2014). According to (Span, 2014) complex dynamics fluids can include concentrated slurries, highly viscous fluids or non-Newtonian fluids. The unit is essentially a double pipe heat exchanger where the heating or cooling media flow in the outer pipe (Aneke, 2012). Complex dynamics fluid flows in the inner pipe, in which a blade rotates to scrape any solids from the heat exchanger wall (Law, 2014). This prevents fouling build up and ensures uniform heat transfer throughout the unit. A large range of scraped-surface heat exchangers are available for liquid-to-liquid heat transfer or evaporation duties (Laval, 2016). It is highly uncommon that one would recover waste heat from a highly viscous and potentially valuable product, thus the use of this heat exchanger for waste heat recovery is uncommon (Law, 2014).

According to (Holman, 2009), the spray recuperator is a type of direct contact heat exchanger designed for gas-to-liquid heat transfer duties. According to (Incropera & De Witt, 2014), the use of the spray recuperator was suggested by Lyle in 1947 to recover latent heat and water from the evaporative processes for use as site wash water. The general configuration of the spray recuperator is a nozzle which sprays a fine mist of liquid, usually water, into a stream of vapour, usually steam, for heat transfer and the resulting hot water then flows out of the spray recuperator for use where needed. (Holman, 2009). This is often done in a counter-current configuration with a packed bed to increase surface area (Incropera & De Witt, 2014).

6.2.2 Heat Pumps

Eckert (2011) suggest that heat pumps are an important technology allowing the upgrade of low-grade waste heat to a more useful temperature according to temperature lift limitations. Heat pumps should be considered when no matching heat sources are available for heat transfer as they required a significantly higher capital expenditure than a simple heat exchanger system (Law, 2014). Various types of heat pumps are available to perform this task, however, the most commonly used types are vapour compression heat pump and mechanical vapour compression heat pumps (Law, 2014). The closed cycle vapour compression heat pump acts as a reverse Rankine cycle where work is put into a compressor most commonly from an electric drive to create a temperature lift between the two heat exchangers of the cycle (Aneke, 2012).

In a vapour compression heat pump, the evaporator utilises waste heat to vaporise the working fluid at the low temperature and pressure end of the cycle (Aneke, 2012). This low-pressure vapour is then compressed to high pressure upon work input from the compressor or drive (Aneke, 2012). The working fluid is then condensed at this higher pressure and temperature in the condenser, thereby heating the heat sink to a higher temperature than possible if one was to use a heat exchanger to directly transfer waste heat from the source to the sink (Aneke, 2012). Finally, a throttle valve is then used to reduce the working fluid pressure and complete the cycle (Law, 2014).

Law (2014) states that mechanical vapour recompression is an open cycle variation of the traditional vapour compression heat pump. Here, low-pressure vapour from an evaporative process is re-compressed via a mechanical compressor or drive to a higher pressure and then used as a heating medium in the process, thereby negating the need for an external utility at steady-state operation (Aneke, 2012). Mechanical vapour recompression can provide a waste heat recovery for many evaporative processes used throughout the processing industries (Ahmad & Shafie, 2011). Heat pump performance is determined by the coefficient of performance (COP) which is defined as the heating duty provided after the compressor over the work of the drive (Bai & Xu, 2011). The COP determines both the economic and environmental impact of the heat pump, thus high COP suggests low payback period and high emissions reduction (Bai & Xu, 2011).

6.2.3 Organic Rankine Cycle

The waste-heat-driven power generation is an increasingly attractive proposition for process plants due to the rising cost of electricity (Beggs, 2002). There are a number of methods available for waste heat driven power generation with the common one being the Organic Rankine Cycle (ORC) (Chen, 2010). The ORC has a number of advantages over other methods, such as, the ability for using low-temperature waste heat which makes it a better alternative for low-grade temperature applications (Aneke, 2012). Another advantage is that the working fluid of the ORC tends to have a lower heat of vaporization than water, and becomes superheated more easily using low-grade waste heat (Aneke, 2012). ORC systems contain the four key components: the working fluid pump, the pre-heater or evaporator, the turbine or generator and the condenser, and each component's efficiency have a direct impact on the performance of the entire cycle (Chen, 2010).

6.3 Selection Process

There are a number of considerations that need to be taken into account in the selection process for the type of heat transfer technology to be used. While there is a number of decision-making methods related to the selection of suitable heat transfer method that can be applied in a heat recovery process, the decision tree method is usually the easiest to apply. Normally known as the IF-THEN method, in a decision tree method, decisions can follow in succession or in a parallel manner to build multiple parameters in a decisions process. The use of the decision tree method is distinct in that selection of the process is based on the user-defined decisive factor and the process does not require complex analytical processes to make a decision (Law, 2014).

Quantitative and qualitative input data of both the heat source and the heat sink is crucial in the development of user-defined specifications for the decision criteria in the decision tree method. According to (Law, 2014), the heat source and heat sink phases, mass flow-rates, temperatures, specific heat capacities, pressures, densities, viscosities, thermal conductivities and heat transfer coefficients are the most common quantitative data that must be known by the user to formulate the decision in the decision tree method. Furthermore, the target or output results must be clearly known to the user to formulate a decision. For qualitative data, information related to nature of solids contained in heat source and heat sink, fouling tendencies, scaling possibilities, corrosive nature of fluids, required material of construction, need for cleaning and accessibility becomes very important to select the best heat transfer method for the particular process (Law, 2014).

The initial step in the selection of the heat transfer technology for the heat recovery system is fairly easy, as it relates to selecting the category of heat transfer which could either be: heat exchangers, heat pumps or ORC for power generation. Taking into account the quantitative and qualitative data discussed above, the initial process eliminates the technology that cannot be used as per the user-defined criteria. According to (Aneke, 2012) the initial step can be stated as follows:

- If there is no heat sink to match the available heat source, then one cannot use a heat exchanger.

- If a lower limit of air source is below -10 ^{0}C, then one cannot use an air source heat pump.

- If the heat source volumes are too low for cogeneration, then one cannot use the ORC.

In the initial step, in order to eliminate the possible use of heat exchangers, there must be no match between the heat sink and the heat source. This means that the heat source or heat sink streams are probably at the same temperature or the differential temperature between the two streams is very low. For an air source heat pump, the inlet air needs to be above -10 ^{0}C in order to be able to use the heat pump, air temperature below -10 ^{0}C tends to affect the performance of the heat pump and thus do not add any benefit to operating the heat pump for energy saving (Law, 2014). The decision tree method for the initial step in the selection process is shown below.

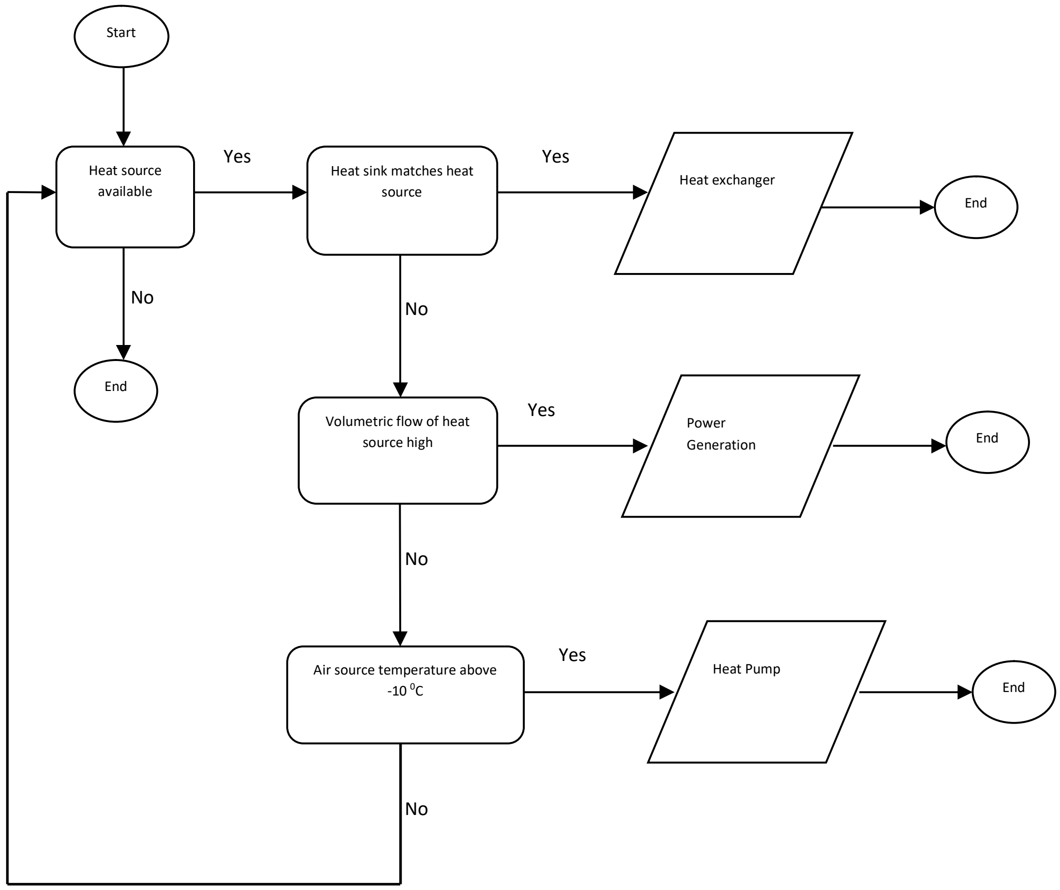


Figure 6.20: The initial step selection decision tree

When the decision has been made in relation to the technology category to be used for heat recovery, the final step, which involves the selection of the type of design features in that particular technology, is very important. For heat exchangers, the decision involves selecting between the gas-to-gas, gas-to-liquid or liquid-to-liquid heat exchanger design based on the phases of the streams. Furthermore, the pressure and temperature rating of the fluids, fouling tendencies, cleaning requirements and corrosive nature of fluids determine the type of heat exchanger to be used. In the selection of heat pumps and ORC for power generation, the decision focuses on the selection of the most appropriate working fluid for the operating range and the health and safety compliance related to the use of such a working fluid. The lower volumetric or mass flow rates of the heat source extend greatly the payback period for costs associated with the installation of the power generation and thus do not support the installation of the ORC for cogeneration.

6.4 Conclusion

Different heat transfer technologies were theoretically evaluated to outline the different applications suited for a particular heat transfer technology. Different types of heat exchangers were compared based on their capabilities and suitability for different stream phases and qualities thereof. The use of heat pumps and performance ratings was briefly discussed. Quantitative and qualitative data required for the selection of the heat transfer technology appropriate to the heat recovery process was discussed. A structured method of decision making with regards to selecting the most appropriate heat transfer technology was discussed.

The next chapter outlines the contributions to the optimization of production and energy demand as a result of recovering exhaust heat.

CHAPTER SEVEN – PRODUCTION OPTIMIZATION

7.1 Introduction

This chapter assesses the methods by which the energy that is lost to the atmosphere can be used to optimize production or energy demand, as a result of using recovered energy. The scope of this chapter includes the selection of appropriate heat recovery technique suitable to recover energy losses for each phase across the different stages of production. Moreover, the way in which the recovered energy is utilised in the process is discussed in this chapter. A theoretical method of analysing suitable heat transfer methodology is used to select the most appropriate method for each phase of heat loss. A mathematical method of evaluating uses of recovered energy is employed to ascertain the best way of using recovered energy. Literature related to the subject under evaluation is briefly discussed where necessary.

7.2 Air Heater Losses Recovery

When large volumes of hot exhaust gases from the air heater are discharged to the atmosphere, a substantial quantity of valuable heat is lost. This heat can be easily recovered and used for pre-heating of incoming fresh air. The low specific heat capacity of flue gases, however, presents a challenge in making them viable for being directly used as a heating medium of the incoming fresh air. For this reason, designers have concluded that one of the best ways by which to recover hot exhaust gases is by making use of the condensing economiser. A direct contact condensing stack economiser recuperates the heat contained in the stack flue gases and transfers it to a cold water stream to render a hot water stream at temperatures as high as 70 ^{0}C (Pawel & Kazimierz, 2014).

In a condensing economiser, cold water enters the unit at the top and is then uniformly distributed over the upper surface of a packing of stainless steel nodules which constitute the heat transfer zone (Kamal, 2017). The cold water percolates down through the packing where it comes into direct contact with the hot, rising flue gases. The most common practice is to ensure that the incoming water enters the flow distributor where it is evenly spread out over the packed mass heat exchanger. The hot water collecting at the bottom of the economiser is then pumped directly to the process heat exchanger to transfer its energy to a process fluid. According to (Pawel & Kazimierz, 2014), the efficiency of heat transfer in condensing stack economisers is very high, and the exhaust discharged to the atmosphere is normally cool.

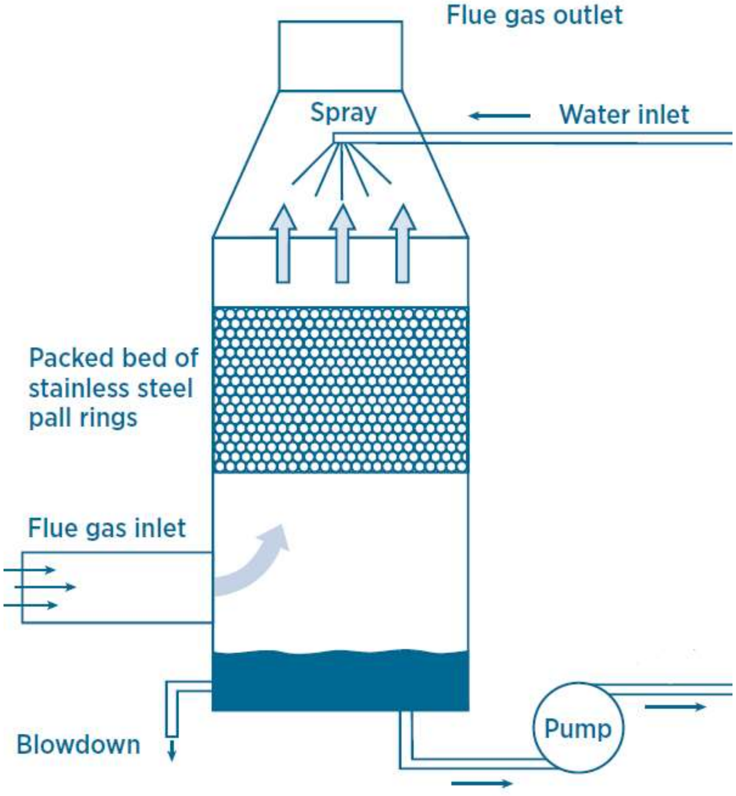


Figure 7.31: A direct contact condensing economiser **(Laval, 2016)**.

In condensing economisers, both the sensible and latent heat contained in the flue gases is transferred to the water. The advantages of the direct contact condensing stack economiser over traditional surface heat exchangers are that they are highly efficient, easy, cheap and safe to install, reduce the emission of atmospheric pollutants, does not require supervision, operates at atmospheric pressure and requires minimal maintenance (Pawel & Kazimierz, 2014). A direct contact heat exchanger eliminates metallic heat transfer surfaces which are prone to corrosion and fouling. It can be operated at very low-temperature differences or heat transfer driving forces and allows lower mass flow rates of transferring fluids (Kamal, 2017). Considering a condensing economiser as a simple direct-contact heat exchanger:

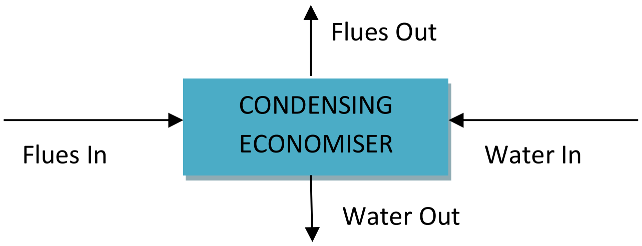


Figure 7.32: Material movement in the economiser

The overall mass balance is given by:

$$\dot{m}_{fli} + \dot{m}_{wi} = \dot{m}_{flo} + \dot{m}_{wo} \tag{7.1}$$

The total energy balance:

$$\dot{m}_{fli}h_{fli} + \dot{m}_{wi}h_{wi} = \dot{m}_{flo}h_{flo} + \dot{m}_{wo}h_{wo} \tag{7.2}$$

The above can be written in terms of specific heat capacity and temperature as:

$$\dot{m}_{fli}c_{p_{fli}}T_{fli} + \dot{m}_{wi}c_{p_{wi}}T_{wi} = \dot{m}_{flo}c_{p_{flo}}T_{flo} + \dot{m}_{wo}c_{p_{wo}}T_{wo} \tag{7.3}$$

Finally, the temperature exiting the economiser is determined from:

$$T_{wo} = \frac{\left(\dot{m}_{fli}c_{p_{fli}}T_{fli}\right) + (\dot{m}_{wi}c_{p_{wi}}T_{wi}) - (\dot{m}_{flo}c_{p_{flo}}T_{flo})}{\dot{m}_{wo}c_{p_{wo}}} \tag{7.4}$$

From equation 7.4, the mass of water flow out can be presented with respect to the overall mass balance in equation 7.1.

$$T_{wo} = \frac{\left(\dot{m}_{fli}c_{p_{fli}}T_{fli}\right) + (\dot{m}_{wi}c_{p_{wi}}T_{wi}) - (\dot{m}_{flo}c_{p_{flo}}T_{flo})}{(\dot{m}_{fli} + \dot{m}_{wi} - \dot{m}_{flo})c_{p_{wo}}} \tag{7.5}$$

The equation above gives an indication of the temperature of the water outlet stream that can be achieved in a direct contact heat exchanger based on the flow rates and temperatures of the water inlet streams assuming that the flue gases stream does not change throughout the process. It is also normal that the water inlet stream temperature remains the same throughout the process as it is usually at atmospheric temperature. The temperature of the flue gases exiting the stack is critical in ensuring that the operation of the economiser is efficient. According to (US Energy, 2014), condensing economisers can be designed such that the exiting flue gases temperature is minimized to 65 ^{0}C.

Equation 7.5 above can be used to calculate the water outlet stream temperature.

Assuming water inlet flow of 5000 kg/hr at a temperature of 25 ^{0}C and flue gases exiting at 65 ^{0}C. Referring to Figure 4.21, at this temperature, the specific heat capacity of the gases is approximately 1.006 kJ/kgK.

$$T_{wo} = \frac{\left(\dot{m}_{fli} c_{p_{fli}} T_{fli}\right) + (\dot{m}_{wi} c_{p_{wi}} T_{wi}) - (\dot{m}_{flo} c_{p_{flo}} T_{flo})}{(\dot{m}_{fli} + \dot{m}_{wi} - \dot{m}_{flo}) c_{p_{wo}}}$$

$$= \frac{(4645.320 \times 1.14 \times 265) + (5000 \times 4.187 \times 25) - (4645.320 \times 1.006 \times 65)}{(4645.320 + 5000 - 4645.320) \times 4.187}$$

$$= 77.52\ ℃$$

The driving force of the condensing economiser is the temperature of the exiting flue gases. If the temperature of the flue gases is low, it means that the rich energy of the incoming flue gases was effectively transferred to the water. The relationship between the water outlet temperature and water inlet flow rate is derived from the equation 7.5 and presented graphically below.

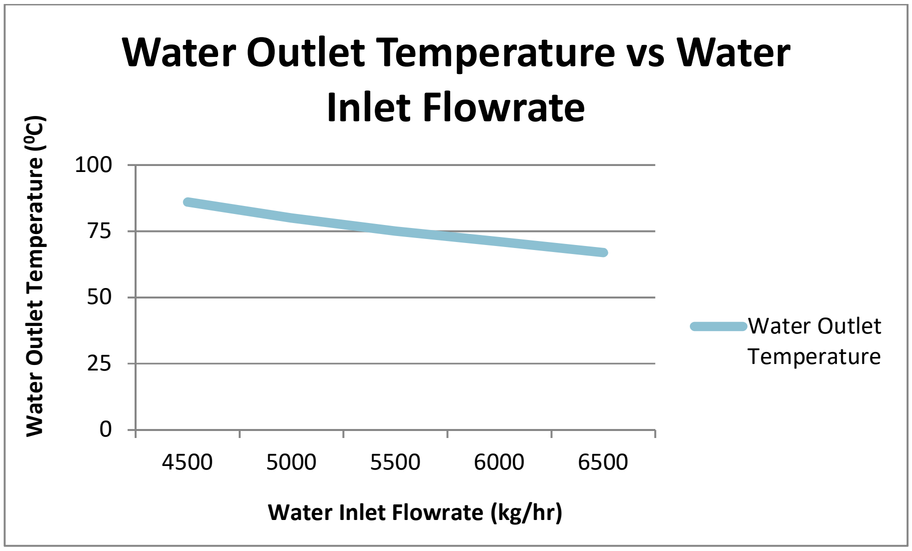


Figure 7.33: Relationship of water inlet flow rate and water outlet temperature in a condensing economiser.

The water outlet stream temperature decreases with the increase in water inlet stream flow rate. This increase in water inlet stream flow rate increases the heating duty of the system. The higher the inlet flow rate, the more the energy needed to get the stream to the required temperature. Heat duty is defined as the amount of energy the heat exchanger must transfer to the process fluid per unit time in order to heat it to the desired temperature (Eckert & Drake, 2013). The heat duty is vital as the suitability of the heat exchanger to give the process fluid the desired specifications is determined from it.

According to (Perry, 2014) if the system air heating is properly sealed for air leakages, it is expected that the non-condensable gases be minimal; however, it cannot be always guaranteed that the air heating system is completely sealed from ingress of air.

The properties of the recovered water can be written as:

$$\dot{m}_{wo} = 5\,000\ kg/hr\,; \quad c_{p_{wo}} = 4.187\ kJ/kgK \quad \& \quad T_{wo} = 77.52\ ℃$$

The recovered energy:

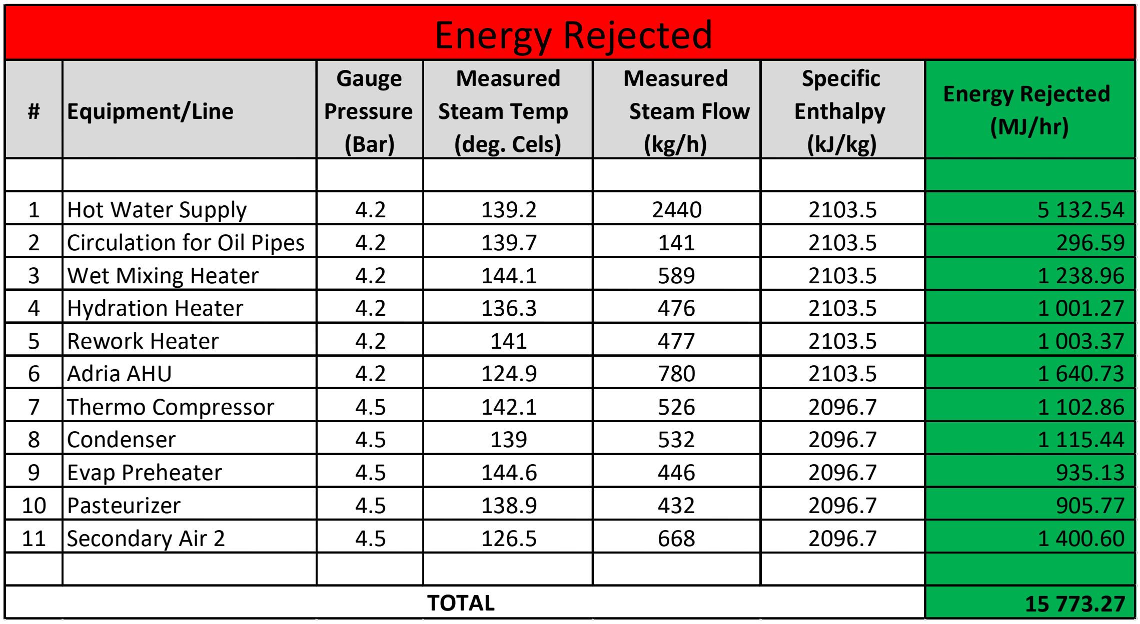
$$Q_{wo} = \dot{m}_{wo} \times c_{p_{wo}} \times (T_{wo} - T_{wi})$$

$$= 5000 \times 4.187 \times (77.52 - 25)$$

$$= 1\,099.51\ MJ/hr$$

Due to efficiency losses, reticulation losses, exhaust losses and other environmental losses associated with the heat exchange process, it is expected that not all of the recovered energy would be put to effective use. Moreover, the calculated recovered energy is based on a 100% effective recovery system, which is not normally the case in process plants. The assumption is that at least 50% of the recovered energy would be put to effective utilization while the other 50% would be lost in other forms as mentioned above. The losses incorporate both the energy recovery losses and energy utilization losses. This assumption renders the effective utilization of recovered energy equivalent to 549.76 MJ/hr.

With reference to Table 5.2; the energy rejected by steam can be calculated by multiplying the steam flow rate and specific enthalpy of steam at a particular pressure. The energy rejected by steam in different process steps is an essential parameter in comparison to the energy received by the process fluid. While the process fluid indicates the amount of heat effectively removed from the heat exchanger, it neglects the energy lost by the system during the heat exchange process.



Energy Rejected

#	Equipment/Line	Gauge Pressure (Bar)	Measured Steam Temp (deg. Cels)	Measured Steam Flow (kg/h)	Specific Enthalpy (kJ/kg)	Energy Rejected (MJ/hr)
1	Hot Water Supply	4.2	139.2	2440	2103.5	5 132.54
2	Circulation for Oil Pipes	4.2	139.7	141	2103.5	296.59
3	Wet Mixing Heater	4.2	144.1	589	2103.5	1 238.96
4	Hydration Heater	4.2	136.3	476	2103.5	1 001.27
5	Rework Heater	4.2	141	477	2103.5	1 003.37
6	Adria AHU	4.2	124.9	780	2103.5	1 640.73
7	Thermo Compressor	4.5	142.1	526	2096.7	1 102.86
8	Condenser	4.5	139	532	2096.7	1 115.44
9	Evap Preheater	4.5	144.6	446	2096.7	935.13
10	Pasteurizer	4.5	138.9	432	2096.7	905.77
11	Secondary Air 2	4.5	126.5	668	2096.7	1 400.60
TOTAL						**15 773.27**

Table 7.3: Energy rejected by steam on different process steps.

From the table above, the energy needed for the circulation of oil pipes matches the energy recovered by the condensing economiser process. This means that the energy recovered by condensing economiser process from the exhaust of the air heating system can be used as a substitute heat source for the circulation for oil pipes process. The decision tree in Figure 6.30 shows that if there is a matching heat sink to the heat source, the heat exchanger can be used. Since shell-and-tube heat exchangers can facilitate heat transfer between two liquid streams, and they can handle high pressures and temperatures applications, and can tolerate fouling; they can be used in this process. The suitable heat exchanger must have a temperature approach of about 10 ^{0}C. Temperature approach is defined as the difference between the initial temperature of the heat source and the final temperature of the heat sink (Botes, 2007). This temperature approach is reasonably well within the limits of industrial heat exchanger designs, and thus the recovery will be possible.

7.3 Reticulation Losses Recovery

Product drying air reticulation losses are of a major concern because they occur before work is done in the system. Air heater losses and drying losses occur after work is done in their respective systems and thus their recovery is a matter of energy demand optimization only. Since product drying air reticulation energy losses occur before the energy is used to do work, recovery of these losses has a potential to optimize both production capacity and energy demand of the spray-drying process.

In the existing set-up, the product drying air exiting the air heater is initially at 355 ^{0}C. This air is distributed to the drying chamber through a 70 m long insulated pipe. At the exit from the pipe, which is the inlet to the drying chamber, the air temperature is 340 ^{0}C. The 15 ^{0}C temperature drop through the 70 m pipe is significant as it reduces the water holding capacity of drying air. In new designs, the norm is to place the air heating system as close as possible to the drying chamber (Hugo, 2017). This reduces the temperature drop, heat losses and thus maintains a higher drying air temperature. There is sufficient space available on the drying chambers to move existing air heaters closer to the drying chamber (Hugo, 2017). It is hypothesized that the new designs can reduce the reticulation length from 70 m to 10 m.

Reduction in reticulation length increases the temperature gradient along the pipe length. The temperature gradient is inversely proportional to the temperature drop, and thus, an increase in temperature gradient results in reduced temperature drop. The temperature gradient is simply defined as a rate of change of temperature over a given length. In heat transfer, the rate at which energy is conducted as heat between two bodies is a function of the temperature gradient between the two bodies and the properties of the conductive medium through which the heat is transferred (Eckert, 2011).

For a 70 m pipe length:

$$T_{g_{70}} = \frac{355}{70}$$

$$= 5.07\ ℃/m$$

For a 10 m pipe length:

$$T_{g_{10}} = \frac{355}{10}$$

$$= 35.5\ ℃/m$$

The factors which affect temperature drop through air reticulating ducts include: the temperature at which air enters the duct, the ambient temperature surrounding the duct, the flow rate of air reticulated in the duct, the length of the duct, the perimeter of the duct and the resistance offered by the insulation used around the duct (NMIC, 2016). In calculating temperature drop, it is vital that the specific heat capacity of air remains constant, and the impact of duct leakage is neglected as it complicates the calculations (NMIC, 2016). In moving the air heating system closer to the drying chamber, all parameters remain the same and the only parameter that changes is that of the length of the duct. Therefore, the product of the temperature drop and temperature gradient of the existing system would be equal to that of the new system.

$$T_{d_{10}} \times T_{g_{10}} = T_{d_{70}} \times T_{g_{70}}$$

$$T_{d_{10}} = \frac{15 \ \times 5.07}{35.5}$$

$$= 2.14\ ℃$$

The reduced temperature drop above will result in the product drying air entering the drying chamber at a higher temperature and the heat energy lost through reticulation will also be reduced. The heat loss through reticulation in the new design of shortened length can be calculated from the ratio of change of length between the existing system and the new system, assuming that all parameters in the system remain unchanged.

$$Q_{L_{10}} = Q_{L_{70}} \times \frac{L_{10}}{L_{70}}$$

$$= 139.32 \times \frac{10}{70}$$

$$= 19.90\ MJ/hr$$

The new system results in a saved energy loss equivalent to 119.42 MJ/hr. The energy saved by adopting a new system design is then used in the drying chamber for removal of water in the concentrated product. The energy rejected by drying air in the existing system yields to the current final product output as calculated in the existing drying system. Therefore, the energy rejected by air in the new system will yield to a new final product output.

Energy rejected in the existing system:

$$Q_{70} = \dot{m}_a \times c_{p_a} \times (T_a - T_{atm})$$

$$= 25\,147.96 \times 1.006 \times (340 - 15)$$

$$= 8\,222.13 \ MJ/hr$$

Energy rejected in the new system:

$$Q_{10} = Q_{70} + 126.93$$

$$= 8\,349.06 \ MJ/hr$$

The final product output from the new system is given by:

$$\dot{m}_{FP_{10}} = \dot{m}_{FP_{70}} \times \frac{Q_{10}}{Q_{70}}$$

$$= 4670 \times \frac{8349.06}{8222.13}$$

$$= 4742 \ kg/hr$$

This shows that adopting a new reticulation system design would result in an increase in production capacity equivalent to 72 kg/hr.

7.4 Drying Chamber Losses Recovery

The energy losses in the drying chamber occur in both the primary and secondary dryer respectively. The recovery of these energy losses must occur in two stages, as the primary and secondary dryer losses are at different temperatures. The calculated energy losses in the secondary dryer are 938 MJ/hr and that of the primary dryer are 2 850 MJ/hr. The recovery of these losses can occur in a shell-and-tube heat exchanger since periodic cleaning will be required due to powder particles contained in the exhaust air. In a shell and tube heat exchanger, product drying air at ambient temperature will receive the heat from the exhaust air of the drying chambers to increase its temperature before it goes to the air heater. The effective recovered heat that will be utilized from the dryers is 469 MJ/hr and 1 425 MJ/hr respectively, based on a 50% effective utilization assumption.

For the Secondary Dryer:

$$Q_{EXH2_{eff}} = \dot{m}_a \times c_{p_a} \times (T_2 - T_{atm})$$

$$469 \times 10^3 = 25\ 147.96 \times 1.006 \times (T_2 - 15)$$

$$T_2 = \frac{469 \times 10^3}{25\ 147.96 \times 1.006} + 15$$

$$= 33.5\ ℃$$

Utilization of energy from the secondary dryer would increase the temperature of product drying air from 15 ^{0}C to 33.5 ^{0}C before entering the main air heater.

For the Primary Dryer:

$$Q_{EXH1_{eff}} = \dot{m}_a \times c_{p_a} \times (T_3 - T_2)$$

$$1425 \times 10^3 = 25\ 147.96 \times 1.006 \times (T_3 - 33.5)$$

$$T_3 = \frac{1425 \times 10^3}{25\ 147.96 \times 1.006} + 33.5$$

$$= 89.8\ ℃$$

The final stage of product air heating before it enters the main air heater would increase the temperature of product drying air from 33.5 ^{0}C to 89.8 ^{0}C. This means that the main air heater would now be required to raise the temperature of product drying air from 89.8 ^{0}C to 355 ^{0}C as opposed to the existing requirements of air temperature from 15 ^{0}C to 355 ^{0}C. From the main heater efficiency calculation, the mass flow rate of oil required for the new conditions can be calculated:

$$\eta_o = \frac{\dot{m}_{pa} \times c_{p_{pa}} \times (T_{pa\ out} - T_{pa\ in})}{\dot{m}_{oil} \times CV_{oil}}$$

$$\dot{m}_{oil} = \frac{25\ 147.96 \times 1.006 \times (355 - 89.8)}{0.83 \times 39.57 \times 10^3}$$

$$= 204.28\ kg/hr$$

The use of recovered energy from the exhaust air of the drying system reduces the consumption of fuel used in the main air heaters from 261.9 kg/hr to a significant 204.28

kg/hr of HFO. The reduction of 57.62 kg/hr is considerably important as the price of fuel is excessively increasing.

7.5 Conclusions

The methods by which the energy that is lost to the atmosphere can be used to optimize production and energy demand, as a result of using recovered energy was assessed. The suitable method of recovery was selected and explained, moreover, the technology selected for each recovery stream was also detailed. A mathematical method of evaluating effective use of recovered energy was performed to optimize production and energy demand. The aim of recovering exhaust energy was to use it as a substitute heating medium in the place of steam. Steam is usually the most expensive form of energy in process plants. Reduction of steam consumption in the factory results in reduced operational costs associated with steam production. However, the low-grade heat from the exhausts of the spray-drying process does not match the requirements of the process heat sinks, due to inadequate temperature approach.

The next chapter compares the cost associated with the recovery of exhaust energy, in order to identify the financial benefit of recovering exhaust energy in the process.

CHAPTER EIGHT – FINANCIAL ANALYSIS

8.1 Introduction

When it has been established that the project is viable technically, there is a need for economic viability evaluation to assist the decision-makers with the decision of whether to or not to implement the project. This chapter establishes whether the project is worth being executed economically. The economic analysis of this project is evaluated through profitability assessment study. The scope of this chapter includes multiple investments analysis criteria, to have a holistic view of the support factors for decision making. A mathematical method of investment analysis is employed to ascertain the profitability of this project. Literature related investment analysis is briefly discussed where necessary.

8.2 Project Profitability Assessment

In today's highly competitive global economy, businesses can no longer take their profitability for granted. Maintaining the status quo of the business does not ensure business success. Every project initiative needs to be efficient, sustainable and productive in order for businesses to remain competitive in the marketplace. Profitability results from revenues and expenses. Revenues are generated from selling goods or services and expenses are incurred as part of delivering those goods or services, or supporting the delivery of goods or services. A key profitability measure is the net profit margin. While organisations use different profitability ratios to measure their overall profit margin, the most commonly used profitability ratios in supporting engineering projects are Payback Period, Net Present Value, and Accounting Rate of Return. Profitability ratios are a class of financial metrics that are used to assess a business's ability to generate earnings compared to its expenses (Fisher, 2002).

The payback period is the period of time that a project requires to recover money initially invested in it. The length of time is usually expressed in years, and the project with the lower payback period is usually favoured over the one with a higher payback period. The disadvantage of a payback period calculation for profitability assessment is that the time value of money is not considered. Time value of money states that money that is available at the present time is worth more than the identical sum in the future due to its potential earning capacity (Ardalan, 2012). The advantage of the payback period in profitability assessment is that it is a useful and quick capital budgeting method and it is very easy to compute. When the net annual cash inflow is even, the payback period can be computed by applying the formula below:

$$Payback\ Period = \frac{Total\ Capital\ Investment}{Net\ Cash\ Flow} \tag{8.1}$$

The Net Present Value (NPV) is the present value of the cash flows at the required rate of return on your project compared to your initial investment. It is the value of all future cash flows incoming and outgoing, over the entire life of an investment discounted to the present. The NPV can either be positive or negative. A positive NPV indicates a net gain, and a negative NPV indicates a net loss, and thus projects with a negative NPV are rejected. While positive NPV indicates a net gain, projects with higher NPV are usually given preference where multiple projects are considered. NPV methodology is preferred over the payback period calculation because it considers the time value of money by considering the discounted rate of return. The rate of return usually varies throughout the project, and thus the NPV in which the discounted rate changes is considered to reflect the real-life behaviour of the project profitability. NPV is calculated from the formula below:

$$NPV = -C_o + \sum_{i=1}^{T} \frac{C_i}{(1+r)^i} \tag{8.2}$$

Where,

C_o is the initial investment

C_i is the cash flow

r is the discounted rate

T is the time in years

The Accounting Rate of Return (ARR) is used in investment appraisal. It is defined as the ratio of the average accounting profit of a project to the average investment made in the project. In comparing two projects, the decision rule is that the project with a greater ARR is more viable than the project with a lower ARR. The advantage of ARR is that it is easy to calculate and it recognizes the profitability factor of investment. However, like the payback period calculation, it ignores the time value of money (Ardalan, 2012). According to (Fisher, 2002), ARR is calculated in different ways and thus presents an inconsistency problem. ARR is not suitable for projects with high maintenance costs because their viability depends on the timely inflows of cash (Ardalan, 2012). The ARR of the project can be computed from the formula below:

$$ARR = \frac{Average\ Profit}{Average\ Investment} \tag{8.3}$$

8.3 Costing Estimations

The project profitability assessment requires that the outflow costs and inflow revenues for each project be determined. These outflow costs normally consist of initial capital investments and installation costs. The inflow revenues usually consist of sales profit, savings, operational and maintenance costs throughout the cycle of the project. Equipment suppliers play a crucial role in helping engineers estimate costs related to any project. Equipment suppliers provide specifications of equipment which help in identifying sources of savings as a result of initiating a particular project. Moreover, in the absence of such specifications, the available literature and economic trends are used to estimate the costs involved in a project.

Engineering projects costs are usually divided into direct costs and indirect costs. Direct costs are easy to estimate. In outflow costs, they usually consist of cost to purchase equipment, installation cost, material and labour costs. Indirect costs are difficult to estimate. Indirect costs can be for the outflow costs and inflow revenues. They are usually insurance costs, taxes, engineering costs, and other expenses. Normally, engineers use the correlation of similar or previous projects to estimate the percentage of these costs and thus use the percentage to calculate the indirect costs for each project. The costs associated with the implementation of the energy recovery process can be deduced from the information in the previous chapter.

8.4 Estimation of Outflows

For the project in question, the estimation of cash outflows for the first system of recovery includes the cost of purchasing a new condensing economiser, the purchasing of the pump system for the economiser, the installation of economiser and pump, anchoring in place, rerouting of fuel supply, the lagging and cladding of pipe systems. Due to unavailability of purchase and installation costs from the condensing economiser suppliers, the cost-size relationship from the price lists of the previous various similar projects was used to estimate the outflow cost. The effective estimated cost equates to R 1 800 000.00

The removal of air heaters from their original position to the new proposed position was estimated using the correlation relationship of moving machinery from one location to another. The factors that played the part in cost estimation included rigging, cutting, moving, installation and lagging and cladding. The effective estimated cost equates to R 18 700 000.00

Finally, the cost of purchasing shell-and-tube heat exchangers, that can handle about 25 000 kg/hr of air was estimated by comparing the price of smaller shell-and-tube heat exchangers which have been purchased before. The installation cost, including the lagging and cladding, was also estimated using the similar approach. The effective estimated cost for dryer recovery equates to R 586 000.00

In all the estimation of outflow costs, 25% was added to incorporate all unforeseen costs. Moreover, this would incorporate indirect costs that could be incurred. The total capital investment was estimated at R 26 357 500.00

8.5 Estimation of Inflows

The cost gained in relation to the recovery of exhaust energy from the air heaters could be derived from the amount of steam that would be saved as a result of exhaust energy recovery. The cost of generating steam would be reduced, and thus, the amount saved would be the revenue generated. The steam savings amounts to 300 MJ/hr, and considering that the circulation of oil pipes operates approximately throughout the year, the steam energy saved can be approximated as follows:

$$Energy\ Saved = \frac{296.59\ MJ}{1\ hr} \times \frac{24\ hrs}{1\ day} \times \frac{365\ days}{1\ year}$$

$$= 2\ 598\ GJ/year$$

In order to translate the above energy savings into the mass of fuel saved, the following equation can be used:

$$Mass\ of\ fuel = Energy\ saved\ \times \frac{1}{Calorific\ Value\ of\ fuel}$$

$$= \frac{2598\ GJ}{year} \times \frac{1\ kg}{39.65\ MJ}$$

$$= 65\ 523\ \ kg/year$$

For the price of HFO fuel at R6.35 per kilogram, the effective revenue generated from the above saving of HFO equates to R 416 071.05 annually.

In the previous chapter, calculations conducted revealed that production capacity could be increased by 75 kg/hr and the HFO oil reduced by 57 kg/hr as a result of employing energy recovery systems. For the efficiency of the production plant of 51%, the annual savings in fuel can be calculated.

$$Mass\ of\ fuel = \frac{57\ kg}{1\ hr} \times \frac{168\ hrs}{1\ week} \times \frac{52\ weeks}{year} \times 51\%$$

$$= 253\ 955\ \ kg/year$$

The effective revenue generated from the saving HFO equates to R 1 612 614.25 annually.

Applying the same formula for the calculation of the increase in powder produced would give:

$$Mass\ of\ powder = 334\ 152\ \ kg/year$$

However, the assumption used in the powder production capacity increase is that all recovered energy would be transferred to a drying chamber. Incorporating the losses assumed at 50% of both recovery system and usage system would result in powder increase of about 167 076 kg/year. Working on the retail price of R 51.65 for 1 kg of milk powder; the generated revenue for increased production capacity would give R 8 629 475.40 annually. The total revenue generated was estimated at R 10 658 160.70 annually.

In all the estimation of inflow costs, 25% was removed to incorporate all unforeseen costs that would most likely reduce the final outcomes. It is important to remove such costs as there may be losses and changes that could potentially reduce the estimated cost. The final total revenue generated would effectively be R 7 993 620.53.

8.6 Profitability Calculations

The Payback Period:

$$Payback\ Period = \frac{Total\ Capital\ Investment}{Net\ Cash\ Flow}$$

$$= \frac{R\ 26\ 357\ 500.00}{R\ 7\ 993\ 620.53}$$

$$= 3.3\ years$$

A payback period of 3.3 years is lucrative for organisations that are stable and sustainable. It is inevitable that almost all organisations with at least 10 years in operation and looking to expand or stay in business for many more years would favour the heat recovery initiative when considering the payback period.

The NPV calculation would be based on the required return of 8% and the 4 years period of the project span. Using Equation 8.2 above, the results are plotted in the table below:

Required Return	8%			
		Year	**Cash Flow**	**Present Value**
		0	-26 357 500.00	-26 357 500.00
		1	7 993 620.53	7 401 500.49
		2	7 993 620.53	6 853 241.20
		3	7 993 620.53	6 345 593.70
		4	7 993 620.53	5 875 549.72
NET PRESENT VALUE				118 385.11

Table 8.4: The NPV calculation of the project

A positive NPV indicates a net gain, and thus a project would most likely be accepted in the economics point of view. Positive NPV indicates a net gain, projects with higher NPV are usually given preference where multiple projects are considered, however, in the case of only one project, a positive NPV gives sufficient information on whether to adopt a project or not.

The ARR

$$ARR = \frac{Average\ Profit}{Average\ Investment}$$

$$= \frac{R\ 7\ 993\ 620.53}{R\ 26\ 357\ 500.00}$$

$$= 0.3$$

An ARR does not give a complete viewpoint of choosing to do the project if it is not compared with another; however, it gives an overview of the average return to be expected for the project.

8.7 Conclusion

The profitability analysis carried out for this project revealed that the project is economically viable. The total capital investment for the project was estimated using data from older similar projects and industry base. Variations in estimations were catered for, and the correlation analysis was used. The payback period, NPV and ARR calculations were well within the norm for industrial projects. It is hypothesized that the price of energy is bound to increase with the lapse of years, and energy recovery will increase. Cost reduction is a vital consideration for manufacturing businesses, and thus, the economic viability of the heat recovery initiatives is supported by both the current and future trends of manufacturing organisations.

The next chapter presents recommendations based on the findings, proposes the scope of work that can be applied in the future research and concludes the entire study.

CHAPTER NINE – CONCLUSIONS

9.1 Introduction

This chapter summarizes the findings from the study, concludes the study based on the findings from the research. The extent to which the study was able to address the research objectives and questions of the study is presented. An evaluation of the link between the reviewed literature and the research findings is provided to maintain the accuracy of the supporting viewpoint to exhaust heat recovery initiatives. Areas of future research that relate to this study are identified and proposed, and a conclusion to the entire study is presented.

9.2 Findings from the Literature Review

An extensive review of the literature was conducted in order to investigate knowledge-base of energy recovery worldwide. A summary of the findings is presented below.

- Increasing world population and industrialization are the two major factors that drive energy demand globally (Aneke, 2012).
- Fossil fuels are the most widely used source of energy due to their level of commercialization (Anderson, 2014). Fossil fuels have, however, detrimental effects on the environment.
- The greatest amount of energy related to greenhouse gases emission is as a result of fossil fuel emissions worldwide.
- There is greater potential for exhaust energy recovery in food and beverages industries; due to their large number in comparison to other industries.
- Roughly 33% of the energy consumed by industries is discharged as thermal losses directly to the atmosphere; and these discharges are the result of process inefficiencies and inability to recover and use waste heat (William, 2015).
- Use of waste heat supplements energy demand and improves the overall efficiency of the system.
- Energy lost in waste gases cannot be fully recovered (Bai & Xu, 2011).

- There are many ways to recover waste heat from the process; and the cascade system of recovery is considered the best practice to ensure the maximum amount of recovery (Smith, 2005).

- Process modification and heat integration are vital in ensuring that the scope of heat recovery is improved.

- The application of pinch technology has resulted in significant improvements in the energy efficiency of industrial facilities globally.

- There are several interrelated barriers that impede the installations of waste heat recovery systems in process organisations.

- Energy assessments are useful in identifying where and how much energy is consumed by an existing process (Fuchs, 2012).

- In process industries, there are potentials for improving energy efficiency through electrical utilities, reticulation improvements, insulation and process control.

- Heat transfer has been applied in various research fields mainly to increase the heat transfer rate, decrease heat transfer rate and keep the temperature in a certain range (Nave, 2015). In industrial applications, the convection method of heat transfer is dominant for heating or cooling of substances (Riffat, 2016).

- Steam globally dominates as the heat transfer fluid in process plants, and steam systems that are designed and maintained for efficiency yield a significant monetary benefit to process plants.

- A possible effective way of improving the heat transfer performance of common fluids is to suspend various types of small solid particles, such as metallic and non-metallic particles, in conventional fluids, such as water and mineral oils to form a colloidal solution (Yang, et al., 2013).

- Energy losses in spray-drying occur at several places, but most energy is lost with the exhaust drying air (Jansen, et al., 1981).

- According to (Kemp, 2012) typical milk powder spray dryers are not integrated whatsoever due to three main factors: economics, particle loading and fouling and relatively low exhaust temperatures.

9.3 Findings from the Primary Research

Findings from the primary research ware presented in relation to research questions which this study strived to answer.

- Major energy losses to the atmosphere in the powder spray-drying process occur in both the air heater and drying chamber. Approximately 5 200 MJ of energy exits to the atmosphere on an hourly basis.

- Some energy losses occur in reticulation of hot air to the drying chamber, and these losses can be controlled by the insulation material used and length of pipe used for reticulation.

- Nave (2015) states that gaseous heat transfer fluids, have low heat transfer coefficients and low densities. The heat transfer coefficient of air was determined and found very low even at excessive temperatures.

- The conductive thermal resistance of the insulated pipe was found to be higher than the convective thermal resistance of it; therefore, more energy is lost through convection as opposed to conduction.

- Air at a higher temperature has the higher moisture content than air at a lower temperature, provided the relative humidity is the same and the temperatures fall within the psychrometric chart base.

- The specific heat capacity of humid air is directly proportional to the moisture content in the air, and high moisture containing air has a higher specific heat capacity in comparison to a low moisture containing air.

- The amount of water removed in the primary dryer is about three times higher than the amount of water removed in the secondary dryer. As such, the energy lost in the primary dryer is approximately three times higher as well.

- The direct measurement of steam consumption is considered as accurate as the calculation method. The overall steam energy consumption is in excess of 15 000 MJ for each hour of operation in the process.

- In a powder spray-drying process, different areas of the process consume steam according to their needs; and process water supply and air handling units are the biggest users of steam in the entire powder spray-drying process.

- There is a wide range of technologies available for low-grade waste heat recovery in process plants. The selection of the most suitable technology is dependent on the process plant, cost, technical viability and other factors.

- In most industries, heat transfer occurs in heat exchangers; the selection of which differs depending on the requirements and applications. The exhaust recovery in the spray-drying process can be done by heat exchangers.

- A decision tree is useful in selecting the appropriate method of heat transfer technology.

- Quantitative and qualitative input data of both the heat source and the heat sink is crucial in the development of user-defined specifications for the decision criteria in the decision tree method.

- The initial step in the selection of the heat transfer technology for the heat recovery system is fairly easy, as it relates to selecting the category of heat transfer which could either be: heat exchangers, heat pumps or ORC for power generation.

- Condensing economisers are vital in exhaust heat recovery systems as they recover heat and also prevent air pollution in residual exhaust being emitted.

- The challenge with the low-grade heat from the exhausts is that it does not match the requirements of the process heat sinks, due to inadequate temperature approach, and thus cannot substitute steam.

- Redesigning the system for optimization of energy demand and production capacity can be employed to optimize the spry-drying process. The impact of reducing energy losses is seen in that the recovered energy can be transferred to process to increase the production capacity of the process.

- The most simple and economical way to use the recovered heat is pre-heating of the fresh drying air (Goula & Adamopoulos, 2005). Research calculations showed that greater results could be achieved by using exhaust heat from the spray-drying process to pre-heat incoming air.
- The capital cost of adopting exhaust heat recovery is high. However, the return on investment renders the investment lucrative. Lower payback period and positive NPV add motivation to energy recovery initiatives as it is expected that the price of energy will increase.

9.4 Concluding Remarks

The study aimed at assessing and evaluating the benefit of recovering the heat from the exhaust of the powder spray-drying process with the hypothesis that recovered heat energy can be used to optimize production and energy demand of the entire process. The study extended from the literature on exhaust energy recovery and made use of the primary research to answer the research questions in the study. The objectives of the study were to determine the energy lost to exhaust, evaluate the overall steam energy consumption, selection of appropriate heat transfer method, examination of the potential of optimization and evaluating the cost of exhaust heat recovery.

The findings from the primary research concluded that there is a significant amount of energy lost to exhaust which is recoverable. From both primary and secondary research, challenges of matching the recovered heat sink to a heat source in the process surfaced. Primary research confirmed that the best area of recovered exhaust energy usage is the pre-heating of incoming air. The study also highlighted the importance of well-designed, energy-oriented systems to achieve the desired results.

9.5 Areas for Further Research

The study was limited in that it only focused on heat recovery in milk powder spray-drying process only. Future work should focus on providing effective and reliable cost estimation for heat recovery equipment. The ability to simulate any process system and determine the heat loss, the effective method of heat recovery, the potential for optimization and the cost involved can also be investigated for each process group (steel manufacturing, spray-drying, sugar milling, paper milling, etc.) to easily evaluate the potential and benefit of energy recovery.

9.6 Conclusion

This chapter concluded the study and suggested areas of further research. Achieving the objectives and answering the research questions fulfilled the overall aim of the study. The study utilized critical data from theoretical, practical and mathematical base to determine the energy lost to exhaust in the powder spray-drying process. Evaluation of the overall steam energy consumption was performed, and a theoretical base was employed to select the appropriate heat transfer method. Production capacity and energy demand optimization were done, and the cost associated with exhaust heat recovery estimated.

In the competitive and energy demanding environment, organisations rely on the efficiencies of their systems to cut down as much as possible cost losses that occur due to inefficiencies. This makes an investment in heat recovery an important aspect for any competing organisation. Factors surrounding the recovery of exhaust heat must, therefore, be investigated to ensure profitable investments are made.

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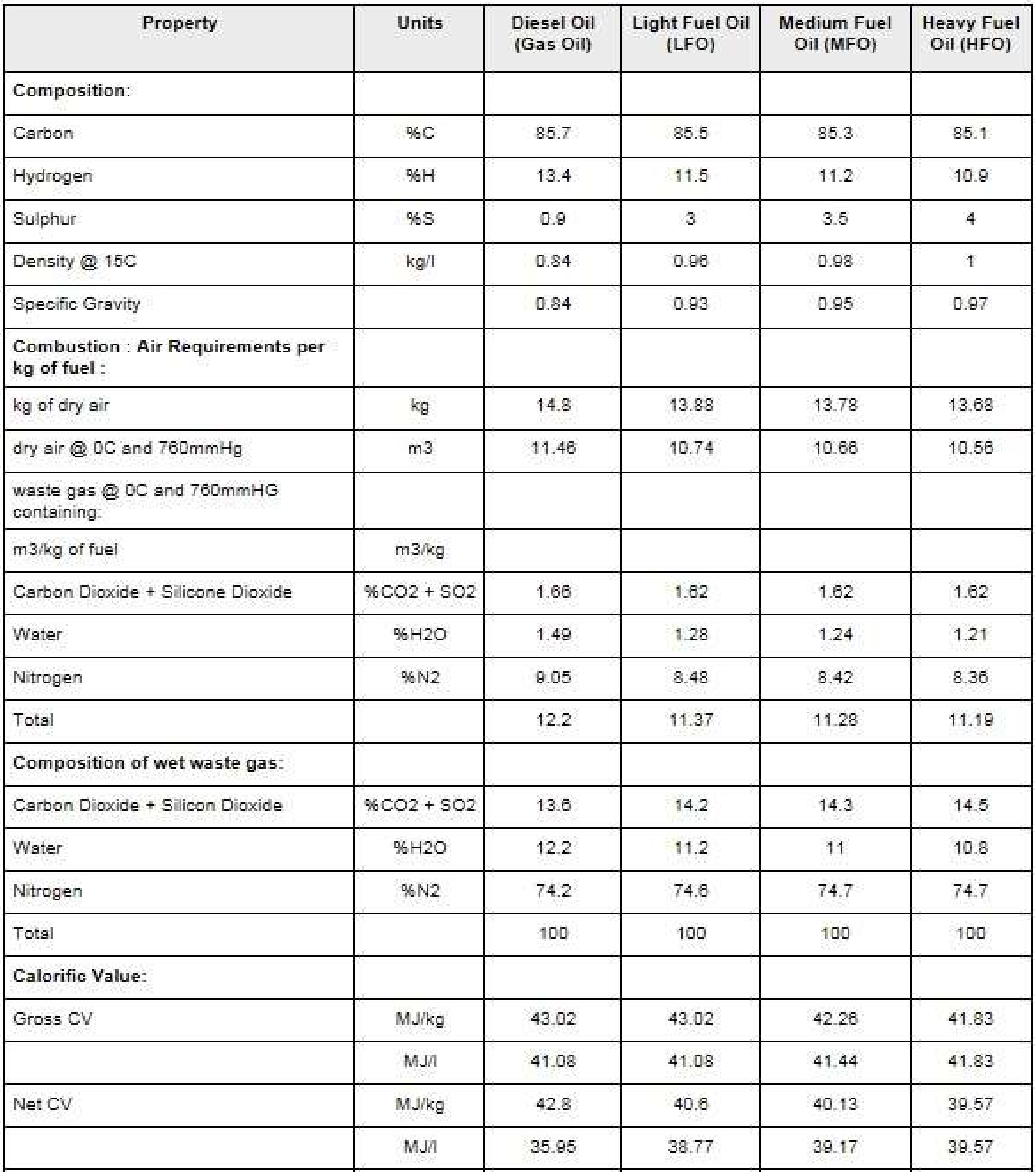
APPENDICES

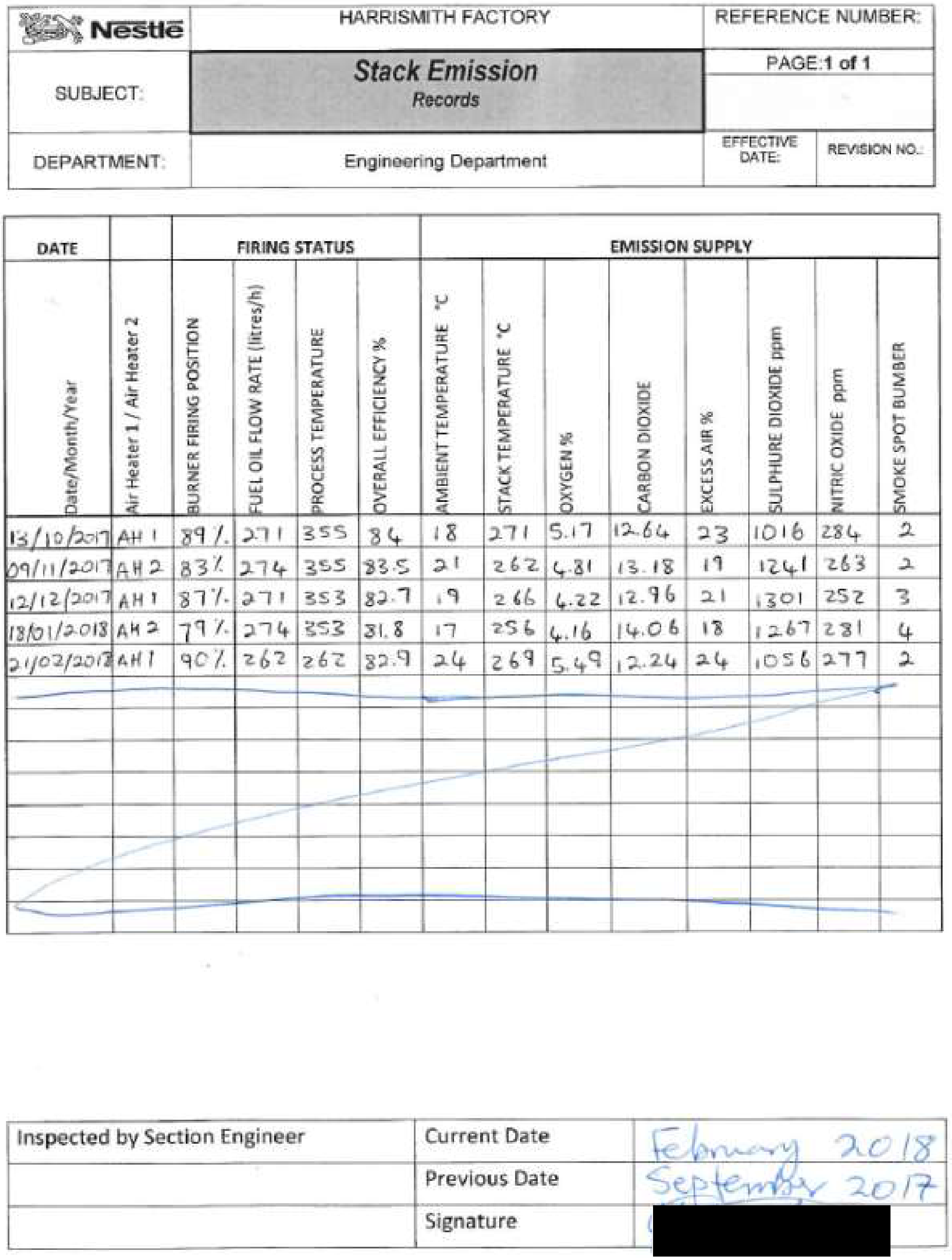
Appendix A: Specifications for fuel oils.



Property	Units	Diesel Oil (Gas Oil)	Light Fuel Oil (LFO)	Medium Fuel Oil (MFO)	Heavy Fuel Oil (HFO)
Composition:					
Carbon	%C	85.7	85.5	85.3	85.1
Hydrogen	%H	13.4	11.5	11.2	10.9
Sulphur	%S	0.9	3	3.5	4
Density @ 15C	kg/l	0.84	0.96	0.98	1
Specific Gravity		0.84	0.93	0.95	0.97
Combustion : Air Requirements per kg of fuel :					
kg of dry air	kg	14.8	13.88	13.78	13.68
dry air @ 0C and 760mmHg	m3	11.46	10.74	10.66	10.56
waste gas @ 0C and 760mmHG containing:					
m3/kg of fuel	m3/kg				
Carbon Dioxide + Silicone Dioxide	%CO2 + SO2	1.66	1.62	1.62	1.62
Water	%H2O	1.49	1.28	1.24	1.21
Nitrogen	%N2	9.05	8.48	8.42	8.36
Total		12.2	11.37	11.28	11.19
Composition of wet waste gas:					
Carbon Dioxide + Silicon Dioxide	%CO2 + SO2	13.6	14.2	14.3	14.5
Water	%H2O	12.2	11.2	11	10.8
Nitrogen	%N2	74.2	74.6	74.7	74.7
Total		100	100	100	100
Calorific Value:					
Gross CV	MJ/kg	43.02	43.02	42.26	41.83
	MJ/l	41.08	41.08	41.44	41.83
Net CV	MJ/kg	42.8	40.6	40.13	39.57
	MJ/l	35.95	38.77	39.17	39.57

Table A.1: Specifications of oil used for combustion (Global Combustion Systems)

Appendix B: Air heater performance management

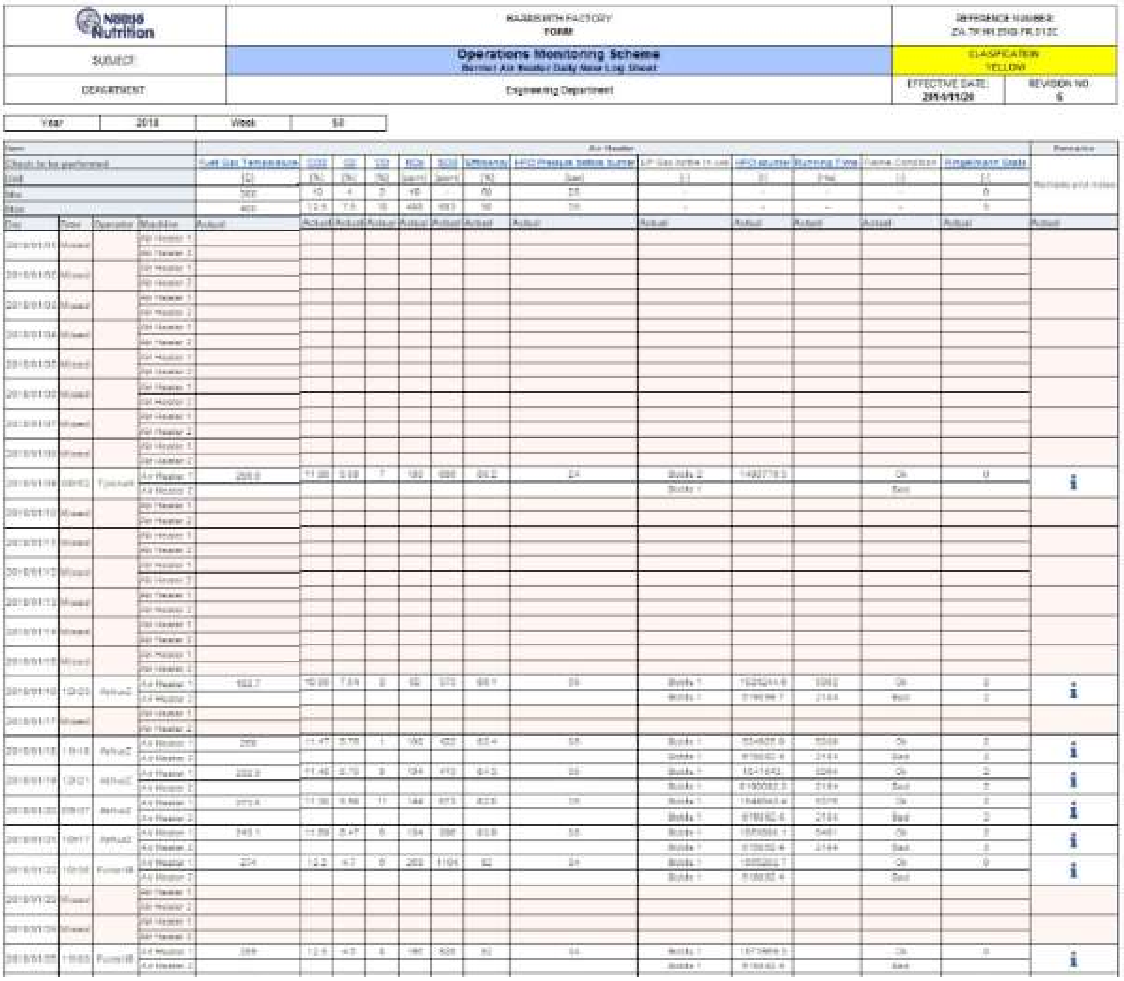


Nestlé	HARRISMITH FACTORY	REFERENCE NUMBER:
SUBJECT:	**Stack Emission** **Records**	PAGE:**1 of 1**
DEPARTMENT:	Engineering Department	EFFECTIVE DATE: / REVISION NO.:

DATE		FIRING STATUS				EMISSION SUPPLY							
Date/Month/Year	Air Heater 1 / Air Heater 2	BURNER FIRING POSITION	FUEL OIL FLOW RATE (litres/h)	PROCESS TEMPERATURE	OVERALL EFFICIENCY %	AMBIENT TEMPERATURE °C	STACK TEMPERATURE °C	OXYGEN %	CARBON DIOXIDE	EXCESS AIR %	SULPHURE DIOXIDE ppm	NITRIC OXIDE ppm	SMOKE SPOT BUMBER
13/10/2017	AH 1	89%	271	355	84	18	271	5.17	12.64	23	1016	284	2
09/11/2017	AH 2	83%	274	355	83.5	21	262	4.81	13.18	19	1241	263	2
12/12/2017	AH 1	87%	271	353	82.7	19	266	4.22	12.96	21	1301	252	3
18/01/2018	AH 2	79%	274	353	81.8	17	256	4.16	14.06	18	1267	281	4
21/02/2018	AH 1	90%	262	262	82.9	24	269	5.49	12.24	24	1056	277	2

Inspected by Section Engineer	Current Date	February 2018
	Previous Date	September 2017
	Signature	[redacted]

Table B.2: Emissions test results (Nestle)

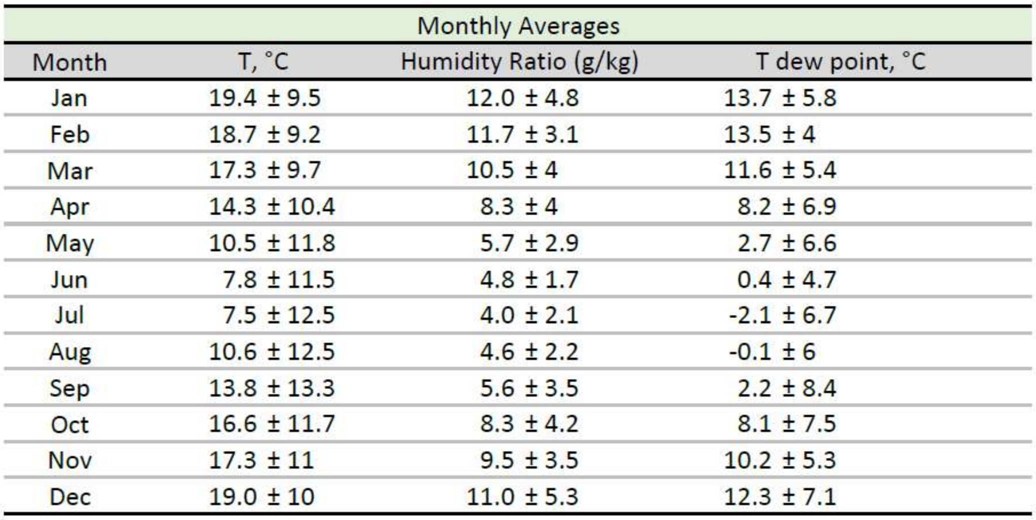


Operations Monitoring Scheme

Engineering Department

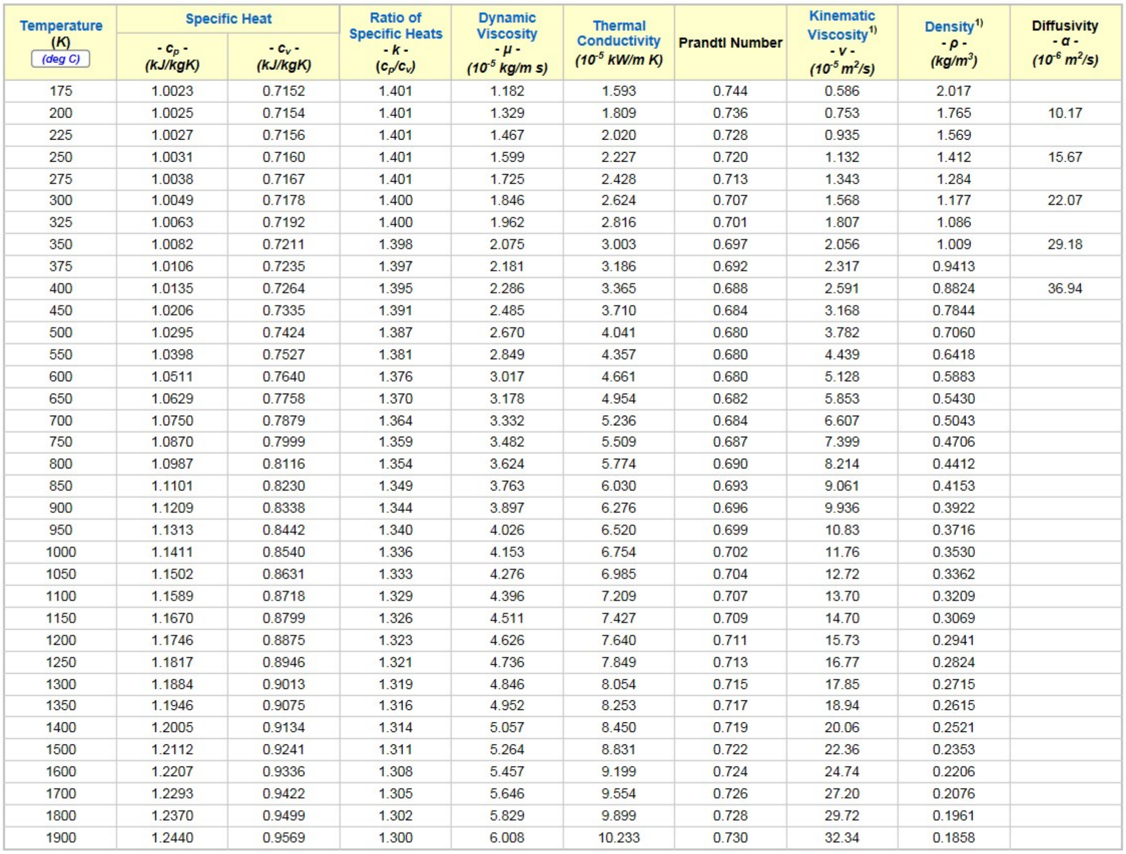
Table B.3: Daily log for air heaters (Nestle)

Appendix C: Air properties and specifications



Monthly Averages			
Month	T, °C	Humidity Ratio (g/kg)	T dew point, °C
Jan	19.4 ± 9.5	12.0 ± 4.8	13.7 ± 5.8
Feb	18.7 ± 9.2	11.7 ± 3.1	13.5 ± 4
Mar	17.3 ± 9.7	10.5 ± 4	11.6 ± 5.4
Apr	14.3 ± 10.4	8.3 ± 4	8.2 ± 6.9
May	10.5 ± 11.8	5.7 ± 2.9	2.7 ± 6.6
Jun	7.8 ± 11.5	4.8 ± 1.7	0.4 ± 4.7
Jul	7.5 ± 12.5	4.0 ± 2.1	-2.1 ± 6.7
Aug	10.6 ± 12.5	4.6 ± 2.2	-0.1 ± 6
Sep	13.8 ± 13.3	5.6 ± 3.5	2.2 ± 8.4
Oct	16.6 ± 11.7	8.3 ± 4.2	8.1 ± 7.5
Nov	17.3 ± 11	9.5 ± 3.5	10.2 ± 5.3
Dec	19.0 ± 10	11.0 ± 5.3	12.3 ± 7.1

Table C.4: Temperature and humidity ratio for Harrismith area (Nestle)



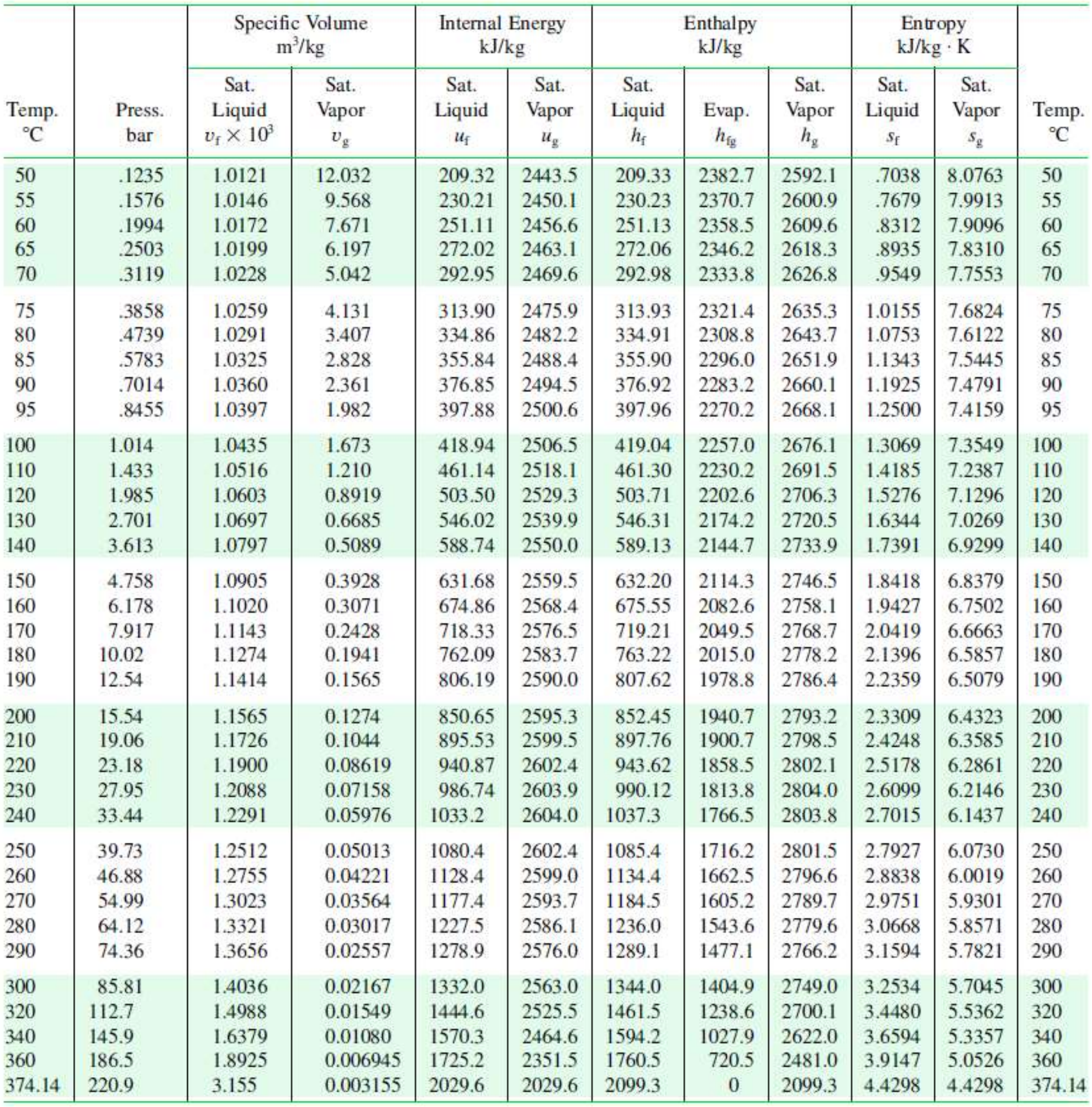
Temperature (K) (deg C)	Specific Heat		Ratio of Specific Heats - k - (c_p/c_v)	Dynamic Viscosity - μ - (10^{-5} kg/m s)	Thermal Conductivity (10^{-5} kW/m K)	Prandtl Number	Kinematic Viscosity[1)] - ν - (10^{-5} m^2/s)	Density[1)] - ρ - (kg/m^3)	Diffusivity - α - (10^{-6} m^2/s)
	- c_p - (kJ/kgK)	- c_v - (kJ/kgK)							
175	1.0023	0.7152	1.401	1.182	1.593	0.744	0.586	2.017	
200	1.0025	0.7154	1.401	1.329	1.809	0.736	0.753	1.765	10.17
225	1.0027	0.7156	1.401	1.467	2.020	0.728	0.935	1.569	
250	1.0031	0.7160	1.401	1.599	2.227	0.720	1.132	1.412	15.67
275	1.0038	0.7167	1.401	1.725	2.428	0.713	1.343	1.284	
300	1.0049	0.7178	1.400	1.846	2.624	0.707	1.568	1.177	22.07
325	1.0063	0.7192	1.400	1.962	2.816	0.701	1.807	1.086	
350	1.0082	0.7211	1.398	2.075	3.003	0.697	2.056	1.009	29.18
375	1.0106	0.7235	1.397	2.181	3.186	0.692	2.317	0.9413	
400	1.0135	0.7264	1.395	2.286	3.365	0.688	2.591	0.8824	36.94
450	1.0206	0.7335	1.391	2.485	3.710	0.684	3.168	0.7844	
500	1.0295	0.7424	1.387	2.670	4.041	0.680	3.782	0.7060	
550	1.0398	0.7527	1.381	2.849	4.357	0.680	4.439	0.6418	
600	1.0511	0.7640	1.376	3.017	4.661	0.680	5.128	0.5883	
650	1.0629	0.7758	1.370	3.178	4.954	0.682	5.853	0.5430	
700	1.0750	0.7879	1.364	3.332	5.236	0.684	6.607	0.5043	
750	1.0870	0.7999	1.359	3.482	5.509	0.687	7.399	0.4706	
800	1.0987	0.8116	1.354	3.624	5.774	0.690	8.214	0.4412	
850	1.1101	0.8230	1.349	3.763	6.030	0.693	9.061	0.4153	
900	1.1209	0.8338	1.344	3.897	6.276	0.696	9.936	0.3922	
950	1.1313	0.8442	1.340	4.026	6.520	0.699	10.83	0.3716	
1000	1.1411	0.8540	1.336	4.153	6.754	0.702	11.76	0.3530	
1050	1.1502	0.8631	1.333	4.276	6.985	0.704	12.72	0.3362	
1100	1.1589	0.8718	1.329	4.396	7.209	0.707	13.70	0.3209	
1150	1.1670	0.8799	1.326	4.511	7.427	0.709	14.70	0.3069	
1200	1.1746	0.8875	1.323	4.626	7.640	0.711	15.73	0.2941	
1250	1.1817	0.8946	1.321	4.736	7.849	0.713	16.77	0.2824	
1300	1.1884	0.9013	1.319	4.846	8.054	0.715	17.85	0.2715	
1350	1.1946	0.9075	1.316	4.952	8.253	0.717	18.94	0.2615	
1400	1.2005	0.9134	1.314	5.057	8.450	0.719	20.06	0.2521	
1500	1.2112	0.9241	1.311	5.264	8.831	0.722	22.36	0.2353	
1600	1.2207	0.9336	1.308	5.457	9.199	0.724	24.74	0.2206	
1700	1.2293	0.9422	1.305	5.646	9.554	0.726	27.20	0.2076	
1800	1.2370	0.9499	1.302	5.829	9.899	0.728	29.72	0.1961	
1900	1.2440	0.9569	1.300	6.008	10.233	0.730	32.34	0.1858	

Table C.5: Properties of air at constant pressure and varying temperatures (Engineering Toolbox)

Figure C.1: Relative humidity and temperature of the drying building Relative humidity of the treated air.

Temp. °C	Press. bar	Specific Volume m³/kg		Internal Energy kJ/kg		Enthalpy kJ/kg			Entropy kJ/kg · K		Temp. °C
		Sat. Liquid $v_f \times 10^3$	Sat. Vapor v_g	Sat. Liquid u_f	Sat. Vapor u_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Sat. Vapor s_g	
.01	0.00611	1.0002	206.136	0.00	2375.3	0.01	2501.3	2501.4	0.0000	9.1562	.01
4	0.00813	1.0001	157.232	16.77	2380.9	16.78	2491.9	2508.7	0.0610	9.0514	4
5	0.00872	1.0001	147.120	20.97	2382.3	20.98	2489.6	2510.6	0.0761	9.0257	5
6	0.00935	1.0001	137.734	25.19	2383.6	25.20	2487.2	2512.4	0.0912	9.0003	6
8	0.01072	1.0002	120.917	33.59	2386.4	33.60	2482.5	2516.1	0.1212	8.9501	8
10	0.01228	1.0004	106.379	42.00	2389.2	42.01	2477.7	2519.8	0.1510	8.9008	10
11	0.01312	1.0004	99.857	46.20	2390.5	46.20	2475.4	2521.6	0.1658	8.8765	11
12	0.01402	1.0005	93.784	50.41	2391.9	50.41	2473.0	2523.4	0.1806	8.8524	12
13	0.01497	1.0007	88.124	54.60	2393.3	54.60	2470.7	2525.3	0.1953	8.8285	13
14	0.01598	1.0008	82.848	58.79	2394.7	58.80	2468.3	2527.1	0.2099	8.8048	14
15	0.01705	1.0009	77.926	62.99	2396.1	62.99	2465.9	2528.9	0.2245	8.7814	15
16	0.01818	1.0011	73.333	67.18	2397.4	67.19	2463.6	2530.8	0.2390	8.7582	16
17	0.01938	1.0012	69.044	71.38	2398.8	71.38	2461.2	2532.6	0.2535	8.7351	17
18	0.02064	1.0014	65.038	75.57	2400.2	75.58	2458.8	2534.4	0.2679	8.7123	18
19	0.02198	1.0016	61.293	79.76	2401.6	79.77	2456.5	2536.2	0.2823	8.6897	19
20	0.02339	1.0018	57.791	83.95	2402.9	83.96	2454.1	2538.1	0.2966	8.6672	20
21	0.02487	1.0020	54.514	88.14	2404.3	88.14	2451.8	2539.9	0.3109	8.6450	21
22	0.02645	1.0022	51.447	92.32	2405.7	92.33	2449.4	2541.7	0.3251	8.6229	22
23	0.02810	1.0024	48.574	96.51	2407.0	96.52	2447.0	2543.5	0.3393	8.6011	23
24	0.02985	1.0027	45.883	100.70	2408.4	100.70	2444.7	2545.4	0.3534	8.5794	24
25	0.03169	1.0029	43.360	104.88	2409.8	104.89	2442.3	2547.2	0.3674	8.5580	25
26	0.03363	1.0032	40.994	109.06	2411.1	109.07	2439.9	2549.0	0.3814	8.5367	26
27	0.03567	1.0035	38.774	113.25	2412.5	113.25	2437.6	2550.8	0.3954	8.5156	27
28	0.03782	1.0037	36.690	117.42	2413.9	117.43	2435.2	2552.6	0.4093	8.4946	28
29	0.04008	1.0040	34.733	121.60	2415.2	121.61	2432.8	2554.5	0.4231	8.4739	29
30	0.04246	1.0043	32.894	125.78	2416.6	125.79	2430.5	2556.3	0.4369	8.4533	30
31	0.04496	1.0046	31.165	129.96	2418.0	129.97	2428.1	2558.1	0.4507	8.4329	31
32	0.04759	1.0050	29.540	134.14	2419.3	134.15	2425.7	2559.9	0.4644	8.4127	32
33	0.05034	1.0053	28.011	138.32	2420.7	138.33	2423.4	2561.7	0.4781	8.3927	33
34	0.05324	1.0056	26.571	142.50	2422.0	142.50	2421.0	2563.5	0.4917	8.3728	34
35	0.05628	1.0060	25.216	146.67	2423.4	146.68	2418.6	2565.3	0.5053	8.3531	35
36	0.05947	1.0063	23.940	150.85	2424.7	150.86	2416.2	2567.1	0.5188	8.3336	36
38	0.06632	1.0071	21.602	159.20	2427.4	159.21	2411.5	2570.7	0.5458	8.2950	38
40	0.07384	1.0078	19.523	167.56	2430.1	167.57	2406.7	2574.3	0.5725	8.2570	40
45	0.09593	1.0099	15.258	188.44	2436.8	188.45	2394.8	2583.2	0.6387	8.1648	45

Table C.6: Water properties of liquid and saturated vapour at lower temperatures (Keenan & Keyes, 1978).



		Specific Volume m³/kg		Internal Energy kJ/kg		Enthalpy kJ/kg			Entropy kJ/kg · K		
Temp. °C	Press. bar	Sat. Liquid $v_f \times 10^3$	Sat. Vapor v_g	Sat. Liquid u_f	Sat. Vapor u_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Sat. Vapor s_g	Temp. °C
50	.1235	1.0121	12.032	209.32	2443.5	209.33	2382.7	2592.1	.7038	8.0763	50
55	.1576	1.0146	9.568	230.21	2450.1	230.23	2370.7	2600.9	.7679	7.9913	55
60	.1994	1.0172	7.671	251.11	2456.6	251.13	2358.5	2609.6	.8312	7.9096	60
65	.2503	1.0199	6.197	272.02	2463.1	272.06	2346.2	2618.3	.8935	7.8310	65
70	.3119	1.0228	5.042	292.95	2469.6	292.98	2333.8	2626.8	.9549	7.7553	70
75	.3858	1.0259	4.131	313.90	2475.9	313.93	2321.4	2635.3	1.0155	7.6824	75
80	.4739	1.0291	3.407	334.86	2482.2	334.91	2308.8	2643.7	1.0753	7.6122	80
85	.5783	1.0325	2.828	355.84	2488.4	355.90	2296.0	2651.9	1.1343	7.5445	85
90	.7014	1.0360	2.361	376.85	2494.5	376.92	2283.2	2660.1	1.1925	7.4791	90
95	.8455	1.0397	1.982	397.88	2500.6	397.96	2270.2	2668.1	1.2500	7.4159	95
100	1.014	1.0435	1.673	418.94	2506.5	419.04	2257.0	2676.1	1.3069	7.3549	100
110	1.433	1.0516	1.210	461.14	2518.1	461.30	2230.2	2691.5	1.4185	7.2387	110
120	1.985	1.0603	0.8919	503.50	2529.3	503.71	2202.6	2706.3	1.5276	7.1296	120
130	2.701	1.0697	0.6685	546.02	2539.9	546.31	2174.2	2720.5	1.6344	7.0269	130
140	3.613	1.0797	0.5089	588.74	2550.0	589.13	2144.7	2733.9	1.7391	6.9299	140
150	4.758	1.0905	0.3928	631.68	2559.5	632.20	2114.3	2746.5	1.8418	6.8379	150
160	6.178	1.1020	0.3071	674.86	2568.4	675.55	2082.6	2758.1	1.9427	6.7502	160
170	7.917	1.1143	0.2428	718.33	2576.5	719.21	2049.5	2768.7	2.0419	6.6663	170
180	10.02	1.1274	0.1941	762.09	2583.7	763.22	2015.0	2778.2	2.1396	6.5857	180
190	12.54	1.1414	0.1565	806.19	2590.0	807.62	1978.8	2786.4	2.2359	6.5079	190
200	15.54	1.1565	0.1274	850.65	2595.3	852.45	1940.7	2793.2	2.3309	6.4323	200
210	19.06	1.1726	0.1044	895.53	2599.5	897.76	1900.7	2798.5	2.4248	6.3585	210
220	23.18	1.1900	0.08619	940.87	2602.4	943.62	1858.5	2802.1	2.5178	6.2861	220
230	27.95	1.2088	0.07158	986.74	2603.9	990.12	1813.8	2804.0	2.6099	6.2146	230
240	33.44	1.2291	0.05976	1033.2	2604.0	1037.3	1766.5	2803.8	2.7015	6.1437	240
250	39.73	1.2512	0.05013	1080.4	2602.4	1085.4	1716.2	2801.5	2.7927	6.0730	250
260	46.88	1.2755	0.04221	1128.4	2599.0	1134.4	1662.5	2796.6	2.8838	6.0019	260
270	54.99	1.3023	0.03564	1177.4	2593.7	1184.5	1605.2	2789.7	2.9751	5.9301	270
280	64.12	1.3321	0.03017	1227.5	2586.1	1236.0	1543.6	2779.6	3.0668	5.8571	280
290	74.36	1.3656	0.02557	1278.9	2576.0	1289.1	1477.1	2766.2	3.1594	5.7821	290
300	85.81	1.4036	0.02167	1332.0	2563.0	1344.0	1404.9	2749.0	3.2534	5.7045	300
320	112.7	1.4988	0.01549	1444.6	2525.5	1461.5	1238.6	2700.1	3.4480	5.5362	320
340	145.9	1.6379	0.01080	1570.3	2464.6	1594.2	1027.9	2622.0	3.6594	5.3357	340
360	186.5	1.8925	0.006945	1725.2	2351.5	1760.5	720.5	2481.0	3.9147	5.0526	360
374.14	220.9	3.155	0.003155	2029.6	2029.6	2099.3	0	2099.3	4.4298	4.4298	374.14

Table C.7: Water properties of liquid and saturated vapour at higher temperatures (Keenan & Keyes, 1978)

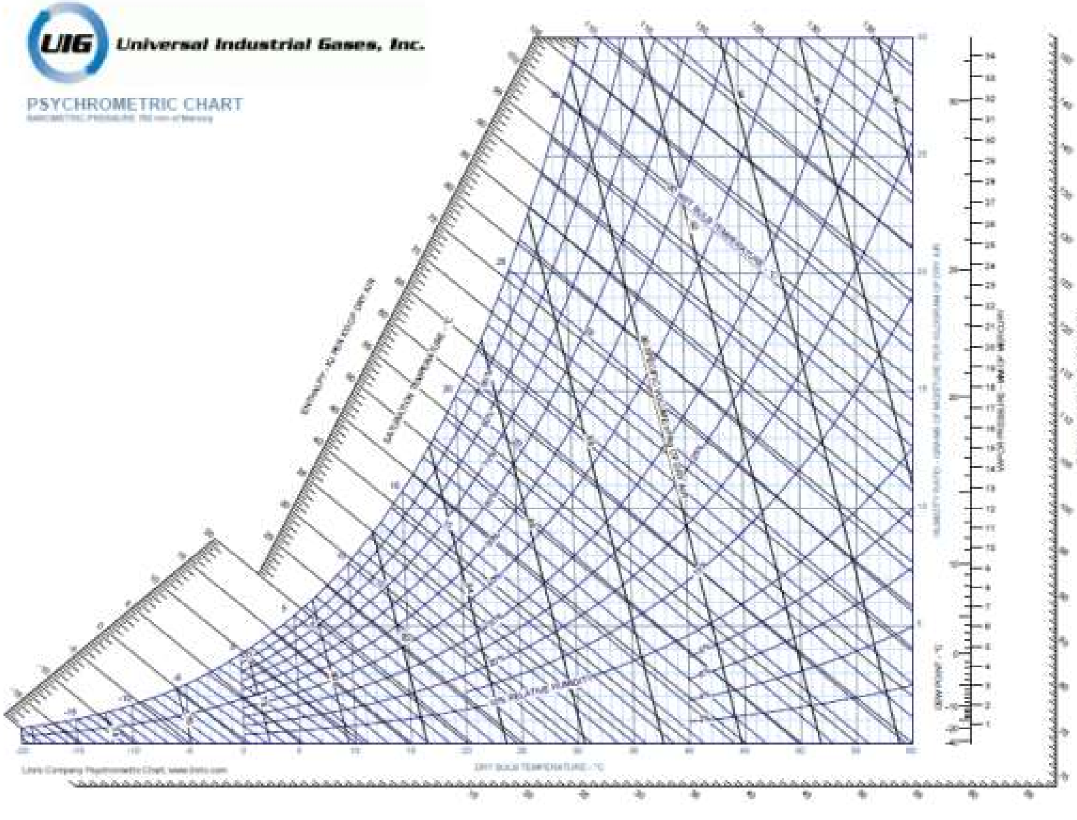


Figure C.2: Psychrometric chart (Universal Industrial Gases Inc.)

Appendix D: Hot air duct dimensions

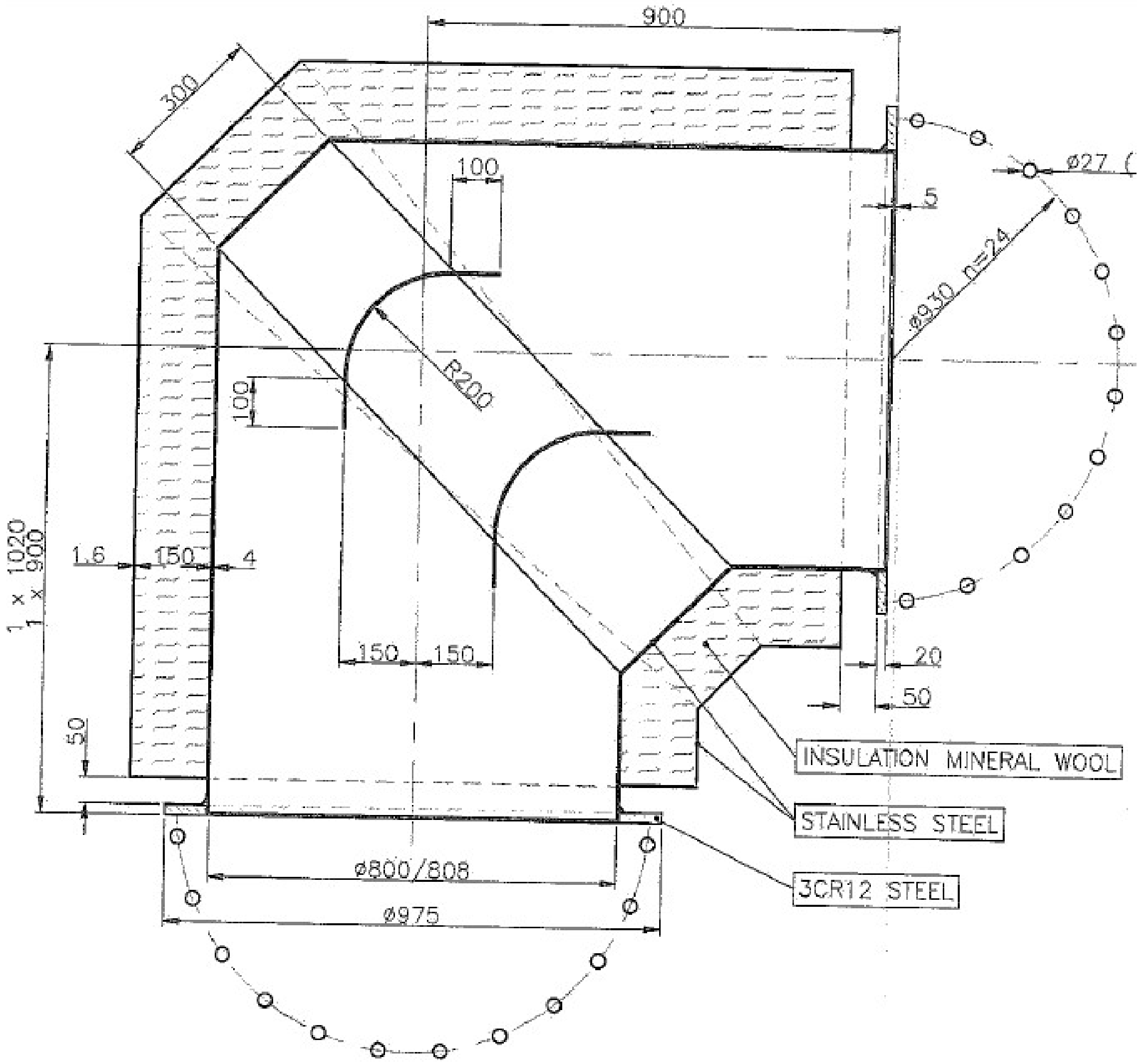
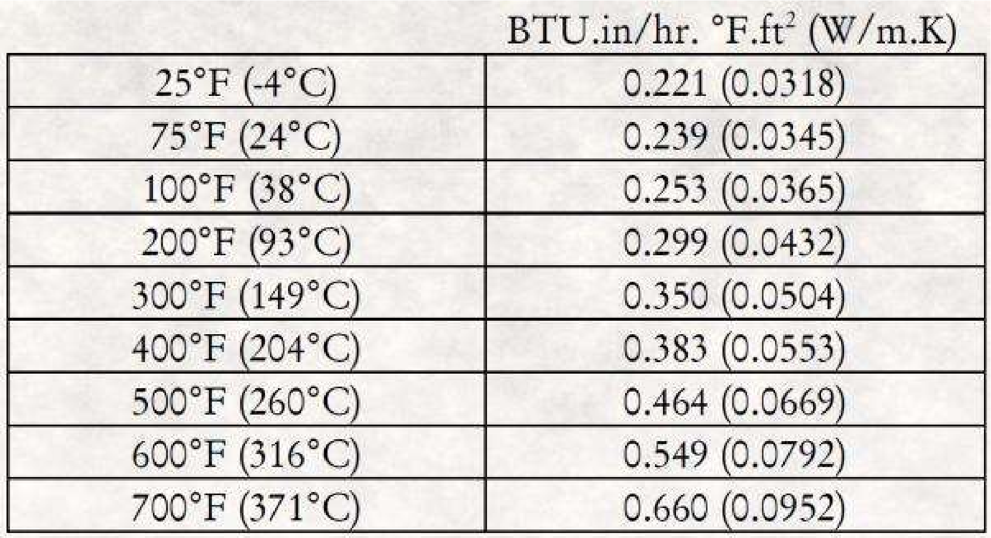


Figure D.3: Hot air duct dimensions and specifications (Nestle)

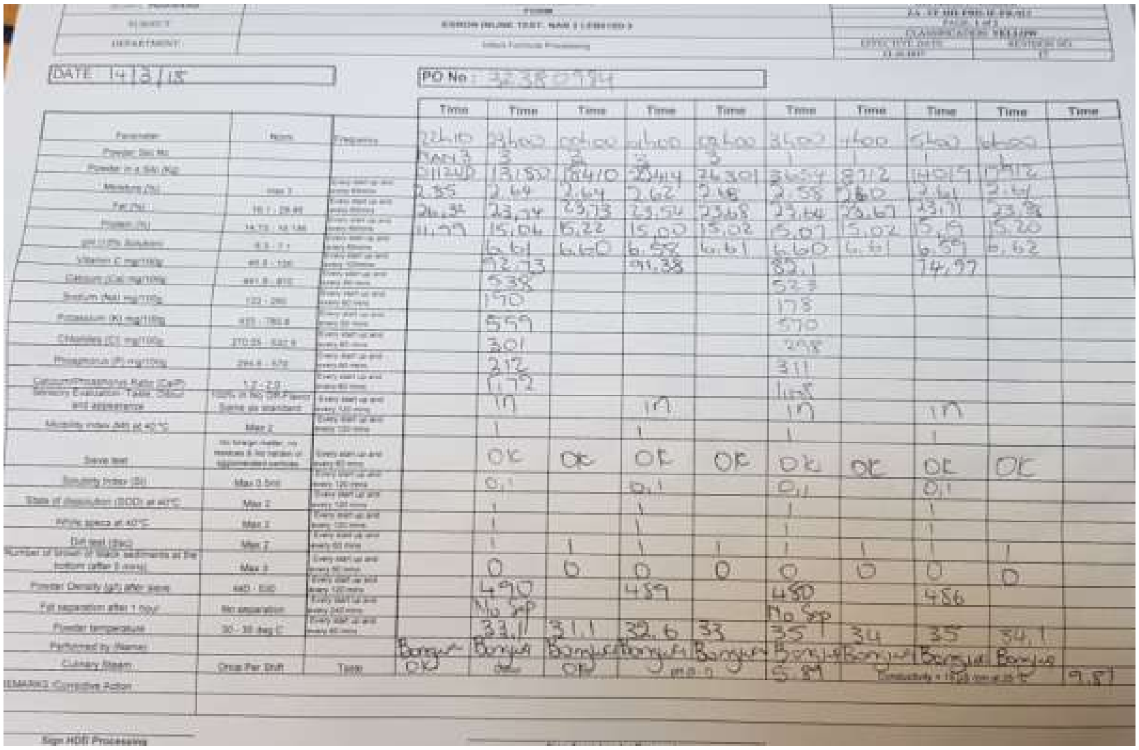
Appendix E: Hot air duct insulating material properties



	BTU.in/hr. °F.ft² (W/m.K)
25°F (-4°C)	0.221 (0.0318)
75°F (24°C)	0.239 (0.0345)
100°F (38°C)	0.253 (0.0365)
200°F (93°C)	0.299 (0.0432)
300°F (149°C)	0.350 (0.0504)
400°F (204°C)	0.383 (0.0553)
500°F (260°C)	0.464 (0.0669)
600°F (316°C)	0.549 (0.0792)
700°F (371°C)	0.660 (0.0952)

Table E.8: Thermal conductivity of mineral wool at varying temperatures (Industrial Insulation Group).

Appendix F: Powder production quality management



DATE: 14/3/18

PO No: 32380794

Parameter	Norm	Frequency	Time	Time	Time	Time	Time	Time	Time	Time	Time	Time
			22h10	23h00	00h00	01h00	02h00	03h00	04h00	05h00	06h00	
Powder Silo No			NAN3	3	3	3	3	1	1	1	1	
Powder in a Silo (Kg)			011240	13152	18410	20414	26301	3654	8712	14019	17912	
Moisture (%)	Max 3	[illegible]	2.35	2.64	2.64	2.62	2.68	2.55	2.60	2.61	2.61	
Fat (%)	[illegible]	[illegible]	26.32	23.74	23.73	23.54	23.68	23.64	23.67	23.71	23.78	
Protein (%)	[illegible]	[illegible]	11.79	15.04	15.22	15.00	15.02	15.07	15.02	15.19	15.20	
pH (10% Solution)	[illegible]	[illegible]		6.61	6.60	6.58	6.61	6.60	6.61	6.59	6.62	
Vitamin C mg/100g	[illegible]	[illegible]		92.73		91.38		82.1		74.97		
Calcium (Ca) mg/100g	[illegible]	[illegible]		538				523				
Sodium (Na) mg/100g	[illegible]	[illegible]		170				178				
Potassium (K) mg/100g	[illegible]	[illegible]		559				570				
Chlorides (Cl) mg/100g	[illegible]	[illegible]		301				298				
Phosphorus (P) mg/100g	[illegible]	[illegible]		212				311				
Calcium/Phosphorus Ratio (Ca/P)	1.2 - 2.0	[illegible]		1.72				1.68				
Sensory Evaluation: Taste, Odour and appearance	[illegible] Same as standard	[illegible]		IN		IN		IN		IN		
Mobility index (MI) at 40°C	Max 2	[illegible]		1		1		1		1		
Sieve test	[illegible]	[illegible]		OK	OK	OK	OK	OK	OK	OK	OK	
Solubility Index (SI)	Max 0.5ml	[illegible]		0.1		0.1		0.1		0.1		
State of dissolution (SOD) at 40°C	Max 2	[illegible]		1		1		1		1		
White specs at 40°C	Max 2	[illegible]		1		1		1		1		
Dirt test (disc)	Max 2	[illegible]		1	1	1	1	1	1	1	1	
Number of brown or black sediments at the bottom (after 5 mins)	Max 3	[illegible]		0	0	0	0	0	0	0	0	
Powder Density (g/l) after sieve	[illegible]	[illegible]		490		489		480		486		
Fat separation after 1 hour	No separation	[illegible]		No Sep				No Sep				
Powder temperature	30 - 35 deg C	[illegible]		33.1	31.1	32.6	33	35	34	35	34.1	
Performed by (Name)			[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	
Culinary Steam	Once Per Shift	Taste	OK		OK			5.89				9.81
REMARKS /Corrective Action												

Sign HOD Processing

Table F.9: Analysis of powder production (Nestle)

Appendix G: Drying system process configuration

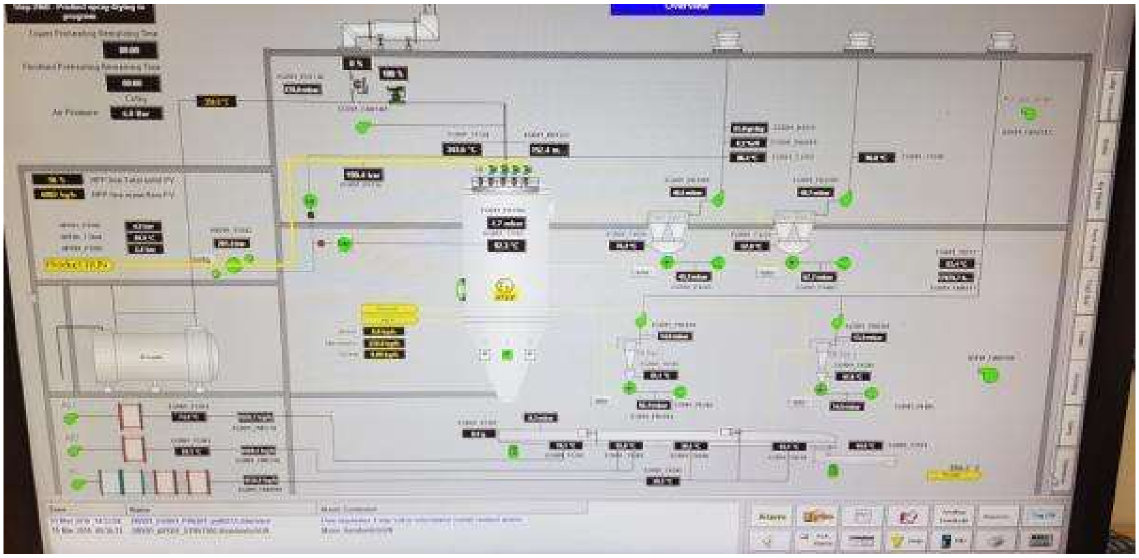


Figure G.4: Working configuration of the drying system (Nestle)

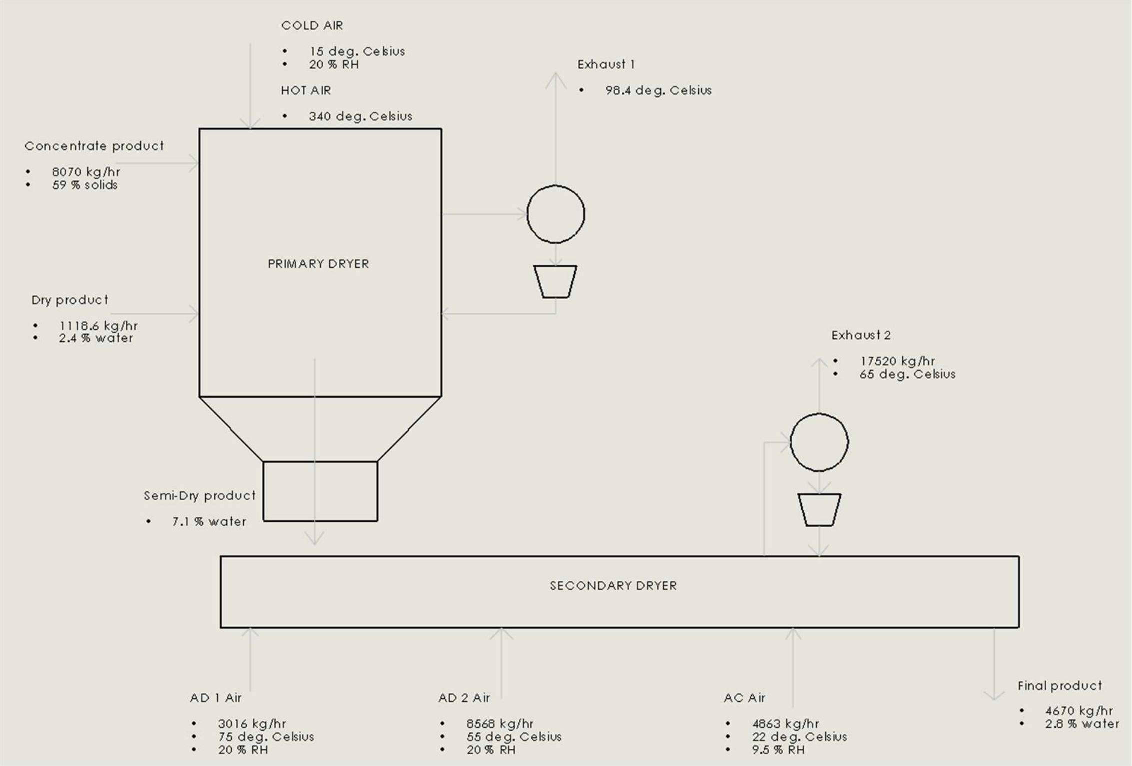


Figure G.5: Simplified working configuration of the dryer system