Optimisation of Power Plant Utility Systems using Process Integration

by

Mkhokheli Ndlovu

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Supervisor: Professor T. Majozi

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Synopsis

Within the chemical and allied process industries, the power production unit is one of the main facilities that use large volumes of water and steam as the prime utilities. These utilities are generated in boilers and cooling tower systems, respectively. Thus, it follows that any optimisation effort aimed at improving the operation and profitability of a power plant should at least focus on these utility systems. Indeed, optimisation methods for the two systems are available in the published literature. However, most of these published methods have ignored the complementing and synergistic aspect of the two utility systems as found in real power plants - and have treated their optimisation in a separate and discrete manner. Such individual analysis creates a false dichotomy between the boiler and cooling water systems and generally gives suboptimal results. The two systems constitute a comprehensive utility system and should be optimised in a single approach.

In this work, a holistic method of analysis and optimisation of the comprehensive power plant utility system is presented. Nonlinear programming (NLP) models are formulated first for each system and these are then combined to give the comprehensive utility system model. In the mathematical models, nonlinear equations are extensively used in the mass and energy balances, in calculating performance in terms of thermodynamic parameters (these include enthalpy, entropy and efficiency) and also in calculating physical properties of water, air and steam. Different performance indices are used to assess the performance of the comprehensive utility system and these include specially formulated objective functions which are in turn functions of parameters in both the cooling tower and boiler systems. In addition cost as an objective function was used for the comprehensive system, wherein the net annual profitability of the system was maximised. These objective functions have collectively shown that better system performance is obtained when the system is modelled as a single entity as opposed to individual treatment. The work further looked at the synthesis of such a power plant and gives a method of designing a power plant, providing the sizes of the boiler and cooling tower systems that are required for a given power output. The new design approach gives better results and performance when compared to traditional methods.

To my lovely daughter
Londiwe Tsenolo Ndlovu

Preface

The work contained in this thesis was performed at the University of Pretoria and was supervised by Professor Thoko Majozi.

I, Mkhokheli Ndlovu with student number 29615250, declare that:

1. I understand what plagiarism is and am aware of the University's policy in this regard.
2. I declare that this thesis is my own original work. Where other people's work has been used (either from a printed source, Internet or any other source), this has been properly acknowledged and referenced in accordance with departmental requirements.
3. I have not used work previously produced by another student or any other person to hand in as my own.
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M Ndlovu

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Chapter One

1 Introduction

Water and electricity will always be on top of the list of all the utilities found in chemical and allied process industries. In these industries, water is mainly used, apart from ordinary human consumption and its use as a cleaning agent, as an energy carrier to provide both cooling and heating requirements while electricity is the main source of energy.

Refineries and power generation plants are typical industries that use large volumes of water particularly as cooling water and for steam production. The cost of producing and delivering water suitable for these purposes has increased over the years, thus affecting the economic viability of such process plants. Electricity production has also recently received increased attention due to adverse environmental impact deriving from CO_2 and other greenhouse gas emissions. This has led to renewed interests in renewable energy sources like hydroelectric power, wind and solar energy, together with the development of clean coal technologies, which include Integrated Gasification Combined Cycle (IGCC) and Underground Coal Gasification (UCG). As a result, efforts have been put into developing systematic procedures for optimising utility usage in both grassroots and retrofit plant designs. One outcome of such developmental work was Process Integration, which emphasizes a holistic, rather than discrete treatment of a process (El-Halwagi, 1998; 2006).

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1.1 Background to the research problem

Following the pioneering work of Linnhoff & Hindmash (1983) on Pinch Analysis, a number of researchers have developed methodologies for the reduction of the process heat duty for external utilities. Such research work has, however, been focused on the individual components of a plant utility system, i.e. either on cold utility systems (generally cooling water systems) or on hot utility systems (generally boiler and steam systems). To illustrate the foregoing point, a selection of the published work from literature will be discussed briefly in this section. Further details of these and other studies are comprehensively presented in Chapter 3.

Nishio *et al.* (1980) were among the early researchers to look into the optimisation of steam and boiler systems (Nishio & Johnson, 1977, 1979). They presented a method for the optimisation of a steam turbine power plant based on the estimation of thermodynamic losses and irreversibility in the system. Their method was later followed by Chou & Shi (1987) who presented a similar approach on the design and synthesis of power plant utility systems consisting of steam turbines and gas turbines. Papoulias & Grossmann (1983 a, b, c) presented a MINLP approach for the optimisation of heat and power for a total site. Their work was undoubtedly the best and most elaborate illustration of the applicability of mathematical optimisation in the synthesis and optimisation of chemical plants. At the time of their work, there was little confidence and faith on mathematical optimisation as a design tool and thus in a set of 3 papers Papoulias & Grossmann (1983 a, b, c) showed the importance of the technique and other advantages it provided as opposed to the heuristics and thermodynamic approaches that had then found wide acceptance. Petroulas & Reklaitis (1984) also presented a mathematical programming approach for the synthesis of plant utility systems, by applying a decomposition strategy where they decomposed the design task into two sub problems, one of header selection and the other of driver allocation. A different approach using simulated annealing was presented by Maia *et al*. (1995). Mavromatis & Kokossis (1998a, b) presented a more rigorous method for the estimation of power produced in steam turbines. Their method accounted for the variation of the efficiency with size, load and operating conditions. Rodriguez-Toral *et al.* (2001) presented an equation oriented (EO) mathematical modelling approach to the optimisation of utility systems. Their model was constructed based on balance and performance equations on heat and power systems. Varbarnov *et al.* (2004) presented an MINLP method for the simulation and optimisation of utility cogeneration systems. They first presented improved methods for modelling of steam turbines and gas turbines from previous models (Mavromatis & Kokossis, 1998 and Shang, 2000) and later applied these in optimising a utility system.

A common feature in the above discussion is that the discussed methods all focus almost exclusively on power and hot utility system optimisation and do not mention in any detail the equally important cooling water system. Some of these studies looked to a certain extent into the background processes. However, a greater number of them treat the cooling water temperature as a fixed parameter and thus do not look at the cooling water network in any detail. Such an approach thus ignores the complementing aspect of treating the utility system in a comprehensive manner and thus overall system optimality cannot be guaranteed.

Recent work also on steam and boiler optimisation includes that of Coetzee & Majozi (2008) and Chen & Lin (2011). Coetzee & Majozi (2008) reduced the steam requirement for a process by following the work of Kim & Smith (2001). They presented a hybrid graphical and mathematical technique for targeting and heat exchanger network synthesis, were the series connection of heat exchangers is preferred to parallel connection, thus utilising some of the sensible heat in the steam. Chen & Lin (2011) developed a steam distribution network for satisfying the energy requirements of a process. They used the transhipment model of Papoulias & Grossman (1983b) and adapted it to simultaneously consider opportunities for steam usage and steam generation.

With regard to the cooling water or cold utility system optimisation, the trend in the published literature has been the same. Early work includes that of Lefevre (1984) who investigated a number of ways of reducing water losses from wet cooling towers. He identified the major losses being those emanating from evaporation and blowdown and conceded that not much can be done to reduce the evaporation losses as these are dependent on the ambient conditions for which humanity does not have much, if any control. He thus recommended the use of as low air flowrate as is possible. Bernier (1994) investigated the effect of a number of variables including the water and air flow rates and fill height on the performance of a cooling tower as indicted by the cooling tower characteristics. Castro *et al.* (2000) presented a model for a cooling tower system incorporating the heat exchanger network. Their work

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was different in that it attempted to account for the pressure drop in the network, and that it was also based on correlations to predict the water properties and cooling tower performance.
Kim & Smith (2001) were amongst the first to treat the cooling water system and the associated heat exchanger network (HEN) in a single approach, with most of the previous research having treated the cooling water network and the cooling tower in a discreet manner without consideration of the entire cooling system components. Their method of optimising the HEN was based on Water Pinch principles and a graphical technique was used to arrive at the minimum water requirement for the HEN. Their work was recently extended by Panjeshahi & Ataei (2008) who used ozone treatment to improve the quality of the circulating water in the cooling system. Papaefthimiou *et al.* (2006) also investigated the effect of atmospheric conditions, including the inlet water and air temperatures on the performance of the cooling tower. More recent work on cooling water optimisation was by Gololo & Majozi (2011) who presented a mathematical technique for the optimisation of a heat exchanger network with multiple cooling water sources. Most of the work done on the cooling water system optimisation thus, like that of the hot utility systems, focuses almost exclusively on the different components of the complete utility system.

Limited research work is available from the literature that presents a combined and holistic treatment of the utility systems and two notable examples are the work of Barigozzi *et al*. (2011) and Gharaiea *et al*. (2013). These however have the drawback of presenting a simplistic integration of the utility systems.

From the preceding discussion, it is evident that a false dichotomy has been created between utility systems that are found in chemical or process plants. Both cooling water and boiler systems are almost invariably linked as found in process and power plants and thus complete optimisation of industrial utility systems can only be realized by considering the utility system as a whole. This work therefore sought to address this oversight by a number of researchers and develop a holistic method for the analysis and optimisation of a plant water system using Process Integration.

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1.2 Motivation for the study

Figure 1-1 below shows hot and cold process composite curves as found in literature. As seen in the figure, optimisation of the process-process section is achieved through heat integration as described by Linnhoff & Hindmash (1983). Again both external cold and hot utilities sections can be optimised, for example following the methods of Kim & Smith (2001) and Coetzee & Majozi (2008), respectively.

As has already been highlighted above, the three optimisation methodologies outlined in the foregoing paragraph, focusing on the three sections of the composite curve, have the common goal of reducing the heat duty requiring external utilities but treat the entire water system as three individual systems. This observation suggests that even better results could be found by treating the water system as a whole. This work was thus aimed at developing a methodology for the design and optimisation of a plant utility system as outlined in Figure 1-2 that exploits the integrated nature of such a system.

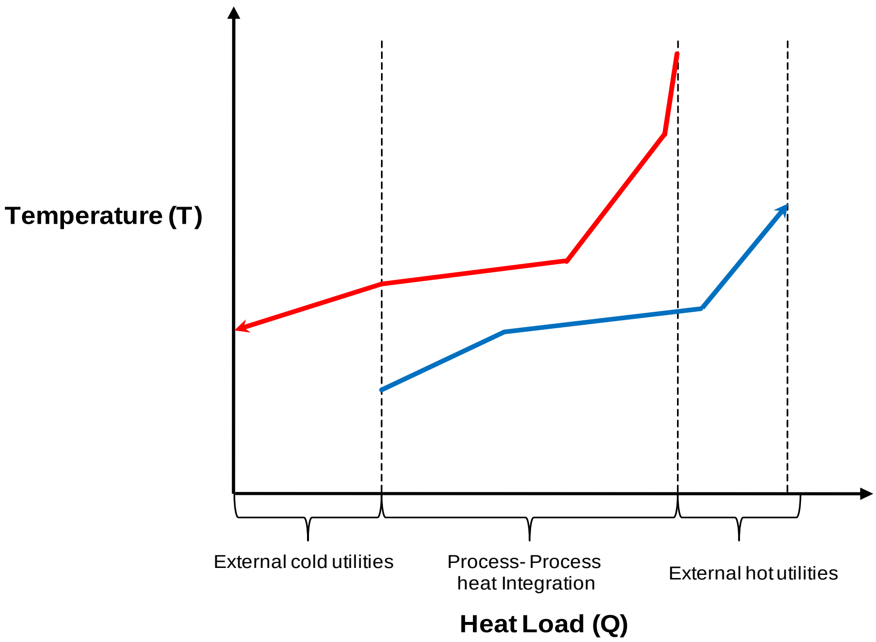

Figure 1- 1, The hot and cold process composite curves

The following section presents the scope of the research study, giving an overview of a typical industrial process which lends itself to such a holistic analysis and which formed the basis for the research work.

1.3 Scope of the study and problem definition

This section presents the problem which this research work aimed at solving. Figure 1-2 below shows a simple Rankine cycle for power production. This is a basic cycle consisting of four main components; the boiler, the turbine, the condenser and the boiler feed water pump. The simple cycle is different to the Rankine cycles found in modern power plants, where additional components are usually added as a means of enhancing the performance of the cycle (e.g. reheat and regenerative cycles). The treatment of such advanced systems follows the basic analysis presented in this work and thus will not be discussed further, except for a regenerative cycle with feed pre-heating as shown in Figure 1-3, which will serve as an illustration of the applicability of the principles outlined in this work. Also shown in Figures 1-2 and 1-3 is the cooling water system which supplies water used as cooling medium in the condenser. The water system consists of the cooling tower and the circulating system including the water pump.

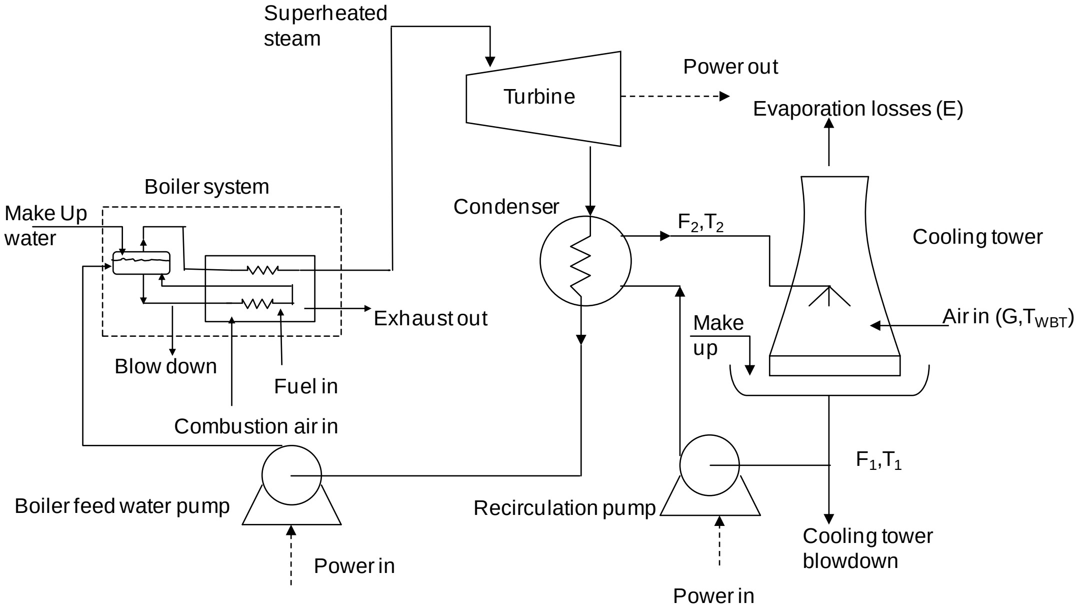

Figure 1 - 2, Simple Rankine cycle for power generation

The above figure combines two utility systems, the boiler and the cooling tower, which constitute a comprehensive utility system. These two systems, although complementing each other in real operations, have been treated individually by a number of researchers (e.g. Rodriguez *et al*., 2001, Dincer & Muslim, 2001 and Khan

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et al., 2003). This is testament from the huge amount of research that has been conducted over the years on either system (See Chapter 3). Thus a lot of work has been done previously on optimising the simple Rankine cycle and similarly on optimising the cooling water system. However very limited work is available for the comprehensive system and it is this realisation that led to this work - the focus being to develop an integrated model for the optimisation of the complete system as shown in Figure 1-2.

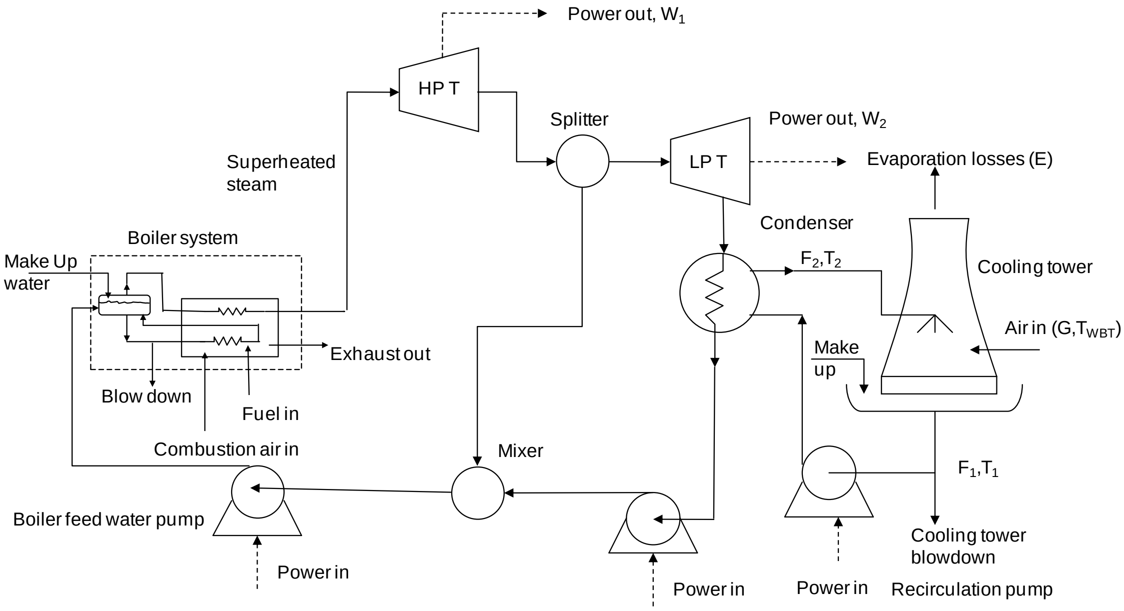

Figure 1 - 3, Regenerative steam turbine cycle with a single feed pre-heater

The regenerative cycle, shown in Figure 1-3 with a single feed pre-heater, is a modification of the simple Rankine cycle where a feed water heater is incorporated. A fraction of the circulating steam is extracted from the turbine and used to preheat the water leaving the condenser prior to it being pumped directly to the boiler. This has the effect of increasing the thermal efficiency of the cycle.

Kim & Smith (2001) presented a technique for the optimisation of a cooling water system comprising a cooling tower and its associated heat exchanger network. They showed that improved system performance is obtained from a complete system analysis as opposed to focusing on the individual components of the cooling water system. This work builds from a similar understanding in evaluating and optimising power plant utility systems.

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1.4 Problem statement

Most of the published methods for the optimisation of power plant utility systems have ignored the complementing and synergistic aspect of the cooling tower and boiler systems as found in real power plants, and therefore a need exists for a comprehensive system model. Individual and discrete analysis creates a false dichotomy between the boiler and cooling water systems and generally gives suboptimal results and thus the two systems, which constitute a comprehensive utility system, should be optimised in a single approach.

In this work detailed nonlinear mathematical models are developed first for each of the two subsystems and later combined to give the overall system model. Such a holistic model is necessary for the design of new and optimisation of existing power plants.

1.5 Aims of the study

The intent of the research work is to develop a process integration approach for efficiency improvement in a typical power plant utility system. The aims are therefore:

(i) To develop a comprehensive utility system model for the simple power plant shown in Figure 1-2

(ii) To determine the optimum operating conditions of a power plant similar to that shown in Figure 1-2 for known operating conditions and equipment sizes

(iii) To determine the optimum sizes for the cooling tower and steam systems for a given power output from the facility

(iv) To repeat the above analysis as sensitivity based on a regenerative steam cycle as shown in Figure 1-3.

1.6 Thesis outline

Following this introductory chapter is the literature review, Chapter 2, which starts with an overview of the basic concepts and principles involved in boiler and cooling tower systems. The fundamentals of Process Integration are presented before a

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discussion on the mathematical techniques that have successfully been applied by a number of researchers in solving both Energy and Mass Integration problems. The chapter also focuses on the historical methods that have been developed over the years where the literature review on steam and energy systems optimisation is presented first and is followed by that on cooling water systems as found in chemical and allied process industries. Specific energy integration methodologies for both external cold and hot utilities (cooling water and boiler systems, respectively) are presented including a review of the methods that have attempted to present a combined approach for both cold and hot utility integration in optimising utility consumption in process industries.Chapter 3 presents the models used in this work. In this chapter, the development of the comprehensive system model which entails mathematical formulation and development of solution procedures and techniques is presented. The penultimate chapter presents the results from the application of the developed method of analysis and Chapter 5 which wraps the report presents the conclusions and recommendations from the research work.

1.7 Contributions to the body of knowledge

Apart from the development of a new approach to both the optimisation and design of power plant utility systems, the research work has highlighted the oversight by a number of researchers in not taking the advantage of a total optimisation approach in optimising power plant utility systems. It indicated that when operating a dedicated cooling tower system in a power plant, optimisation of the cooling tower outside of the steam system and vice versa does not generally give overall optimum performance. The operation of the boiler system depends on the operation of the cooling tower and vice versa, and thus, for both existing and in the design of new power plants, the two systems should always be optimised in a combined approach.

The work done not only explores new ways of improving the efficiency and operations of a plant utility system, but also provides in-depth knowledge and understanding of the operation of the individual utility systems typically found in power plants. The knowledge can also be applied in the optimisation of similar processes found in the chemical and allied process industries.

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2

Chapter Two

2 Literature review

2.1 Introduction

This chapter presents a literature review on the optimisation methods for utility systems as applied in the broader field of Process Integration. Focus is on Energy Integration with a particular emphasis on hot and cold utility optimisation. It is shown in the discussion that complete optimisation of a plant utility system can only be achieved from a holistic treatment of the utility system as opposed to discreet and individual treatment of the systems' components.

The chapter starts with an outline of the fundamental principles involved in power production by presenting a description of the boiler as used in the production of steam together with the different thermodynamic power cycles for electricity generation. The discussion on boiler systems is followed by a similar discussion on cooling towers or cold utility systems, where the different types of cooling towers are presented. The origins of Process Integration are then explored where the key concepts involved in Process Integration are discussed ahead of a mathematical treatise that underpins most of the modern methodologies of Process Integration, methodologies that indeed can be easily programmed for solution on a modern digital

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computer. The chapter also presents the fundamental principles involved in Energy Integration with the focus being on heat integration (Pinch Technology) and on combined heat and power integration (CHP). A historical outline of the Energy Integration methods that have been developed for both external cold and hot utilities forms a greater part of the discussion. The shortcomings in the previous research are identified thereby highlighting the research gap that this work had aimed at closing. The discussion also includes those methods that have attempted to treat external cold and hot utility optimisation in a single combined approach.

The chapter ends with a discussion on the other two main branches of Process Integration, viz. Mass and Property Integration.

2.2 Fundamental principles – Boiler systems

Boiler systems as found in chemical and power plants include the generation of steam and its subsequent use in electricity production or as a hot utility. This section thus presents first a discussion on steam generation which is followed by a discussion on the different steam cycles for power generation, cycles which formed the basis for this work.

2.2.1 Steam generation

Steam is the main energy carrier in process industries, carrying energy both into and out of processes (Marechal & Kalitventzeff, 1997). When used as an energy carrier from a process, steam is usually produced by heat exchange between the process and water in waste heat boilers. The process streams are cooled while the water is vaporised, and at times even superheated producing steam. On the other hand, when used to carry energy into a process, the steam is usually generated from an external heat source in a boiler and used as hot utility in a process. Such use of steam is discussed in terms of the composite curves later in this chapter.

Two types of industrial boilers for steam generation are available and these are the fire tube and water tube boilers. As the names suggest, in the fire tube boilers, water

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is in the shell side of the boiler with the hot gases (fire) inside the tubes of the boiler, and for the water tube boilers, the opposite occurs with water in the tubes and fire on the shell side. Because fuel combustion in boilers occurs at low pressure (very low relative to the steam pressure), the economical arrangement for high pressure steam generation is therefore the water tube boilers where the smaller sizes of the tubes allow for stronger and more expensive exotic material use.

Of all the available fuels for boilers, which include solid, liquid and gaseous fuels, coal is the main fuel that is used for steam production in the process and power industries. The main reason why coal has found greater use is its general abundance in most parts of the world and as a result being a cheaper fuel. Large coal fired power plants typically use pulverised coal as a fuel. Circulating fluidised bed boilers (the so called CFB's) are the equipment of choice in modern power plants with the water-tube arrangement being common (Perry & Green, 1997). Other fuels include diesel, natural gas, oils and other hydrocarbons and more recently biomass due to environmental attractiveness. In the power industry, the heat source could also be from nuclear reaction, geothermal or solar energy, or heat from gas turbine exhaust (Tanaka & Wicks, 2010).

2.2.2 Simple Rankine cycle

Figure 2-1 below shows a simple Rankine cycle for power generation. The cycle consists of four interconnected components, the central being the steam turbine which produces shaft power (and ultimately electric power when connected to a generator), a condenser which uses external cooling water to condense the steam to liquid water, a feed water pump to increase the pressure of the circulating water to that corresponding to the turbine inlet pressure, and finally a boiler which is externally heated by combusting a fuel producing steam from water as described in the previous section.

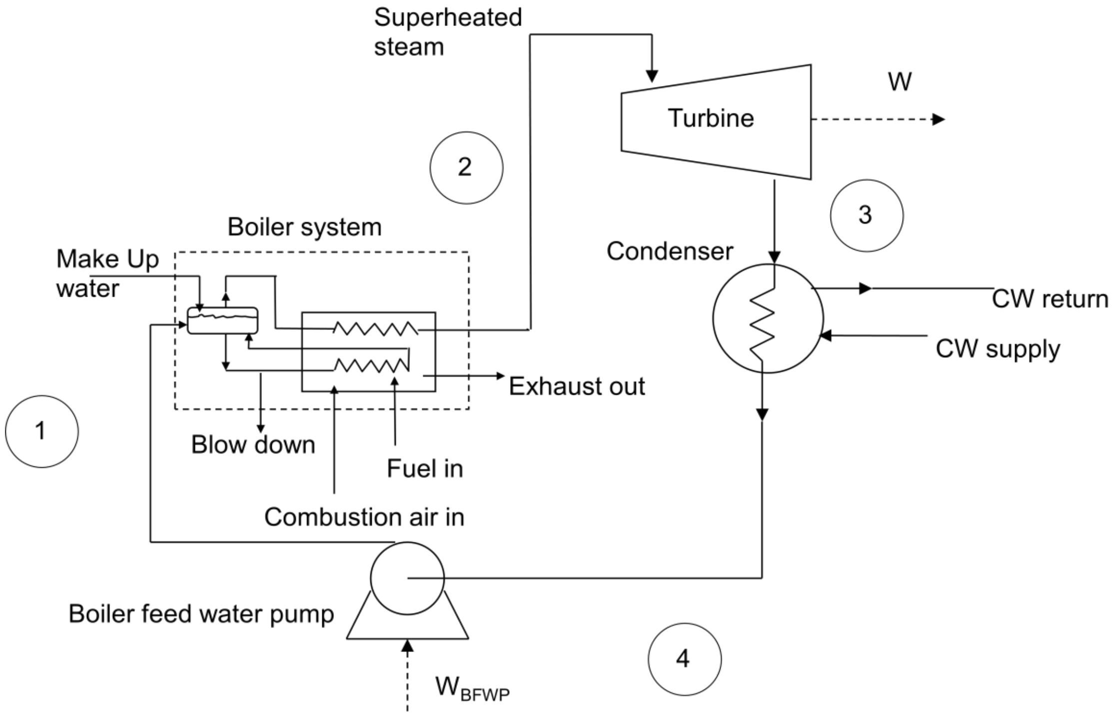

Figure 2 - 1, Simple Rankine cycle

The processes that occur as the circulating water, also called the working fluid (Smith *et al.*, 2004), flows around the cycle are represented by lines on a temperature entropy diagram (T-S) diagram in Figure 2-2. Step 1-2 is the vaporisation processes which takes place at a constant pressure in the boiler where three processes take place, the first being the heating of the sub cooled liquid to the saturation temperature and the second being the vaporisation of the water into steam at a constant temperature and pressure and lastly the superheating of the vapour to a temperature above the saturation temperature.

Step 2-3 is the isentropic expansion of the vapour (steam) to the pressure of the condenser, generally crossing the saturation curve producing a wet exhaust. As the steam expands, its pressure and kinetic energy act on the turbine blades, turning the shaft and the coupled electrical generator. The turbine thus converts the steam energy to work which the electrical alternator converts into power (Tanaka & Wicks, 2010). Step 3-4 is a constant pressure process where the vapour is fully condensed to saturated liquid water at point 4. The last step, Step 4-1 is the pumping of the water to the pressure of the boiler, at which point the cycle begins.

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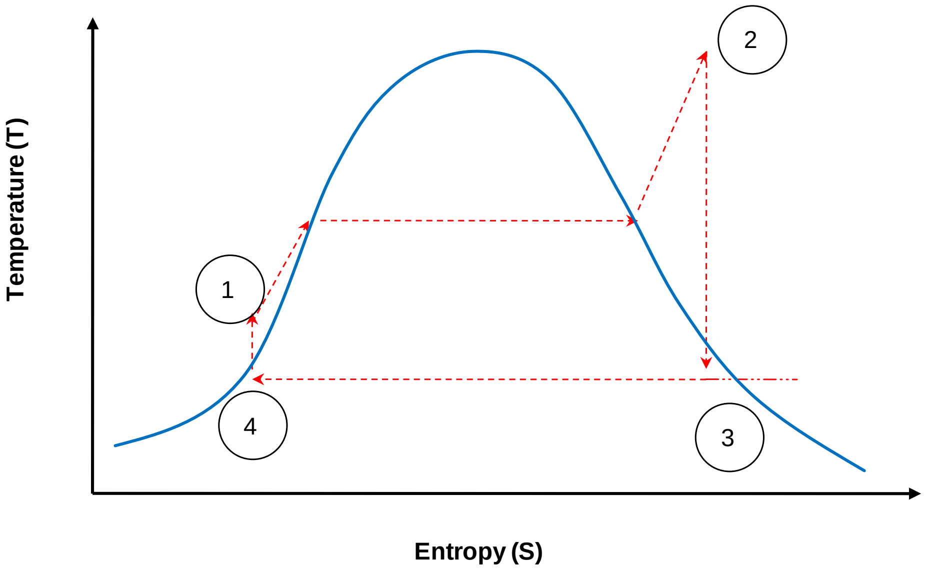

Figure 2 - 2, The Rankine cycle on a T-S diagram

Like all heat engines the efficiency of the Rankine cycle is a function of the heat source (boiler) and heat sink (condenser) temperatures. For an Ideal engine (Carnot engine) operating reversibly between a high temperature T_H where heat Q_H is absorbed and a cold temperature T_C where heat Q_C is released, the work produced by the cycle is given by Smith *et al.* (2005) as:

$$W = Q_H - Q_C \tag{2-1}$$

and the thermal efficiency of the engine as

$$\eta = \frac{W}{Q_H} = 1 - \frac{T_C}{T_H} \tag{2-2}$$

From the preceding two equations it is clear that the wider the range of temperature between the heat source and sink, the more efficient the cycle becomes. According to Rogers & Mayhew (1997), the lowest possible temperature of the cycle is governed by two factors, the first being the temperature of the sink of heat (which could be the atmosphere in air coolers, supply temperature in cooling towers, ocean or river water temperatures) and the second being the temperature difference required for the heat transfer processes in the condenser.

On the other hand the maximum possible temperature of the working fluid is governed by the strength of the materials available for the highly stressed parts of the plants (boiler tubes and turbine blades) and the current limit is in the 600 to 650 ^{o}C range (Rogers & Mayhew, 1997).

In light of the foregoing discussion and the fact that real power plants have a lot of irreversibility, the Carnot efficiency is thus never attained in real plants. It is therefore a theoretical limit indicating the potential of a system. The Carnot efficiency also assumes that all the heat addition to the working fluid is added at the higher temperature T_H, whereas in reality most of the heat is added as latent heat at the boiler saturation temperature (Tanaka & Wicks, 2010). As such, real plants thus have efficiencies much lower than the Carnot efficiency.

Two ways of improving the efficiencies of real plants are through reheating and regenerative steam cycles. These are discussed in turn below after a brief discussion on extraction turbines.

2.2.3 Extraction turbine

In an extraction turbine, steam is extracted from the turbine and can be used as a hot utility for process heating or for use in feed water preheating when applied to regenerative steam cycles. The use in a regenerative steam cycle is discussed in more detail below.

2.2.4 Regenerative cycle

A regenerative cycle, shown in Figure 2-3 for a single feed pre-heater, is a modification of the simple Rankine cycle where feed water heaters are incorporated. Steam that is extracted from the turbine is used to preheat the water leaving the condenser prior to it being pumped directly to the boiler. In modern power plants, more than one feed pre-heater is installed with several of these generally installed where steam is taken from the turbine at successive intermediate states of expansion. According to Tanaka & Wicks (2010), some plants may employ up to 12 stages of feed water heating.

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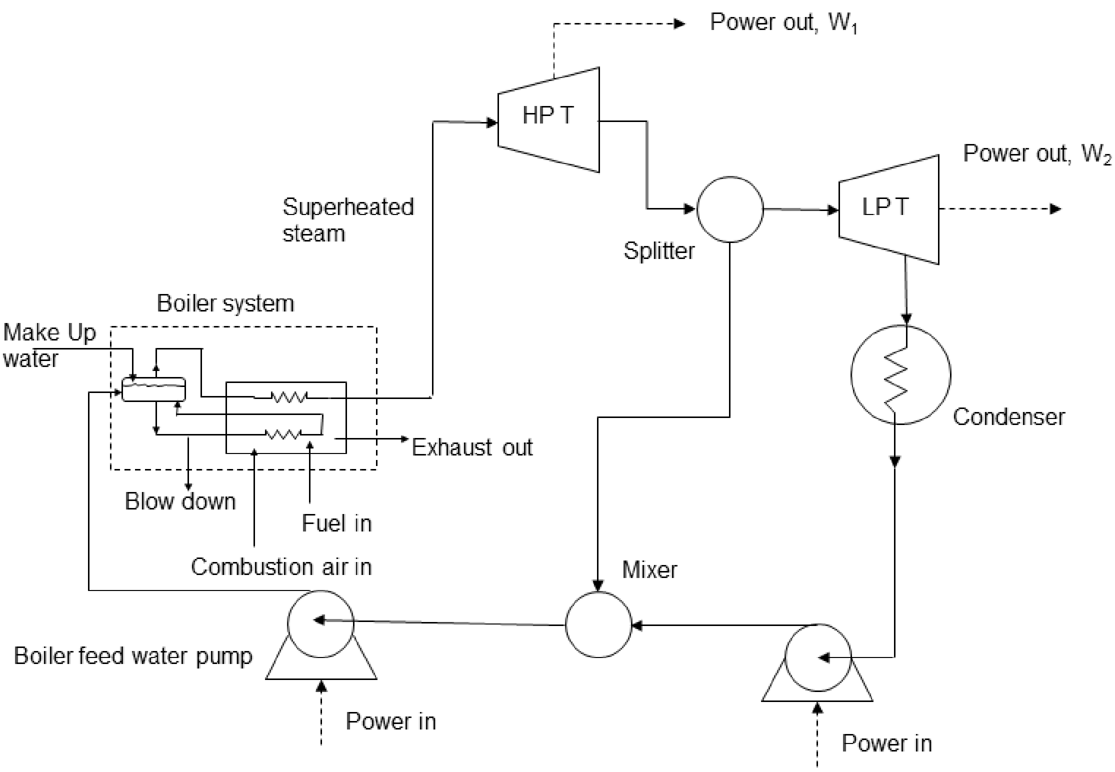

Figure 2 - 3, A regenerative steam cycle

In the above figure, the expansion is shown as taking place in two turbines, mainly for modelling purposes. In real plants, however, a single turbine is used with several extraction ports along the turbine, from which steam at different pressures is extracted.

Two approaches to feed water heating are available and these are 1, the mixing of the bled steam with the working fluid in open feed water heaters and 2, heat exchange between the bled steam and the working fluid in closed feed water heaters. Power plants generally use a combination of these, with de-aeration being an example of direct or open feed water heating (Rogers & Mayhew, 1997).
Figure 2-4a below shows a T-S diagram for an ideal regenerative cycle where part of the heat is removed from the turbine through heat exchange and added to the boiler feed water.

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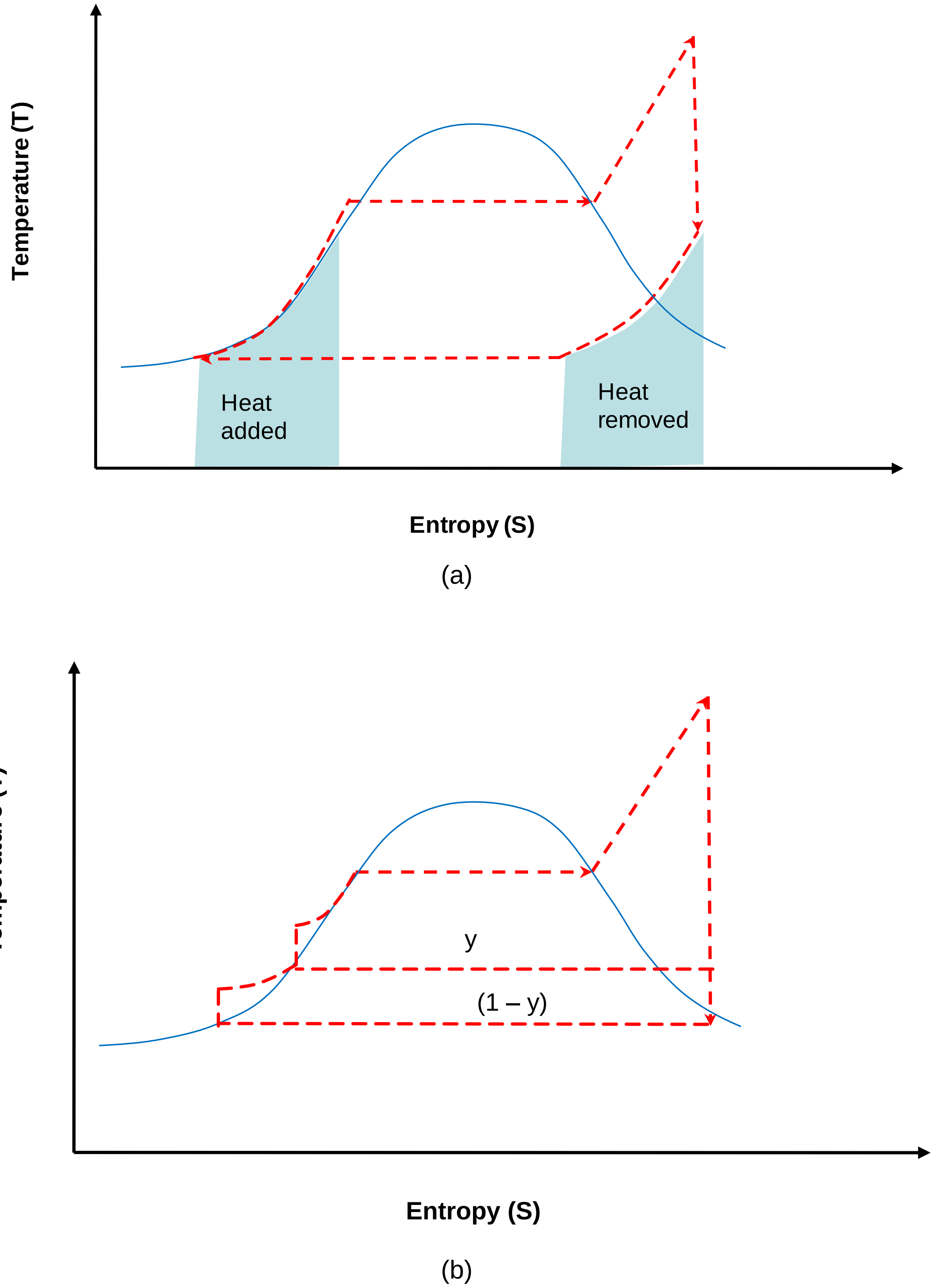

Figure 2 - 4, T-S diagram with condensed water heated by turbine steam (a) – Ideal regeneration, (b) practical regeneration (Rogers & Mayhew, 1997, Tanaka & Wicks, 2010).

As seen in the above figure, the effect of feed water heating is the reduction of the heat supplied from the heat source which is coupled with a reduction of the heat

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rejected. However the regenerative cycle shown in Figure 2-4a is impractical as it would be impossible to design a turbine that would operate efficiently both as a turbine and as a heat exchanger (Rogers & Mayhew, 1997). As such the simplest approximation of a practical regenerative cycle is shown in Figure 2-4b where a fraction y of circulating steam is extracted from the turbine and taken to a feed water heater. The remaining 1 – y fraction expands to the condenser pressure after which it is pumped to the bleed pressure where it mixes with the extracted steam before the combined stream goes to the boiler. As highlighted above, the average temperature at which heat addition in the boiler takes place increases and the resultant effect is that the efficiency of the regenerative cycle increases above that of a simple cycle (Smith *et al.*, 2005; Tanaka & Wicks, 2010). However feed water heating reduces the amount of work produced by the cycle and also adds cost and complexity to the plant.

2.2.5 Reheat steam cycle

In reheat cycles, the expansion of the steam occurs in two turbines, where the steam first expands in the high pressure turbine to some intermediate pressure, from where it is taken back to the boiler where it is reheated at constant pressure, generally to the original superheat temperature (Rogers & Mayhew, 1997). From there it goes to the LP turbine where it expands to the condenser temperature as in the other cycles discussed above.

Figure 2-5 below shows the reheat cycle and Figure 2-6 shows the T-S diagram of a reheat cycle.

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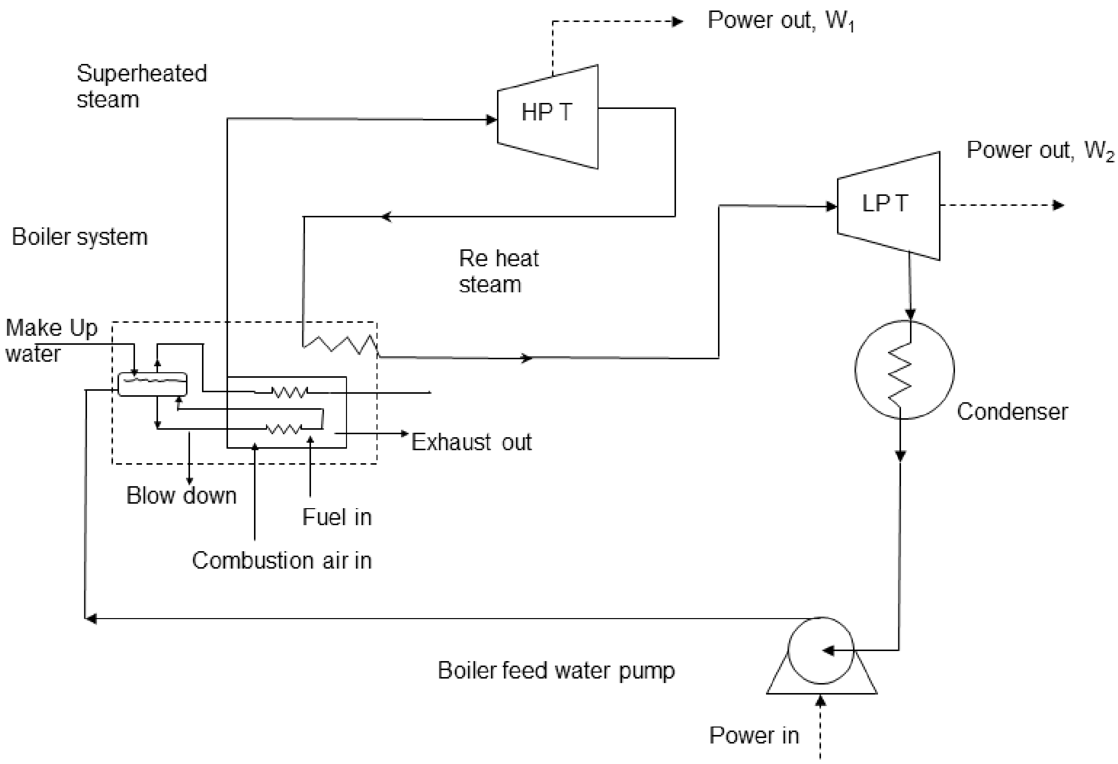

Figure 2 - 5, The reheat cycle

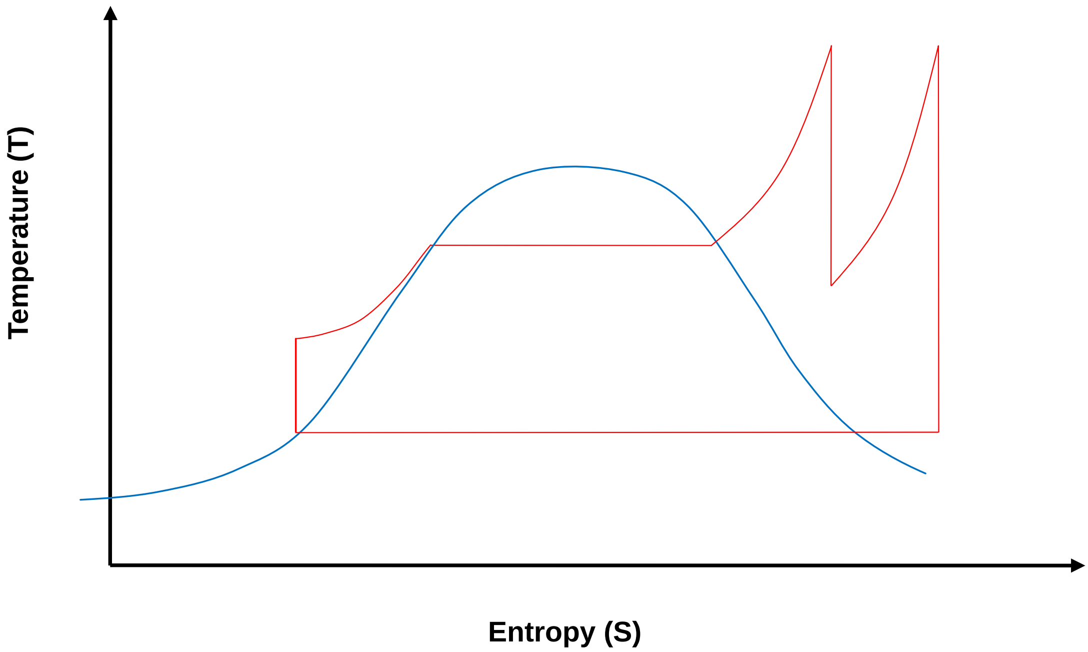

Figure 2 - 6, T-S diagram of a reheat cycle.

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The effect of reheating the working fluid is increased work output from the cycle for a given steam flowrate. Thus steam consumption per unit of work produced reduces as indicated by the increased area under the TS diagram in Figure 2-6. According to Rogers & Mayhew (1997), the reduction in the steam flowrate is particularly important in very high pressure cycles because it implies a smaller boiler – an expensive piece of equipment in a high pressure plant. They however conclude that the decrease in size cannot completely offset the drawback of added complexity and thus the main reason for reheating is to avoid too wet conditions in the turbine.

The effect of the reheating depends on the pressure (or point along the expansion) at which reheating is carried out. Rogers & Mayhew also compared typical modern supercritical cycles (see following Section for a discussion on supercritical cycles) where reheating is assumed to occur when the steam is just saturated (with the quality x being equal to 1). Their results, showing the cycle efficiency, the specific steam consumption per unit of power produced and the quality of the exit steam from the turbine are shown in Table 2-1 below.

Table 2 - 1, Effect of reheat pressure on efficiency (Rogers & Mayhew, 1997)

	Superheat to 600 °C		***superheat and reheat to 600 °C***	
	160 bar	**350 bar**	**160 bar**	**350 bar**
Thermal efficiency	0.453	0.486	0.468	0.485
specific steam consumption (kg/kWh)	2.31	2.37	1.76	1.8
turbine exit steam quality	0.772	0.708	0.954	0.871

The table shows the effect on reheating on the quality of the exit steam where an increase in the quality is obtained through reheating. This, as highlighted above, is important in high pressure cycles where the impact of moisture in the exhaust is very severe.

The table shows that the increase in efficiency resulting from the use of supercritical pressures is small but according to Rogers & Mayhew, this should not be discounted and they give a case where for a cycle having 40% efficiency, a 1 % increase in efficiency will translate to 2.5% saving in fuel.

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2.2.6 Supercritical steam cycles

Supercritical cycles were referred to in the previous section. They have found increased application in modern power plants and in this section an overview of these cycles is presented.

Supercritical cycles are cycles where the pressure of the cycle is such that the critical temperature of water is exceeded in the boiler (221 bar and 374 ^{o}C). Is such a case boiling and evaporation in the boiler no longer occurs at constant temperature and the operation is termed "supercritical" (Tanaka & Wicks, 2010). The average temperature of heat addition and thus the cycle efficiency increases as was explained for the Carnot cycle above. Figure 2-7 below shows the T-S diagram for a supercritical steam cycle

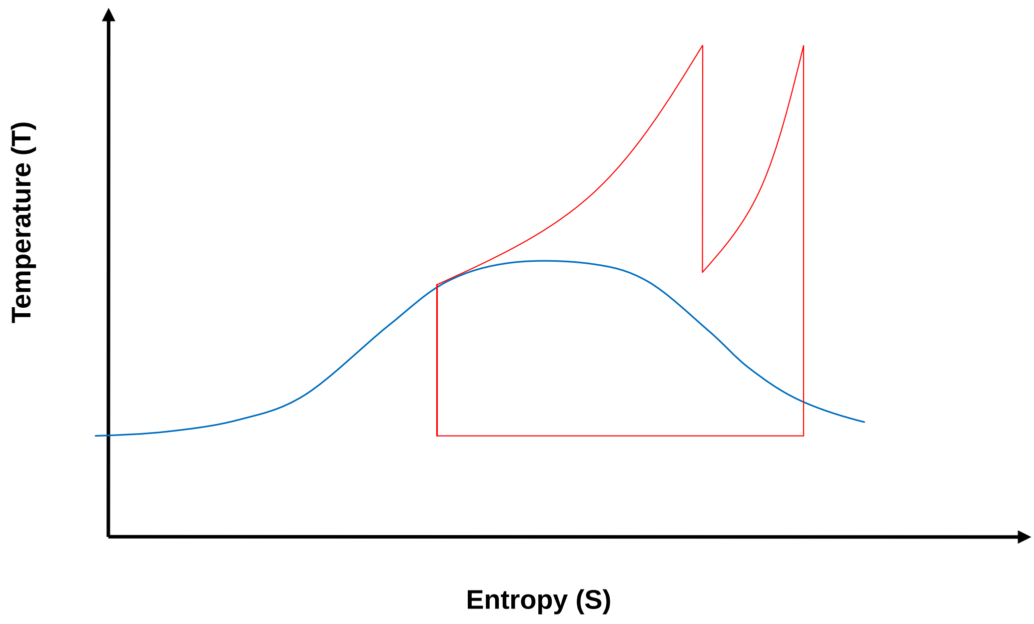

Figure 2 - 7, T-S diagram for a supercritical steam cycle (Tanaka & Wicks, 2010)

The boilers used in supercritical steam cycles are called once through boilers as no separation of water and steam is required. When the temperature of the steam is increased to more than 593 ^{o}C, the cycle is referred to ultra-supercritical (Tanaka & Wicks, 2010).

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The preceding sections gave an overview of steam production for use as a hot utility together with an overview of the different power cycles. In this work the simple Rankine cycle as discussed in Section 2.2.2 was used as further discussed in the following chapter, chapter 3. The regenerative cycle as discussed in Section 2.2.3 was also modelled to illustrate the applicability of the principles presented in this work.

The following section, 2.3 gives an outline of the fundamental principles involved in cooling towers as found in chemical and allied process industries.

2.3 Fundamental Principles - Cooling systems

This section presents a discussion on cooling water (cold utility) systems, and as was highlighted above. The different types of cooling towers are presented after an introduction on the principles involved in cooling towers.

2.3.1 Principles of evaporative cooling

When a warm liquid is brought into contact with unsaturated gas, part of the liquid evaporates into the gas with the result that the liquid temperature drops. This is the principle that is used in cooling towers where the temperature of circulating water from process equipment (which includes condensers, heat exchangers and power plants) is lowered (McCabe *et al.,* 2005).

According to Kroger (2004), the once through cooling systems using water bodies such as rivers and dams as heat sinks, have over the years declined, more so in industrialised countries, due to changes in legislation. Limits on maximum allowable temperature rise in these water bodies have been implemented, and the result has been that the cooling tower has become the equipment of choice for process cooling.

From the foregoing discussion, a cooling tower can thus be defined as a device that uses a combination of heat and mass transfer to cool water (Kroger 2004). The hot water that is cooled is fed into the tower at the top via spray nozzles which help distribute the water over a large area where it gets into contact with the rising air.

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Different fills are available and these are described below (See section 2.3.4). Two main types of cooling towers exist and these are the:

- Mechanical draft cooling towers and
- Natural draft cooling towers.

The main difference between the two classes is the mode of air movement in the cooling towers. In mechanical draft cooling towers, a fan is used to drive the air through the cooling tower whereas in natural draft cooling towers, the air is moved by the chimney effect which is a result of the presence in the tower of air and vapour of higher temperature and therefore of lower density than the surrounding atmosphere (Coulson & Richardson, 1996). The different types of cooling towers are presented in more detail below.

In all cooling towers (both mechanical and natural draft) the heat-transfer process takes place by two mechanisms. The first being the transfer of latent heat due to vaporization of a small portion of the water and the second being the transfer of sensible heat due to the difference in temperature of water and air. According to Perry & Green (1997) approximately 80% of this heat transfer is due to latent heat and 20% to sensible heat. Bernier (1994) reports that about 90% of the total heat exchange is by evaporation.
The amount of heat released from the water depends on the temperature and moisture content of the air for which the wet bulb temperature is used to give an indication of the moisture content (Perry & Green, 1997). Thus the wet bulb temperature represents the theoretical limit to which the water can be cooled.
The difference between the wet bulb air temperature and the outlet water temperature in a cooling tower is called the “approach”. This is shown in Figure 2-8, taken from Perry & Green (1997), which is an enthalpy - temperature diagram of the processes taking place in a countercurrent cooling tower (see following section on the different types of cooling towers). The approach temperature is generally 3 to 8 ^{o}C (McCabe *et al.*, 2005; Perry & Green, 1997). The change in the water temperature is known as the range and this is generally 6 to 17 ^{o}C (McCabe *et al.*, 2005).

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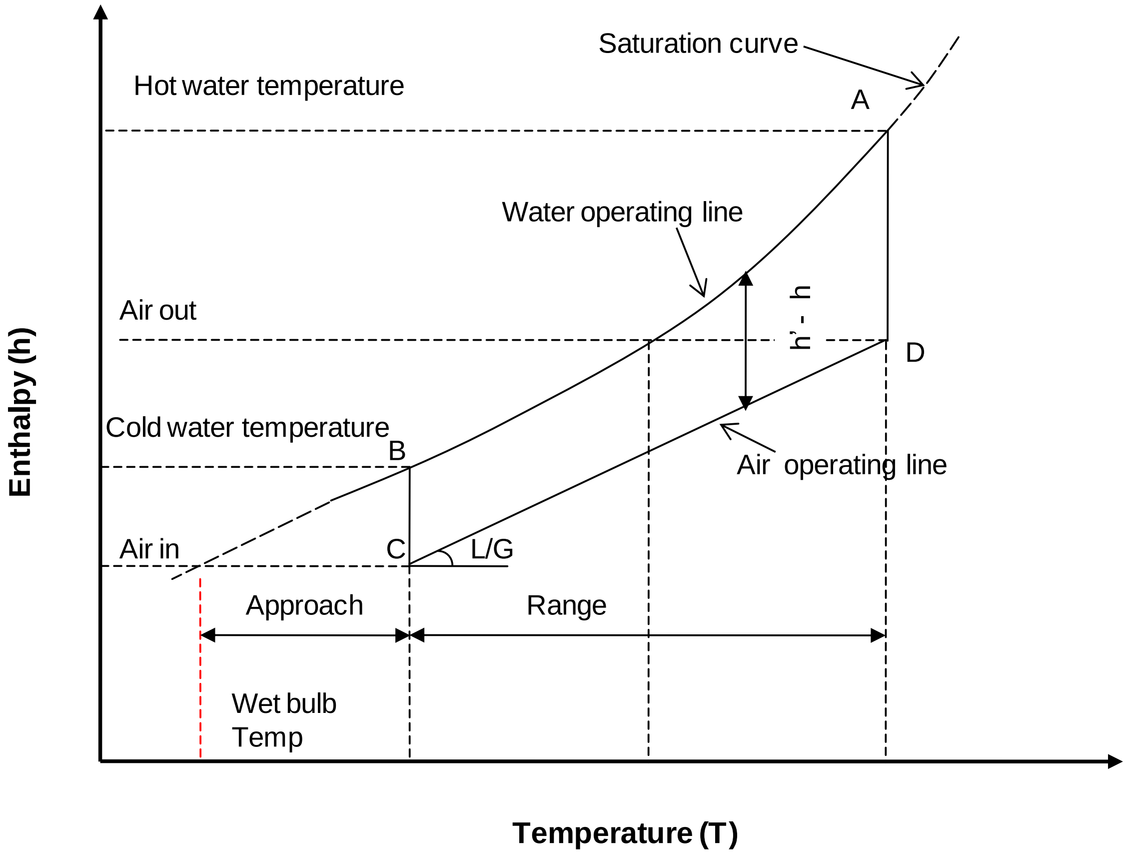

Figure 2 - 8, Cooling tower process heat balance (Perry & Green, 1997)

Line AB in the above figure represents saturated air conditions and is thus an equilibrium line which can be drawn from values obtained from humidity or psychrometric charts (Coulson & Richardson, 1996). The region above the line represents supersaturated air and that below it represents unsaturated conditions. The start and end of line *AB* are determined by the inlet and outlet water temperatures on the equilibrium line.

The equation of the air operating line CD is:

$$G(h_2 - h_1) = LC_P(T_2 - T_1) \tag{2-3}$$

The above equation is derived (for example by Coulson & Richardson, 1996) from taking mass and enthalpy balances over an incremental height along the cooling tower and integrating the resultant equation over the whole length of the cooling

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tower, after making certain assumptions. These assumptions, originally made by Merkel (1925), include that only a small portion of water is evaporated and thus the liquid flowrate remains constant along the height of the cooling tower and that the fluid properties do not change appreciably along the cooling tower.

The air operating line thus begins at C, vertically below B and at a point having an enthalpy corresponding to that of the entering wet-bulb temperature. Line BC represents the initial driving force for heat transfer which is the difference in enthalpy between the inlet air and saturated air at the outlet water temperature. The water to air ratio L/G is the slope of the operating line. The air leaving the tower is represented by point D and the cooling range is the projected length of line CD on the temperature scale (Perry & Green, 1997).

The Merkel equation, which is one of the widely accepted theories for describing cooling tower heat and mass transfer (Perry & Green, 1997), is given by:

$$\frac{KaV}{L} = \int_{T_1}^{T_2} \frac{dT}{h' - h} \qquad (2\text{-}4)$$

The value of the integral is represented by the area ABCD in Figure 2-8 and is known as the cooling tower characteristics and is a function of the L/G ratio. It is used to describe the performance of a cooling tower and is described further below and in Chapter 3 particularly with regards to cooling tower sizing as was done in this work.

Figure 2-9 below shows the temperature gradients inside the cooling tower and is taken from McCabe *et al.* (2005).

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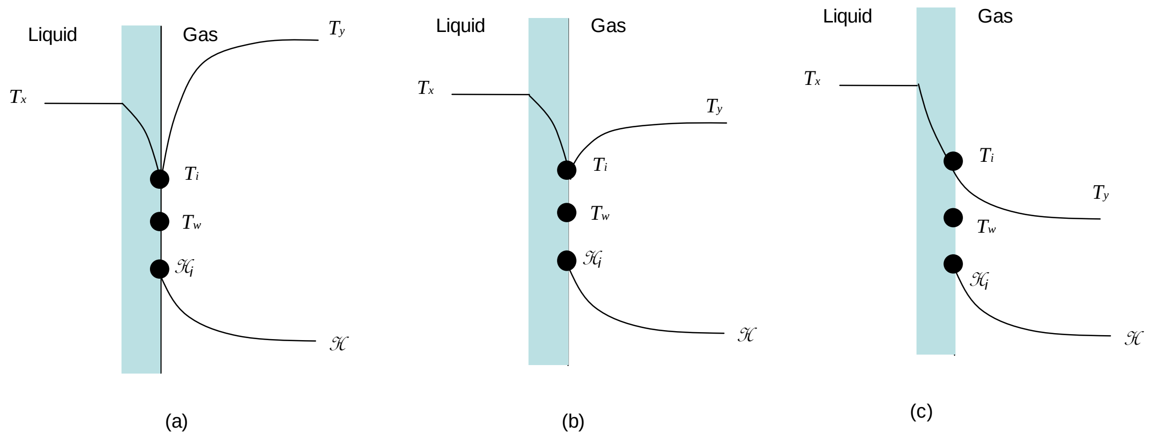

Figure 2 - 9, Temperature gradients inside a cooling tower (McCabe *et al.,* 2005)

At the bottom of the cooling tower, the air temperature may be greater than the water temperature and in that case the water is cooled because the interface temperature T_i is lower than the bulk water temperature T_x. Mass transfer occurs because the humidity at the interface is greater than that in the bulk air. On the other hand, if the air inlet temperature is less than the outlet water temperature as shown in Figure 2-9b, there will be less sensible heat transfer in the air due to reduced driving force but the direction of the gradients remain the same. For cooling to take place in both cases, the interface temperature must always be above the wet bulb temperature, T_w (McCabe *et al.*, 2005).

At the top of the cooling tower, the gradients are as shown in Figure 2-9c, where the heat is transferred from the interface and is used to warm the air as well as to provide the heat of vaporisation, with the evaporation being the dominant mode of heat transfer.

Further details and derivation of the mass and heat transfer equations taking place in cooling towers are available is standard text books for mass transfer and heat transfer and thus will not be presented in further detail here. Excellent texts for which the reader is referred to are Kern (1950), Coulson & Richardson (1996), Kroger (2004) and McCabe *et al.* (2005).

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2.3.2 Natural draft cooling towers

Figure 2-10 below shows the cross section through a natural draft cooling tower (Kroger 2004), which could either be of the cross flow or counterflow type. In these cooling towers, which have been in use since 1916 (Perry & Green, 1997), the air movement in the tower, as was highlighted above is due to the density differences between the air inside and outside the cooling tower. The air inside is warmer and hence lighter that the air outside the cooling tower, with the effect that the warm air continuously rises and is replaced by cold air from outside. The draft in these cooling towers is about 50 Nm^{-2} (Coulson & Richardson, 1996).

The natural draft cooling towers consist of empty shells which are made of timber or reinforced concrete and have only about 10 to 12 % of the total height, which may be up to 180 m tall (Kroger, 2004), fitted with packing for which the cooling takes place. The rest of the height provides increased draught and the hyperbolic shape is mainly for structural reasons (Coulson & Richardson, 1996).

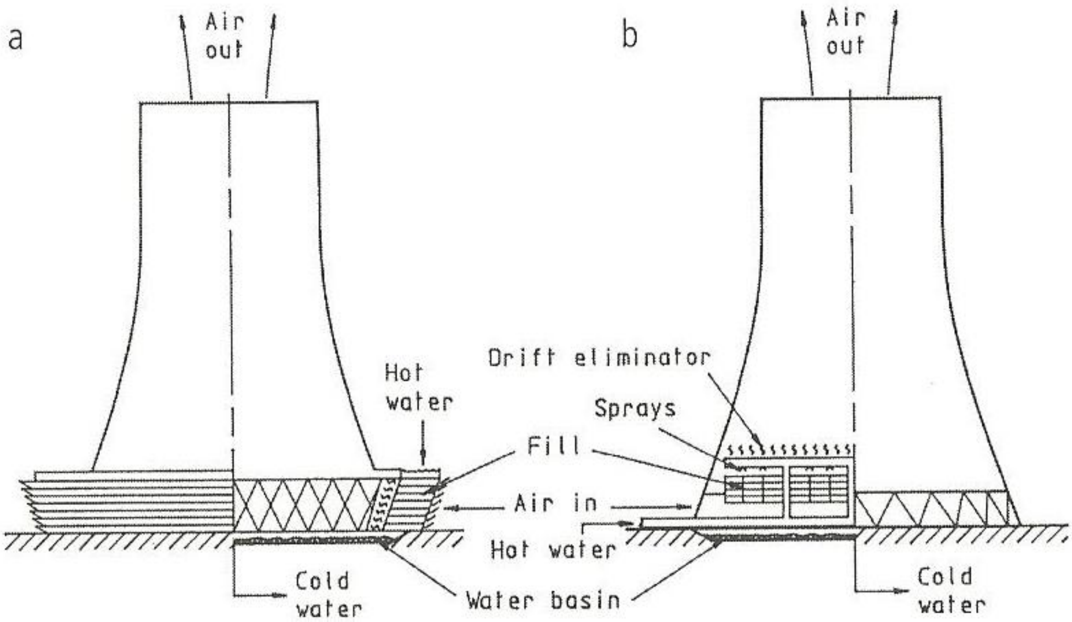

Figure 2 - 10, Natural draft cooling towers (a) Cross flow (b) Counter flow (Kroger, 2004)

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In the cross flow cooling tower, the fill configuration is such that the flow of air is perpendicular to the falling water. The water to be cooled is delivered by risers to the top of the fill where it is distributed by spray nozzles as outlined above.

One of the general problems with natural draft cooling towers is that their effectiveness is based on ambient conditions which cannot be controlled and thus at times excessive plume recirculation occurs. Such a problem can be avoided by having fan assisted natural draft cooling towers as shown in Figure 2-11 from Kroger (2004).

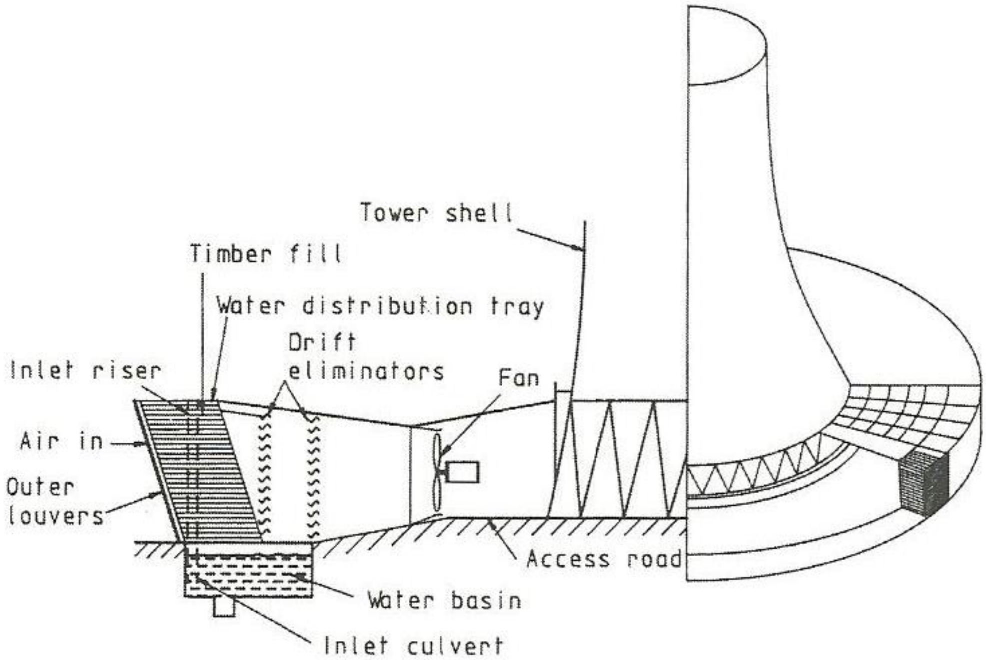

Figure 2 - 11, Fan assisted cross flow cooling tower, Kroger (2004).

2.3.3 Mechanical draft cooling towers

Mechanical draft cooling towers fall into two classes depending on the location of the fan that drives the air through the cooling tower. Induced draft cooling towers have the fan at the top of the cooling tower, where air is drawn from the bottom of the cooling tower and hot air exits at the top as shown in Figure 2-12.

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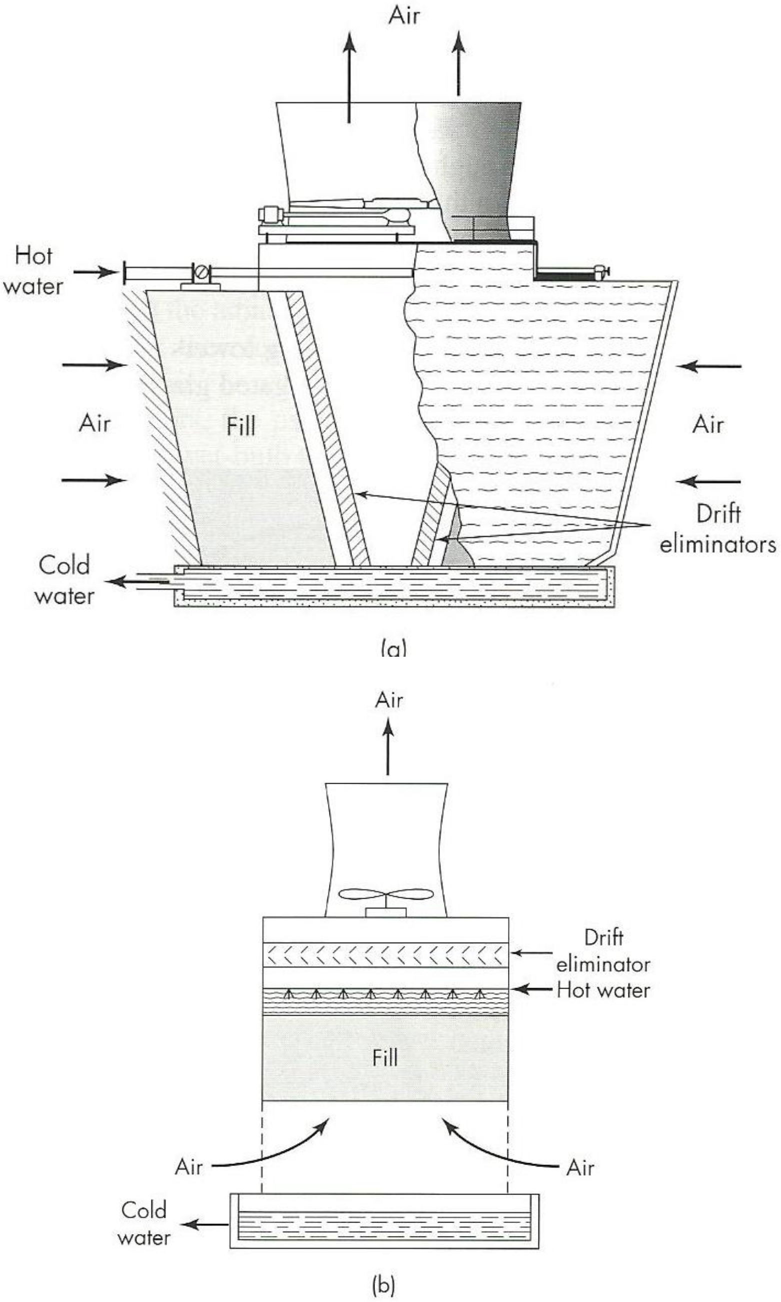

Figure 2 - 12, Induced draft cooling tower (a) Cross flow, (b) Counter flow (McCabe *et al.,* 2005)

On the other hand forced draft cooling towers have the fan located at the bottom of the cooling tower from where the air enters and is forced up through the tower (Figure 2-13).

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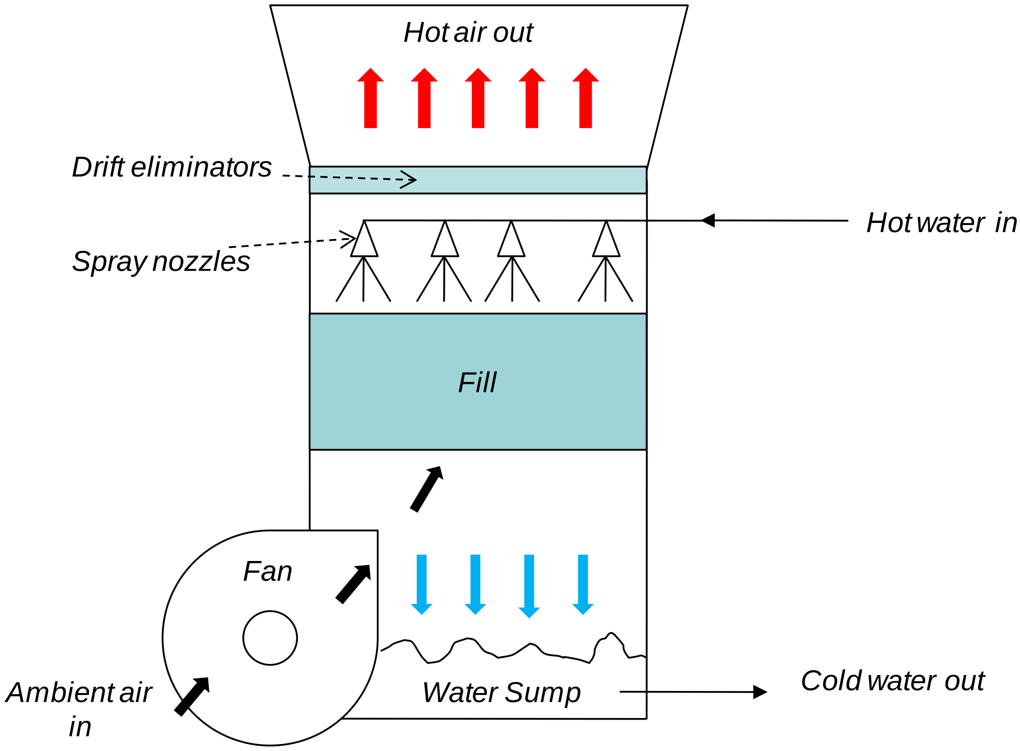

Figure 2 - 13, Forced draft cooling tower

Both configurations have their advantages and disadvantages with the advantage of the forced draft over the induced draft type being that the fan experiences less stringent conditions and thus requires less maintenance. The hot air leaving the induced draft cooling tower passes over the fan which means stronger and more resistant materials of construction will be needed.

Induced draft cooling towers can be of the cross flow and counter flow types as shown in Figure 2-12 and have the advantage of having less recirculation (Kroger, 2004). However, more fan power is required for the same mass flowrate of air due to hot air having a lower density. (Coulson & Richardson, 1997; Kroger, 2004)

2.3.4 Cooling tower fills

Cooling tower fills or packing are an important part of cooling towers and these basically determine the performance of the cooling tower in that they provide a large surface area that brings into contact the water and air phases so that effective mass and heat transfer can take place. The extent of heat transfer that can take place depends on this area (Mohiuddin & Kant, 2006).

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The basic types of fills used in modern cooling towers are the:

- Splash type
- Film type and
- The trickle type

The splash type of fill operates by breaking the water falling through the cooling tower into a large number of drops. As the water falls through the fill, the water collides with successive layers of splash bars thus helping in redistributing the water and heat (Kroger, 2004). Kroger (2004), states one of the drawbacks of the splash fill being that large volumes of water are needed to break the water which then necessitates large cooling towers. He also mentions that splash fills produce more carry over than other types of fills.

In the film type, the larger surface area is achieved by allowing the water to spread in a thin layer on the area of the fill as opposed to droplets from the splash type.

Trickle packs are a combination of splash and film fills and they have much finer grids than splash fills where the water runs down the grid without splashing (Kroger, 2004). Figure 2-14 below shows a picture of the different fill types taken from Kroger (2004).

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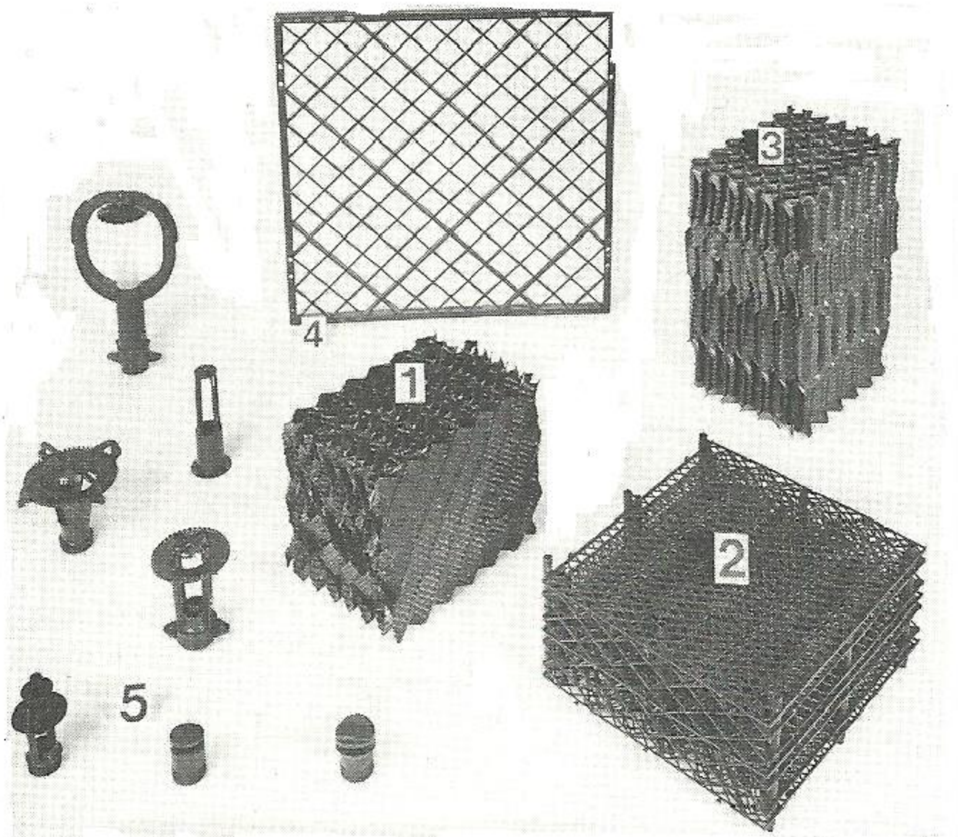

Figure 2 - 14, A picture showing different type of fills (Kroger, 2004), (1) Films, (2) Trickle, (3) Film, (4) Splash, (5) Spray nozzles

The choice of the type of fill depends on a number of factors which include its heat transfer performance, pressure drop, packing cost and durability (Mohiuddin & Kant, 1996). The main factors that determine the performance of a fill are the fill transfer and loss coefficients. These are generally experimentally determined from which empirical correlations are generated (Kroger, 2004; Kloppers & Kroger, 2005b).

The general empirical equations found in literature for the transfer and loss coefficient of fills have the following forms, respectively (Kroger, 2004):

$$\frac{h_d a_{fi}}{G} = a\left(\frac{L}{G}\right)^{-b} \tag{2-5}$$

and

$$K_{fi} = a\left(\frac{L}{G}\right) + b \tag{2-6}$$

A number of authors have given values for the parameters a and b for different types of fills (e.g. Coulson & Richardson, 1996; Lowe & Christie, 1961, and Kloppers & Kroger 2005b).

Kloppers & Kroger (2005c) evaluated a number of empirical correlations found in literature for the transfer and loss coefficient, looking specifically at the effect of fill height, inlet water and outlet temperature and inlet air dry-bulb and wet-bulb temperatures on the transfer characteristics. They extended the general form of the empirical equations (Kloppers & Kroger, 2003) to include the height of fill and the wet-bulb temperatures to be:

$$\frac{Me}{z_{fi}} = c_1 L^{c_2} G^{c_3} z^{c4}{}_{fi} \tag{2-7}$$

and

$$\frac{Me}{z_{fi}} = c_1 L^{c_2} G^{c_3} z^{c4}{}_{fi} T_{WB}^{C_5} \tag{2-8}$$

where c_1 to c_5 are parameters.

They conclude that the inlet dry-bulb temperature has no significant effect on the transfer characteristics and recommend that as much information as possible should be supplied with the empirical correlations of the transfer characteristics, which includes the ranges of applicability and the goodness of fit in the form of correlation coefficient. They point that such information will allow the design of cooling towers to account for the uncertainties and even compensate for these in the design if accurate and reliable designs are to be developed.

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2.4 Process Integration

According to El-Halwagi (1998), "*Process Integration is a holistic approach to process design, retrofitting and operation which emphasizes the unity of a process*". He defines it as offering a comprehensive framework for understanding the global insights of the process, methodically determining its attainable performance targets, and systematically making decisions leading to the realization of these targets (El-Halwagi, 1998). Mass Integration and Energy Integration are both branches of Process Integration and deal respectively with the flow of mass and energy in developing optimal process designs. Although these two branches are related and indeed occur simultaneously in most processes within the chemical industries, the focus of this work was on Energy Integration as applicable to water use and power generation.

Many researchers have over the years pointed out Process Integration as having arisen due to the energy crisis experienced in Europe and the western countries during the 1970s and early 1980s (e.g. Linnhoff, 1994; El-Halwagi & Spriggs, 1998; Smith, 2000; Hallale, 2001; Kemp, 2007 and Coetzee & Majozi, 2008). In those early days, the focus was on developing techniques for solving and optimising heat exchanger networks but these techniques have since been extended into new areas which include raw materials efficiency, multiple resource management and integration, emissions reduction, process operations and finance. Some of these new areas are still subjects of active research and the methodologies developed are yet to gain the same acceptance that energy and Mass Integration have enjoyed over the years. Thus, this review does not refer to these new methods again and the focus is only on mass, energy and Property Integration with a specific emphasis on Energy Integration - the area on which this research work was based.

2.5 Theoretical techniques of Process Integration

Unlike the traditional approaches to process design and development (which include brainstorming, heuristics and adopting or evolving earlier designs) which generally

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have a number of limitations (e.g. not enumerating the infinite alternatives, being time and money intensive), Process Integration eliminates any such limitations through the application of a systematic and integrative methodology with its three key components (El-Halwagi, 1998; 2006):

- Synthesis
- Analysis and
- Optimisation

These three techniques are generally applicable to any process and transcend the specific process challenges. They are discussed in turn below.

2.5.1 Process synthesis

Process synthesis is the first step in Process Integration and involves combining and integrating process units and streams to meet certain objectives. It therefore starts with developing a clear definition of the goals that the process will seek to achieve. The process then involves generating and screening of various process alternatives, technologies, operating conditions and configurations (El-Halwagi, 1998). The outcome from process synthesis is a flowsheet which represents the configuration of the various pieces of equipment and their interconnection. Two main approaches to process synthesis exist and these are classified based on the manner in which the process is synthesized. These are the structure independent approach and the structured based approach (El-Halwagi & Spriggs, 1998).

2.5.1.1 Structure independent approach

The structure independent approach, also referred to as targeting, refers to the identification of performance benchmarks ahead of detailed design (El-Halwagi, 2006). It is based on tackling the synthesis task via a sequence of stages and within each stage a design target can be identified and employed in subsequent stages. Such targets are determined ahead of a detailed design and without the commitment to the final system configuration (El-Halwagi, 1998; 2006).

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2.5.1.2 Structure based approach

The second approach to the generation and selection of alternatives involves the development of a framework that includes in it all potential alternatives. A typical example is the use of superstructures (Biegler *et al.,* 1997 and Floudas *et al.,* 1986). During the use of this approach and its corresponding analysis, mathematical programming is used to develop a flowsheet showing the configuration of the process equipment. Mixed integer nonlinear programs (MINLP's) are typically used in the approach and the method has the main drawback that if the initial system representation does not include all the possible potential alternatives, then a sub optimal configuration could be the result (El-Halwagi, 2006). The other disadvantage is that common to all nonlinear programs for which obtaining a global solution is never guaranteed.

2.5.2 Process analysis

Process analysis is the second step in a Process Integration exercise and as opposed to process synthesis; it aims at decomposing the process flowsheet into the individual elements in order to determine the element performance. Thus, after a process has been synthesized, its detailed characteristics which typically would include flowrates, compositions, temperatures and pressures, are determined using process analysis techniques (El-Halwagi, 1998). Analysis methods involve predicting the performance using mathematical models and empirical correlations including the use of computer simulation tools. Development of pilot plants and even use of data from similar existing facilities together add to the use of analysis tools.

Complete knowledge of the fundamental processes and an understanding of the component thermodynamic properties together with any reactions occurring in the individual units are critical at this stage. Accurate stream properties are either predicted or obtained from physical property databanks. Examples of predictive methods for the thermodynamic property determination are the UNIFAC method (Fredenslund *et al.,* 1977), the molecular simulation group of methods and the Lydersen group contribution methods outlined in Reid *et al.* (1988) and Polling *et al.* (2001). These methods estimate the component properties based on the component molecular structures. Physical property databanks for thermodynamic properties

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include the Dortmund databank (DDB) of physical properties. An advantage of using modern simulation tools is that most of these commercial simulation tools are shipped with these databanks as part of the package, which makes the analysis process less laborious.

Mathematical optimisation, described in the following section has also been applied as a process analysis tool.

2.5.3 Process optimisation

In section 2.5.1 it was stated that process synthesis begins with developing a clear definition of the goals that the process will seek to achieve. Thus after a process has been synthesized and its performance analysed, it is necessary to revisit the synthesis objectives to determine if these have been met (El-Halwagi, 1998). If these objectives are not met, then the synthesis and analysis activities are iterated until the objectives are met. According to El-Halwagi (1998), this realization of the Process Integration objectives does not guarantee that the process is an optimal one, and therefore process optimisation is included in a comprehensive Process Integration methodology.

Process optimisation is carried out following the principles of mathematical optimisation - the area which is also referred to as numerical optimisation, non linear programming or mathematical programming, and may be defined as the science of determining the best solutions to mathematically defined problems, which may be models of physical reality or of manufacturing and management systems (Snyman, 2005). The degree of goodness of the solution is quantified using an objective function which is either minimised or maximised. The objective function will thus be an expression of the process performance index and thus its value at the end of the optimisation will indicate how good or bad the solution is. For this work, the performance index for the process shown in Figure 1-1 (see Chapter 1) is presented and explained in more detail later in this thesis (see Chapter 3).

The optimisation component of Process Integration allows one to iterate between synthesis and analysis towards the optimal solution and in many instances, as already highlighted above, optimisation is also used within the synthesis and analysis activities (El-Halwagi, 1998).

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The three elements of Process Integration, process synthesis, process analysis and process optimisation have over the years become accepted as a way of improving process efficiencies for new process designs. It is with new designs where the greatest of Process Integration benefits can be realized. The same methods have however also been successfully applied to existing facility revamps and it is in both cases that this work is geared - the aim first being, as with all retrofit designs, to identify opportunities for improved performance and later in synthesizing an optimum process.

2.5.3.1 Advantages of process optimisation

El-Halwagi (2006) presents the following as the advantages of formulating Process Integration and design tasks as optimisation problems:

- Providing for a better and effective method for solving large problems for which graphical and algebraic techniques become ineffective and unwieldy.
- The optimality conditions that can be obtained from parametrically solving an optimisation problem can allow for use in developing solutions for visualisation and algebraic methods.
- Better understanding of the system being solved or optimised is obtained as the inherent nature of optimisation requires one to describe the problem using accurate mathematical relationships, and thus have significant understanding of the problem.
- Sensitivity analysis and other permutations are easily evaluated with the optimisation methods once the initial task of developing the computer model has been completed.

El-Halwagi (2006) also presents some of the disadvantages of mathematical optimisation and states the key challenge as being the identification of the global optimum point in problems with multi optimisation variables. The other disadvantages are the failure of the method in shedding light into the system characteristics and overall insights and also the difficulty in embedding the designer's insights and

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preferences into the entire mathematical optimisation process, when compared to, say, the graphical techniques of Process Integration.

This work focuses mainly on optimisation and thus the following section will present in more detail, the principles of mathematical optimisation. Apart from it not being the intention of the section of the thesis, it will certainly not be possible to present a complete analysis of the huge subject of mathematical programming. Only the concepts necessary for understanding the work detailed in this thesis are presented and the reader is therefore referred to excellent texts and reviews on the subject by Wismer & Charttergy (1978), Papalambros & Wilde (1998), and more recently by Grossmann *et al.* (2004) and Snyman (2005).

2.5.4 Principles of mathematical optimisation

According to Snyman (2005), mathematical optimisation is the process of:

(i) the formulation and

(ii) the solution of a constrained optimisation problem of the general mathematical form:

$$\underset{w.r.t.x}{\text{minimise}} f(x), \boldsymbol{x} = [x_1, x_2, ...x_n]^T \in \Re^n \tag{2-9}$$

subject to the constraints:

$$\begin{aligned} g_j(x) &\le 0, \; j = 1,2,.........m \\ h_j(x) &= 0, \; j = 1,2,.........r \end{aligned} \tag{2-10}$$

Maximisation problems are handled in the above form by noting that:

$$\underset{w.r.t.x}{\text{maximise}} f(x) = \underset{w.r.t.x}{\text{minimise}} - f(x) \tag{2-11}$$

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where $f(x)$, $g_j(x)$ and $h_j(x)$ are scalar functions of the real column vector x. The components x_i of $\boldsymbol{x} = [x_1, x_2, ..., x_n]^T$ are called the design variables, $f(x)$ the objective function, $g_j(x)$ and $h_j(x)$ the inequality and equality constraint functions respectively. The optimum vector x that solves the problem as defined in Equation 2-9 above is denoted by x^* and the corresponding optimum function value by $f(x^*)$

When the constraint functions do not exist or are not specified, the problem is called an unconstrained optimisation problem.

2.5.4.1 Classification of mathematical optimisation problems

Optimisation problems are classified based on the problem type and the classification is independent of the solution methods as shown in Figure 2-15 below (Grossmann & Biegler, 2004).

The first and the main level of classification is in terms of the variables used in the problem, i.e. continuous and discrete variables.

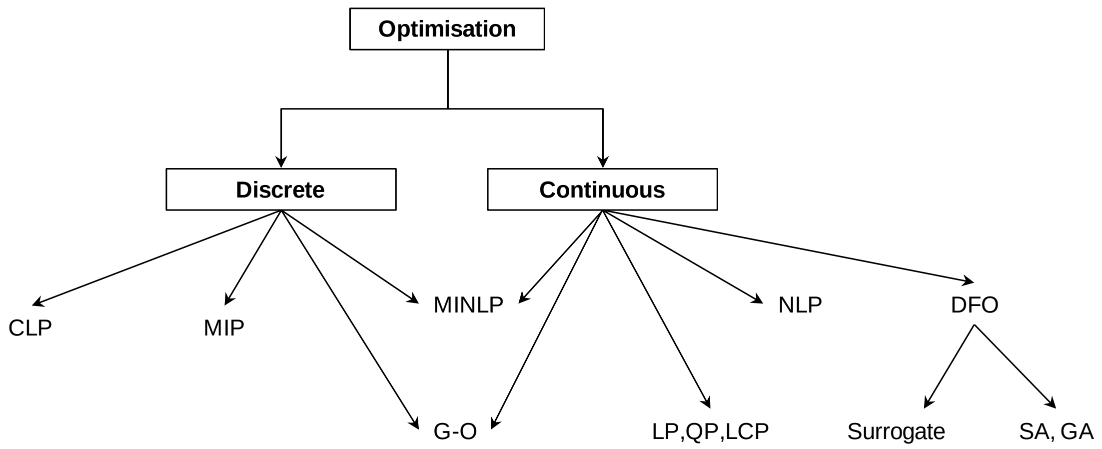

Figure 2 - 15, Tree of classes of optimisation problems (Grossmann & Biegler, 2004)

For the continuous class of problems, the major subclasses include the linear programs (LP) and nonlinear programs (NLP) depending on whether the objective

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function and constraints are linear or nonlinear, respectively. Special subclasses exist for both LP and NLP and these are linear complimentary program LCP for LP and quadratic programming (QP) for NLP. Other distinction methods used for NLP's include whether the NLP is convex or non convex and whether the problem is assumed to be differentiable or not (Grossmann & Biegler, 2004).

The major classes for the discrete problems are the mixed integer linear programs (MILP) and the mixed integer nonlinear programs (MINLP) depending on the linearity of the objective function and constraint functions. These together with the global optimisation (G-O) problems use both continuous and integer variables. A special case when all the variables in the mathematical program are integers gives rise to an integer program (IP).

In most Process Integration mathematical optimisations, MINLP is used with the integer variables typically corresponding to the existence or absence of certain technologies and pieces of equipment in the solution. Continuous variables in such a formulation will indicate the optimal values of non discrete design and operating parameters such as flowrates, temperatures and pressures (El-Halwagi, 2006).

In addition to the classification shown in Figure 2-15 above, Grossmann & Biegler (2004) further present a review of the solution methods of the major types of optimisation problems for both continuous and discrete variable optimisation with an emphasis on methods for nonlinear and mixed integer nonlinear programs.

The other continuous optimisation method presented by Grossmann & Biegler (2004) is the Derivative Free Optimisation method (DFO), where, as the name suggests, derivative information is not required in the solution. They site one of the advantages of this solution method being easy implementation and requiring little knowledge of the optimisation problem. Simulated annealing (SA), Genetic Algorithms (GA) and surrogate models all fall under DFO.

In a way of illustrating the applicability of the different models, Grossmann *et al.* (2004) present the information in Table 2-2 below which shows the application in process systems engineering. They conclude that for the design and synthesis cases, NLP and MINLP dominate the optimisation programs while operations problems are dominated by LP and MILP.

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Table 2 - 2, Application of mathematical programming in process systems engineering (Grossmann & Biegler, 2004)

	LP	***MILP***	***QP,LCP***	***NLP***	***MINLP***	***Global***	***SA/GA***
Design and Analysis							
HENS	x	x		x	x	x	x
MENS	x	x		x	x	x	x
Separations		x			x		
Reactors	x			x	x	x	
Equipment Design				x	x		
Flowsheeting				x	x		x
Operations							
Scheduling	x	x			x		
Supply Chain	x	x			x		
Real tine optimisation	x		x	x			
Control							
Linear MPC	x						
Nonlinear MPC			x	x		x	
Hybrid		x		x	x		

Presented below is the general form of the MINLP similar to that shown in Equation 2-9 for continuous variable optimisation. MINLP combines both discrete and continuous variables and has the general form:

$$\underset{\text{w.r.t.x,y}}{\text{minimise}} \quad f(x,y) s.t. \begin{cases} h(x,y)=0 \\ g(x,y)\le 0 \\ x\in \boldsymbol{x}=[x_1,x_2,.....x_n]^T, y\in[0,1] \end{cases} \qquad \text{(2-12)}$$

where $f(x,y)$ is the objective function, $h(x,y)=0$ are the equality constraints and $g(x,y)\le 0$ are the inequality constraints, x represent the continuous variables and y the discrete variables which are restricted to take either of the values of 0 or 1.

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2.5.4.2 The mathematical modelling process

Solving real world problems using mathematical optimisation is a cyclic process illustrated in Figure 2-16 below for continuous variable optimisation (Snyman, 2005). The process followed is the same for discrete/ mixed integer variable optimisation.

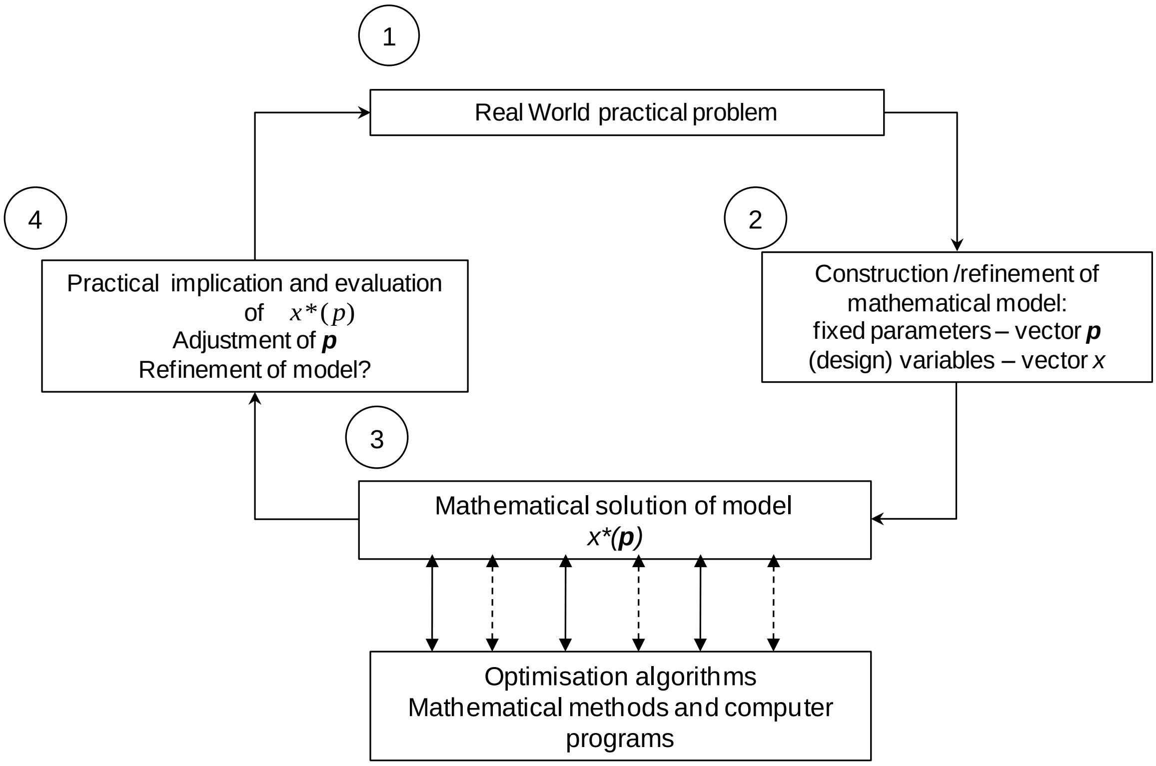

Figure 2 - 16, The mathematical modelling process (Snyman, 2005)

The process shown in Figure 2-16 above has four steps the first being the studying and understanding of the real world situation followed by construction of a mathematical model described in terms of fixed model parameters p , and variables x . The third step in the modelling process is the solution of the mathematical model which gives a parameter dependent solution vector $x^*(p)$. The last step is the evaluation of the solution and its practical implications (Snyman, 2005). The process does not normally end after the fourth step but often it is necessary to adjust the parameters and refine the model giving rise to a new model that needs solution and hence the whole process starts all over again. The cyclic process is continued a

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number of times until an acceptable solution or a predetermined level of accuracy has been reached.

2.6 Categories of Process Integration

In Section 2.4, it was stated that Mass and Energy Integration are both branches of Process Integration and deal respectively with the flow of mass and energy in developing optimal process designs. These two main branches of Process Integration are presented in turn in more detail below. The two categories of Process Integration are followed by a third and relatively new category known as Property Integration, which allows for the optimisation of processes and conservation of resources through the tracking of properties (El-Halwagi, 2006).

2.7 Energy Integration

Heat is the most important and most common form of energy found in chemical and allied process industries including power plants and as such Energy Integration will not be a complete subject if it does not include heat integration. Indeed as has already been highlighted previously, Energy Integration owes its existence to the heat form of energy and as such most of the Energy Integration techniques developed over the years focused on heat optimisation. Of particular interest are the thermal pinch techniques, methods that have found importance in Energy Integration and are used to identify minimum heating and cooling requirements of processes (El-Halwagi, 1998).

The chemical and allied process industries however, do not only require heat but also require power as an energy source. This power is normally supplied in the form of electricity and apart from it providing lighting and its general use in buildings, it is used to drive process equipment including pumps, fans and compressors, and is also used even in the generation of heat in electric heating and tracing.

Electricity was first generated in 1821 by Michael Faraday in his experiments in electromagnetism. In his work he discovered that an electric current flowed in a wire when moved across a magnetic field, the principle that he later used to develop an

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electric dynamo which marked the birth of the modern power generators. Since then, various methods have been developed for producing electricity including hydroelectricity and tidal power arising from renewable energy sources. In most cases, which are still of particular interest in the modern world, the majority of electrical power is generated from heat engines (Kemp, 2007).

A heat engine is defined as a device that converts heat into power. The heat that is used to produce electricity in these engines is normally provided by burning carbonaceous fuels including coal, natural gas, biomass, and other combustible materials. In nuclear power plants, the heat is supplied by the highly exothermic nuclear reactions. Other types of heat engines include the internal combustion engines where liquid and gaseous fuels are combusted producing electricity in the process.

In the power plants that are the focus of this work, the heat produced from the burning of the fuels is used to generate high pressure steam which is in turn used to produce electricity in a turbine. An example of such a power plant was presented in Chapter 1 (See Figure 1-2).

Heat engines, typically the ones described in the foregoing paragraphs, have very low thermal efficiencies, generally less than 40%. The low efficiency is a consequence of the second law of thermodynamics which states that no apparatus can exist in such a way that its only effect is to convert heat absorbed by a system completely into work (Smith *et al.,* 2005). The upper limit of the efficiency of such a heat engine is that of the Carnot Engine, which is defined as a completely reversible heat engine. However, even for the Carnot engine, the thermal efficiency does not reach 100% and this means that heat is always rejected to the surroundings by a heat engine. Instead of wasting this heat, processes have been developed that harness this energy and this has given rise to the concept of combined heat and power (CHP), (Kemp, 2007). CHP systems are systems that have been designed to supply power and useful heat to meet the process heating requirements. A lot of research work has been conducted over the years in CHP systems and continues up to the present day (e.g. Grossmann, 1985; Colmenares & Seider, 1989; Rodriguez-Toral *et al.,* 2001). CHP systems are also particularly important in Europe and other places in the world that get extremely cold in winter where low level heat from the processes is used in district heating.

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Heat and power as described in the foregoing paragraphs are the main forms of energy found in Energy Integration and the remainder of this section focuses on these. First the principles of Pinch Technology are presented focusing on heat integration and this is followed by combined heat and power integration. Further reading and excellent reviews on the subject are presented by Linnhoff (1994), Smith (2000), Hallale (2001) and Kemp (2007).

2.7.1 Pinch technology

Pinch technology emerged as a tool for design and optimisation of heat exchanger networks in the 1970's and can be defined as a systematic methodology for energy saving in processes and total sites that is based on thermodynamic principles. However, according to Linnhoff (1994) and as highlighted previously, the mode of application of Pinch technology has since been broadened, with targeting, rather than design being the major application. Because of its use in process analysis, Pinch Technology has increasingly been referred to as Pinch Analysis rather than Technology (Linnhoff, 1994). As such, from this point onwards, the two terms will be used interchangeably.

Presented in Figure 2-17 below is the Pinch design philosophy which shows the four main sequential steps involved in pinch analysis as:

- Data extraction
- Analysis
- Design and
- Selection of alternatives.

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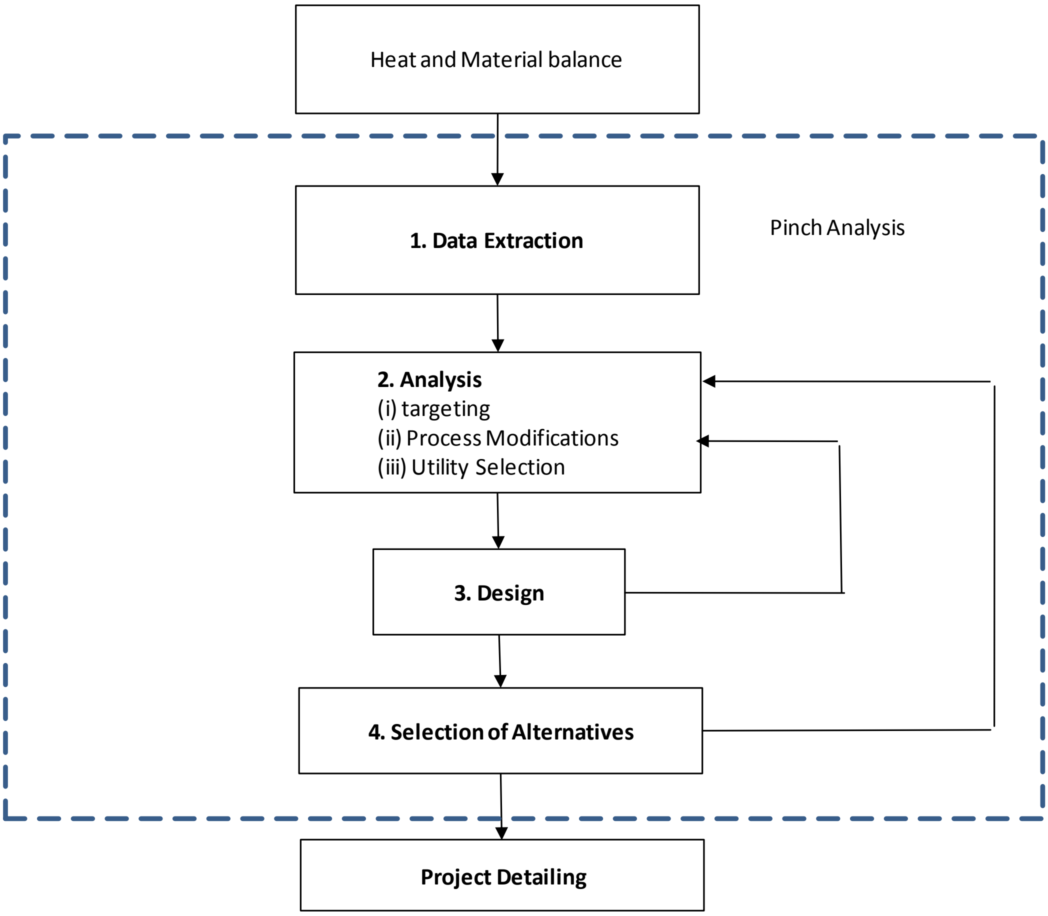

Figure 2 - 17, The Pinch design philosophy

In Figure 2-17 above, the input into pinch analysis are the process material and heat balances and these are the product of the normal process design stage, which is illustrated by the famous onion diagram showing the hierarchy in process design (see Figure 2-18 below). Designing a process begins by focusing on the reactor which gives the reactor product compositions and feed requirements. These then determine the separation system to be employed for separating the reactor products from by-products and unreacted feed streams. Ultimately the various heating and cooling duties for the individual process streams (material and heat balances) are determined and only then, the heat exchanger network (HEN) design can begin as shown in Figure 2-17. The last step in the design process is the specification of the utilities needed for the process. Figure 2-18 shows the design process as proceeding from the inside to the outside of the onion diagram -clearly showing the hierarchy in which the design is carried out.

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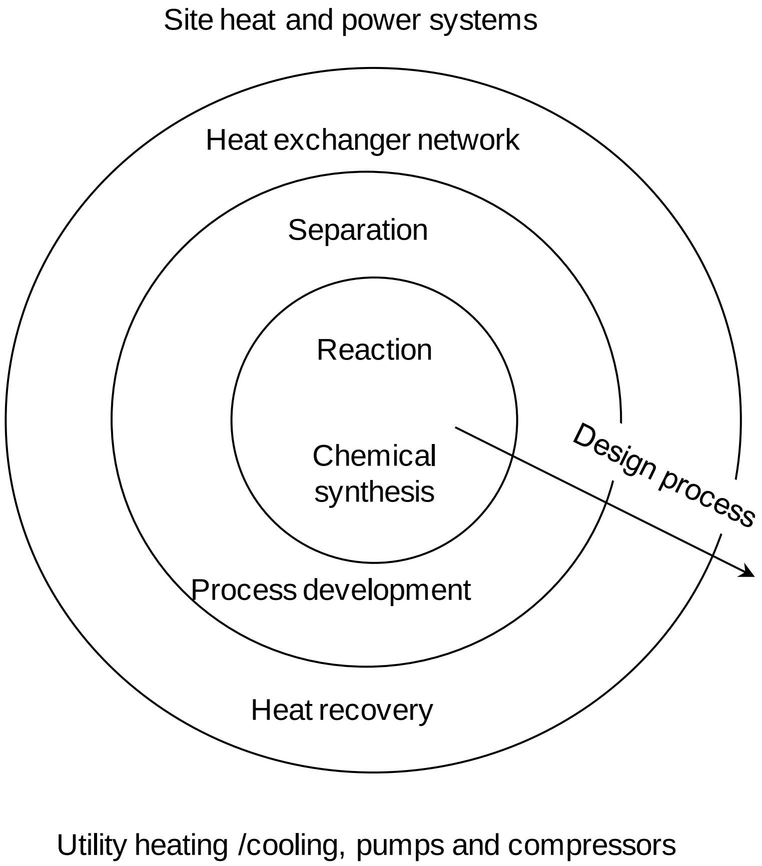

Figure 2 - 18, The Onion diagram for process design (Linnhoff *et al*., 1982)

The discussion in the preceding paragraph on material and energy balances is mostly applicable to new designs, where these balances are easily calculated from design data on flow rates and compositions. For existing plants, the process is no longer straight forward as in most cases design data is usually unavailable and when available it is often significantly different to actual plant performance (Kemp, 2007). This is usually due to changes from the initial design due to adjustments in flowrates and operating conditions normally during start up and the subsequent ramp up to stable plant operation. Furthermore the fouling in heat exchangers and equipment including changes in stream compositions adds to the disparity between the design and actual operating plant data. Kemp (2007) concludes that in such a case, a new heat and mass balance will need to be conducted reflecting the current performance of the plant preferably indicating ranges for different feedstock and conditions for before and after equipment cleaning (of say heat exchangers).

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Pinch Analysis begins by data extraction, which is the process of extracting information required for Pinch analysis from a given heat and material balance. This process involves extracting cold and hot streams information in the form required for pinch analysis. This is an important step in pinch analysis as it determines if one will end up with realistic targets which can be achieved in real equipment or not. The starting point is determining the hot and cold streams from the mass and heat balances. One criterion for determining a stream is that it should change in heat load but not in composition (Kemp 2007). The required stream data available at the end of data extraction will include the stream temperature ranges $(T_1 \longrightarrow T_2)$ where the subscripts 1 and 2 indicate the initial and final conditions, respectively, the stream type as either being hot or cold and the heat capacity flowrate or the stream heat load defined by the below two equations respectively:

$$CP(kW/K) = \text{mass flowrate}\, F(kg/s) \times \text{specific heat capacity}\, C_P(kJ/kgK) \tag{2-13}$$

$$Q = \int_{T1}^{T2} CPdT = CP(T_2 - T_1) = \Delta H \tag{2-14}$$

dT in the last equation is the differential temperature change and the units of the heat load are kW.

The second step in the Pinch Technology process is “Analysis” and this has three main sub steps: targeting, process modifications and utility selection. Targeting, the main step during analysis is based on the pinch principle; which was pioneered by Hohmann (1971) and later extended with great vigour by Linnhoff, Mason & Wardle (1979) and Linnhoff & Hindmash (1983). The pinch principle is founded on minimum temperature difference between the hot and cold streams, $\Delta T_{\min}$ and the fact that a pinch is obtained between the two streams whenever the temperature difference between the streams equals $\Delta T_{\min}$. When several hot and cold streams are involved, one gets composite curves upon combining separately the hot and cold streams in a temperature - enthalpy profile (See Figure 2-19 below). A pinch (Linnhoff *et al.* 1979) is obtained at the point of closest approach of the two composite curves. The pinch divides the system into two distinct thermodynamic regions and the region above the pinch can be considered a heat sink, with heat flowing into it, from the hot utility, but

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not out of it while below the pinch the converse is true, with heat flowing out of the region (Sinnot, 2005).

The region between the two composite curves shows the area where process – process heat integration can occur and is a function of the minimum temperature difference $\Delta T_{\min}$. On the other hand the regions outside of the process- process heat integration zone give the minimum heating and cooling duties for the selected $\Delta T_{\min}$. These are the energy targets for the system and when the system is designed correctly, these are the quantities of the heating and cooling duties that will be needed. Any design requiring more than these targets will not be the optimum one.

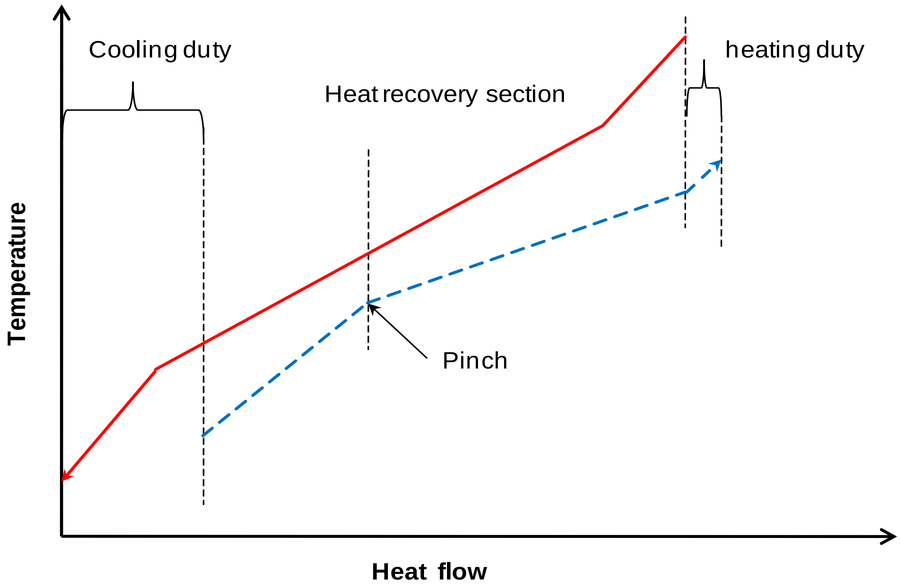

Figure 2 - 19, A typical composite curve for a four stream design problem

An alternative targeting procedure exists which is an algebraic method and thus dispenses with the drawing of composite curves and their manoeuvring to determine the pinch point as required in the graphical technique. The algebraic method, also known as the Problem Table method by Linnhoff & Flower (1978) has the advantage of handling a large number of streams for which with the graphical method, the entire process becomes unwieldy.

The application of the method begins by converting process streams to their shifted temperatures by subtracting half the minimum temperature difference from the hot stream temperatures, and by adding half to the cold stream temperatures:

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For hot streams:

$$T^* = T_{act} - \frac{\Delta T_{\min}}{2} \tag{2-15}$$

For cold streams:

$$T^* = T_{act} + \frac{\Delta T_{\min}}{2} \tag{2-16}$$

where T^* is the shifted temperature.

The process streams are then grouped into intervals with the intervals ranked in order of magnitude while showing the duplicated temperatures only once. A heat balance is then carried out for the streams falling in each interval after which the heat surplus from each interval is cascaded from one interval down to the next interval. The presence of a negative heat flow indicates that the temperature gradient is in the wrong direction and that such an exchange is not thermodynamically possible (Sinnot, 2005). As such, all negative heat flows are reduced to zero by adding just enough heat to the top of the cascade to eliminate the largest negative heat flow. Figure 2-20 below shows a typical heat cascade for a four stream problem before and after elimination of negative heat flows.

Interval temp				
	0 kW		50	
175				
	-90 kW		-90	
145		90 kW		140 kW
	2.5 kW		2.5 kW	
140		87.5 kW		135.5 kW
	137.5 kW		137.5 kW	
85		-50 kW		0 kW
	-60 kW		-60 kW	
55		10 kW		60 kW
	30 kW		30 kW	
25		-20 kW		30 kW
	(a)		(b)	

from (b), pinch occurs at interval temperature = 85^{o}C

Figure 2 - 20, A typical example of a heat cascade (Linnhoff *et al.*, 1982)

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In the energy cascade shown in Figure 2-20 above, the heat introduced to the cascade (50 kW) is the minimum hot utility requirement and the heat removed at the bottom (30 kW) is the minimum cold utility required.

Another composite curve, called the Grand Composite Curve (GCC) is easily constructed from the information in the energy cascade from the Problem Table method. It is a graph of net heat flow against shifted temperature and represents the difference between the heat available from the hot streams and the heat required by the cold streams at any given shifted temperature. The GCC is extremely useful in screening multiple utilities in order to minimise the cost of utilities by ensuring that at each level the use of the cheapest utility is maximised while ensuring its feasibility (El-Halwagi, 2006). Such use of the GCC is illustrated in Figure 2-21 below, where high temperature and low temperature steam together with the cold utility are targeted.

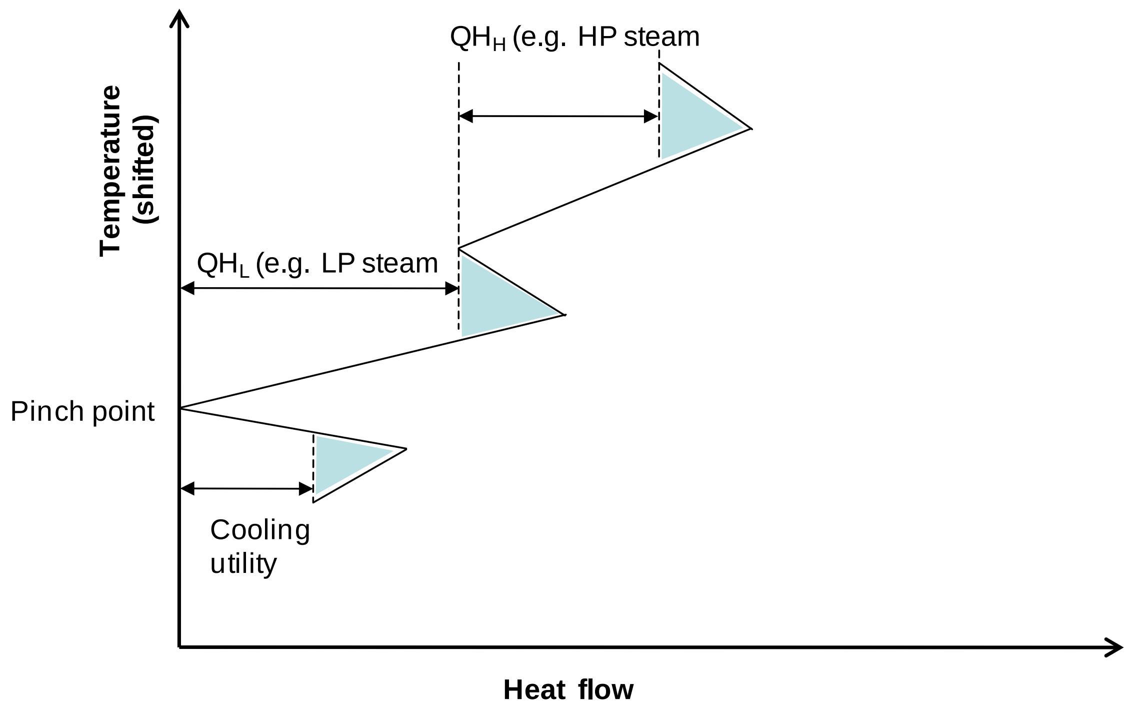

Figure 2 - 21, A typical Grand Composite Curve showing process –process heat exchange pockets and the placement of different utilities (El-Halwagi, 2006)

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The next step in Pinch Analysis after targeting is the heat exchanger network (HEN) design. Of all the proposed methods and tools of HEN design, the grid diagram method of Linnhoff & Flower (1978) is the best and most popular. In a grid diagram, the streams are represented by straight lines with high temperatures on the left and hot streams on top. Any heat exchanger match between two streams is shown on the diagram by two circles joined by a vertical line, as shown in Figure 2-22 below.

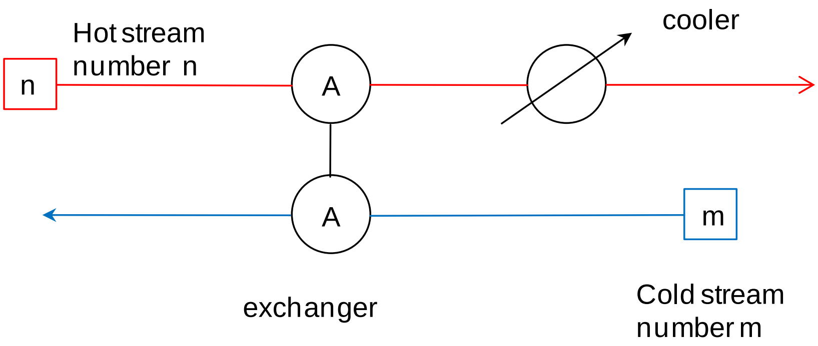

Figure 2 - 22, A typical grid diagram

A number of design principles for maximum energy recovery were developed by Linnhoff (Linnhoff & Flower, 1978; Linnhoff, 1979) and these have also been presented by a number of authors including Linnhoff & Hindmash (1983), Linnhoff (1994), Sinnot (2005), El-Halwagi (2006) and Kemp (2007). The reader is referred to these excellent reviews and texts on the subject.

The design process starts at the pinch and then moving away from the pinch, with each side of the pinch being designed separately. According to Kemp (2007), the basic elements of the pinch design method are:

- Starting the design at the pinch and moving away
- Immediately adjacent to the pinch, the following constraints should be obeyed:

  For all hot streams $CP_{HOT} \leq CP_{COLD}$ (above the pinch)

  For all cold streams $CP_{HOT} \geq CP_{COLD}$ (below the pinch)

- Maximising heat exchanger loads
- Supplying external heat only above the pinch and external cooling only below the pinch.

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Figure 2-23 below shows a typical completed grid diagram for the heat cascade shown in Figure 2-20.

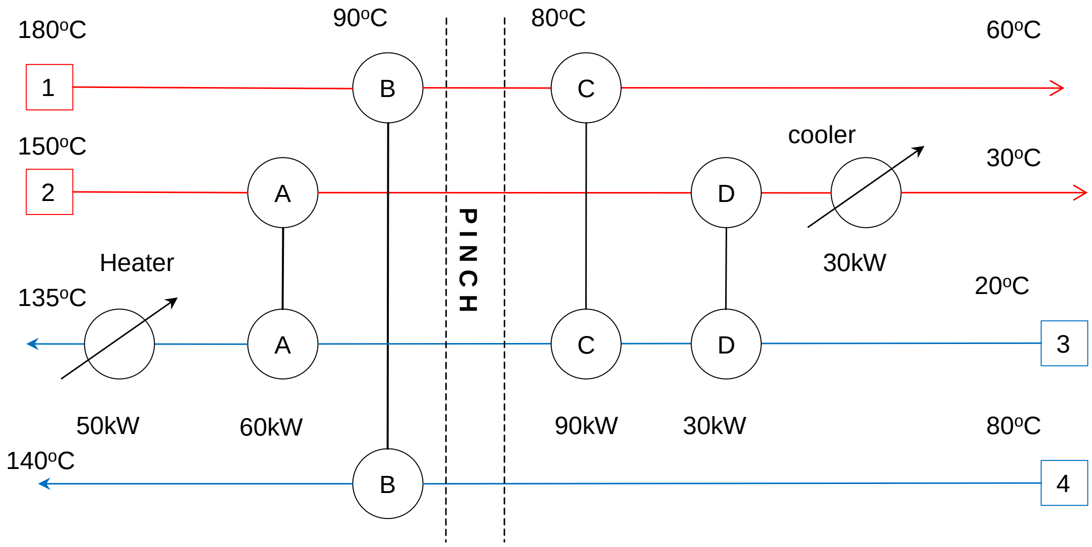

Figure 2 - 23, Typical heat exchanger network using the grid diagram (Sinnot, 2005)

The HEN corresponding to the grid diagram above is that for maximum energy recovery and thus will correspond to minimum utility consumptions. However this will not necessarily be the optimum design for the network. The optimum design will be that which gives the lowest total annual costs and takes into account the capital cost of the system (The number of exchangers in the network, and their size determine the capital cost), in addition to the utility and other operating costs (Sinnot, 2005). The optimum HEN is thus found from determining the total cost for a number of configurations and eventually making a compromise between the capital costs and operating cost. The process of pinch analysis is thus cyclic as shown in Figure 2-17 above and thus will continue until a satisfactory design has been obtained at which point the process of project documentation, which is the last step in the entire Pinch Technology process is carried out.

2.7.1.1 Software for pinch technology

The use of both graphical and the Problem Table methods for Pinch Analysis as outlined above presents challenges as soon as the streams considered begin to

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increase. In a typical process plant, the total hot and cold streams may well be beyond a thousand, making it impractical to use any of the methods. Commercial software is thus available to carry out the targeting and even developing the corresponding heat exchanger network. The most successful commercial applications are Super Target® from KBC (formerly Linnhoff March) and Aspen Pinch®. The advantage of these applications is that they have added capabilities to incorporate capital cost estimation during the optimisation sequence, thus giving better and more realistic results. Aspen Pinch® has for example another added advantage in that the required material balances and stream properties can be extracted from the process simulation, essentially with the click of a button.

2.7.2 Combined heat and power

Combined heat and power (CHP) plants have already been introduced above and these were presented as plants that produce heat as well as electricity. The heat generated can be used for both industrial and district heating purposes and these systems have over the years become an important part of the modern energy conversion process. In this section the common methods of generating CHP are thus presented, methods which have been successfully used to provide the hot utility in Energy Integration. These are the steam turbines, gas turbines and reciprocating engines and are described in turn below.

2.7.2.1 Steam turbines

When steam turbines are used in CHP, high pressure steam is generated in a boiler, similar to the process shown in Figure 1-1 of Chapter 1, and is let down to lower levels (pressures and temperatures) as required by downstream processes through the use of extraction turbines. Figure 2-24 below shows the schematic configuration of an extraction turbine. Useful steam is obtained while producing electricity.

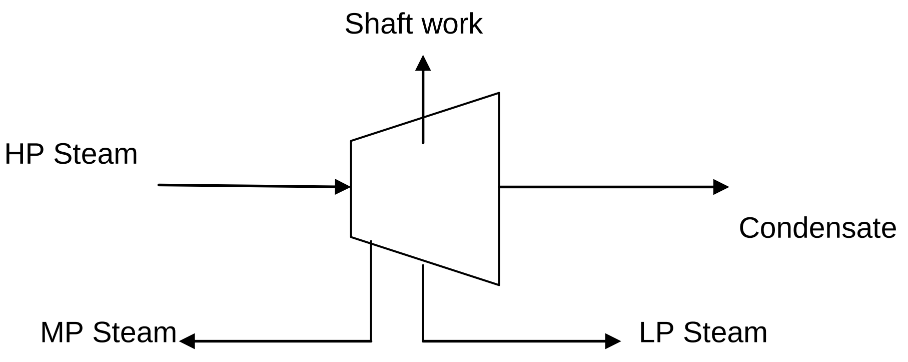

Figure 2 - 24, Extraction turbine layout

2.7.2.2 Gas turbines

In gas turbines, a gaseous fuel, which is usually natural gas, is burnt in compressed air in the combustion chamber of the gas turbine and the resulting gasses at high pressure and temperature pass through a turbine producing shaft work which is either used to drive process equipment or a generator set producing electricity. The exhaust gases can then be used to supply process heating. Figure 2-25 below shows a schematic configuration of a gas turbine.

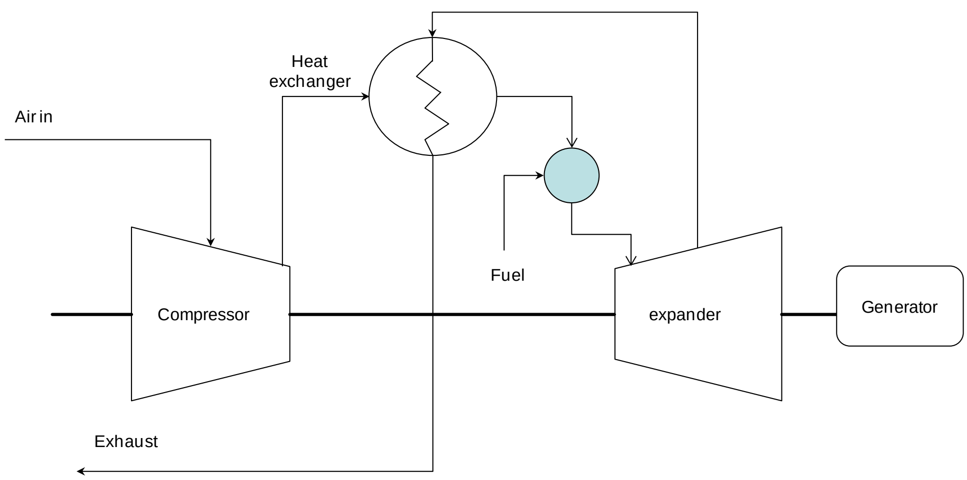

Figure 2 - 25, Gas turbine layout (Kemp, 2007)

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2.7.2.3 Reciprocating engines

These are internal combustion engines and fuel, usually liquid or gas, is combusted producing power in a piston - crankshaft arrangement. As with gas turbines, the exhaust gases can be used as a hot utility. Figure 2-26 below shows a typical reciprocating engine layout.

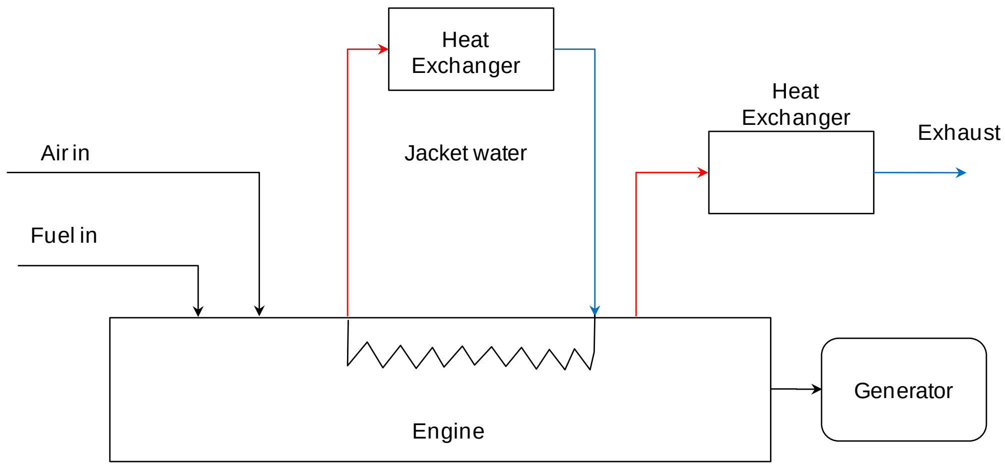

Figure 2 - 26, Reciprocating engine layout (Kemp, 2007)

When the heat released by a CHP system is used as a hot utility, the placement of the heat engine in the process becomes important for effective heat transfer. The "Appropriate Placement Principle" is followed and this requires that the process must be able to make use of all the heat supplied by the heat engine at the exhaust temperature (Kemp, 2007).

Kemp (2007) presents the choice of a particular CHP system as depending on a number of factors including the size and heat to power ratio of the site where it is to be used, the potential for exporting power and the temperatures at which the process requires heat. The GCC is a special tool in the placement of the hot utility in a CHP system. Steam turbines are generally not a good choice when used to supply heat at temperatures above 200°C due to the fact that very little power is produced when the steam is let down to pressures above 15 bar corresponding to a saturation temperature of 200°C. Gas turbines on the other hand release heat in the temperature range 400 to 600°C and are thus suitable for sites with high temperature

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heat requirements. Reciprocating engines also produce exhaust heat at relatively high temperatures (around 400°C) and thus can be used as gas turbines, but are suited for processes with low pinch temperatures and significant heat loads below 100°C.

On a GCC, the exhaust gases from a gas turbine and reciprocating engines are plotted as a varying temperature utility. Figure 2-27 below illustrates the matching for a gas turbine exhaust.

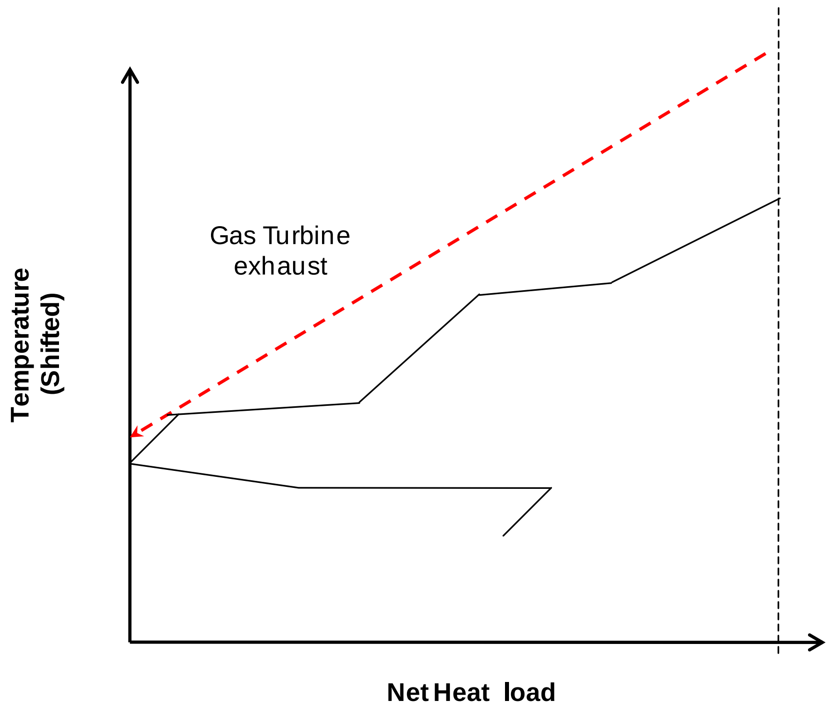

Figure 2 - 27, Matching of a gas turbine exhaust with a process (Kemp, 2007)

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2.8 Historical methods for utility optimisation

The aim of this section is to present the literature review on the historical methods that have been developed over the years for optimising the use of hot and cold utilities in the chemical and allied process industries. Apart from it not being possible, it is not the aim of the section to present all the methods that have been developed, but rather the focus is to present an overview of the most prominent methods illustrating the development over the years. Excellent reviews of the methods for both boiler and cooling systems are available in literature, for example by Grossmann (1985), Biegler *et al.* (1997), Grossman & Biegler (2004) , Kroger (2004), Kloppers & Kroger (2005,a,b) and in El-Halwagi (2006), for which the reader is referred to for detailed analysis of the subject.

2.8.1 Optimisation methods for steam systems

The optimisation methods are discussed in the chronological order in which they appear in the literature.

2.8.1.1 The method of Nishio *et al.* (1980)

It was pointed out in earlier in this chapter that Process Integration arose in the western world during the early 1970's and 1980's due to the energy crises experienced in Europe at that time. Nishio *et al.* (1980) were thus among the pioneers into the optimisation of steam and utility systems (Nishio & Johnson, 1977; 1979)

Nishio *et al.* (1980) presented a method for the optimisation of a steam turbine power plant based on the estimation of thermodynamic losses and irreversibility in the system. They introduced a number of thermodynamic based heuristics for minimising the energy losses in the system. They presented heuristics for the preliminary selection of power generation technologies, with a two-step approach where the problem is classified first as a power dominant or steam dominant problem. They also gave a simple LP algorithm for the allocation of steam turbine drivers.

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In their study, the authors dealt with the problem of designing a steam-power system from the viewpoint of realizing the minimum amount of energy inputs subject to utilizing commercially available units. Thus, they focused on the power plant and not on the downstream users of the produced power and steam utility in their optimisation. It is the authors' understanding that in order to arrive at better and optimum operation of any utility system, complete analysis of the steam system including the heat exchanger network that uses the steam (and power for CHP systems) including the cooling water system servicing the steam turbine need to be considered in a holistic manner.

The other challenge with the method of Nishio *et al.* (1980) was that, like most of the methods developed at that time, it was based on heuristics and as such had the inherent problem of overlooking some of the feasible designs.

2.8.1.2 The method of Papoulias & Grossmann (1983a, b, c)

Papoulias & Grossmann (1983a, b, c) presented a MINLP approach for the optimisation of heat and power for a total site. Their work was undoubtedly the best and most elaborate illustration of the applicability of mathematical optimisation in the synthesis and optimisation of chemical plants. At the time of their work, there was little confidence and faith on mathematical optimisation as a design tool and thus in a set of 3 papers Papoulias & Grossmann (1983a, b, c) showed the importance of the technique and other advantages it provided as opposed to the heuristics and thermodynamic approaches that had then found wide acceptance.
They highlighted some of the challenges with the alternative approaches as not being able to provide a common framework for solving different classes of problems in a systematic manner. The other limitation they pointed was the failure of the heuristic approaches in accounting for the interactions that take place in synthesizing total processing systems that consist of several major components such as utility systems, heat recovery network and chemical plant (Papoulias & Grossmann, 1983a).

Mathematical optimisation was presented earlier as being one of the main parts of Process Integration together with process synthesis and analysis. The advantages

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and disadvantages of the approach were also discussed and the reader is referred to that section for further details.

In their first paper, (Papoulias & Grossmann, 1983a), the authors synthesised a utility system that provides fixed demands of electricity power for drivers and steam at various levels. A superstructure approach as discussed in this early in this chapter that encompassed all of the design options was developed together with the MILP formulation which was solved to give both the structural and parameter optimisation.

In the solution of the MILP model of the utility systems, system state variables like temperature and pressure have fixed variables, which leads to linear constraints that describe the operation of the units such and mass and energy balances. Thus, they came up with different operating conditions and used binary variables to identify the existence of each operating condition. The nonlinear cost objective function, which was the sum of both the fixed and variable costs of the plant, was also linearised, this time by the use of a piecewise linear function.

In the second paper, (Papoulias & Grossmann, 1983b), they discuss heat integration networks. They present from the famous transhipment model formulations from the field of Operations Research, several formulations of the heat recovery networks with the linear programming versions giving the minimum utility cost and the MILP versions giving networks in which the number of units is minimised. In the work, they also tackled the issue of forbidden matches between certain pairs of process streams.

In their third paper, Papoulias & Grossmann (1983c) apply the previous developed models in the design of a total chemical processing system. They present a MILP model for a chemical plant which allows for the synthesis of the plant simultaneously with its heat recovery network and utility system.

Grossmann (1985) also presents a review of the algorithmic approach (MILP formulation) to the synthesis of total processing sites. The author concludes that when coupled with the thermodynamic and heuristic approaches, these, methods can play a useful role in process synthesis providing solutions with reasonable computational expense.

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Although the authoritative work of Papoulis & Grossmann (1983, a, b, c) treats the integrated chemical process plant, it does not consider the cooling water system in any detail. Instead the cooling water temperature is fixed and the model minimised the water usage. It thus, like previous methods, also does not take into account the synergistic effect of the cooling water system in the optimisation of the complete utility system.

The MILP approach of Papoulias & Grossmann (1983a) has also been used for the multiperiod optimisation of utility plants by Hui & Natori (1996) and by Iyer & Grossmann (1997), including the multiobjective approaches for waste minimisation in utility plants by Chang & Hwang (1996).

2.8.1.3 The method of Petroulas & Reklaitis (1984)

Petroulas & Reklaitis (1984) also presented a mathematical programming approach to the synthesis of plant utility systems, by applying a decomposition strategy where they decomposed the design task into two sub problems, one of header selection and the other of driver allocation. Their approach used dynamic programming and linear programming techniques in the synthesis problem.

In their work, the objective was the minimisation of a linear combination of costs. However, they do not consider gas turbines in their work, but focused only on steam turbines and also do not include the capital cost in the optimisation.

As was the case with the work of Nishio *et al.* (1980), and that of Papoulias & Grossmann (1983), Petroulas & Reklaitis also did not consider the cooling water system in their analysis and thus their work suffers from the same shortcomings of not looking at the complete utility system.

Another NLP approach to utility system optimisation was applied by Colmenares & Seider (1989) where utility systems integrated with a chemical process where designed. Their work was based on the temperature interval method with a superstructure of Rankine cycles.

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2.8.1.4 The method of Chou & Shih (1987)

Chou & Shih (1987) proposed a thermodynamic approach to the design and synthesis of plant utility systems, consisting of steam turbines and gas turbines. Their method follows the earlier method of Nishio *et al.* (1980), but differs in that gas turbines are also treated in the analysis. They also indicated scepticism of the mathematical optimisation approach as outlined by earlier researches (Papoulias & Grossman, 1983; Petroulas & Reklaitis, 1984). They site one of the disadvantages as the heavy dependence on mathematical calculations in the optimisation approach and that an inappropriate objective function may even hinder the desired real solution.

In their work, they consider power and heat on the basis of the first law of thermodynamics and thus as being equal forms of energy. This however is not the case when considering the second law of thermodynamics where work is a better form of energy and indeed more useful that heat.

They present a five step procedure for designing the system that starts with the screening of the utility system by the use of the characteristic value of P/H ratio (a ratio of power to process heat), which indicates the complexity of the system.

The method of Chou & Shih (1987), thus suffers from the same drawbacks as that of Nishio *et al.* (1980). The authors also do not consider the cooling water system in any detail and treat the cooling water temperature as fixed. As such they also do not take advantage of the possible improvement of the overall system performance from a comprehensive system treatment.

According to Bruno *et al.* (1998), another shortcoming of their design strategy is that it gives preference to satisfying the heating over the power demands, and back-pressure turbines over condensing turbines and also doesn't give rules for extending the method for the inclusion of electric motors.

2.8.1.5 The method of Maia *et al.* (1995)

Maia *et al.* (1995) presented a simulated annealing approach for the optimisation and synthesis of flowsheets for systems that satisfy fixed demands of steam, electricity and mechanical power.

Their approach allows the selection of equipment available in standard capacities by handling, discrete variables and discontinuous cost functions (Maia *et al.*, 1995). They use a superstructure similar to that of Papoulias & Grossmann (1983a) and compare two annealing schedules by means of a numerical example, where they also demonstrate the importance of considering discrete equipment capacities.
The method of Maia *et al.* (1995) was one of the first attempts to include equipment sizes in synthesis. In their work they do not assume continuous cost functions and show that an error is obtained if such continuity is assumed as in the MILP approach of Papoulias & Grossman (1983). They conclude however that in the case of custom built equipment where costs are continuous functions of capacities there is no advantage of simulated annealing.

Maia *et al.* (1995) uses the same superstructure of Papoulias & Grossman (1983) and thus also do not consider the cooling water system together with the hot utility system.

2.8.1.6 The method of Bruno *et al.* (1998)

Bruno *et al.* (1988) proposed an MINLP synthesis model of utility systems that was based on the approach of Papoulias & Grossmann (1983a), an approach which has already been discussed above. They made improvements with regards to complex steam turbines and determining their efficiency as functions of the load and steam condition and also worked with more precise steam properties. The method of Bruno *et al.* (1998) is based on the same superstructure as that of Papoulias and Grossman (1983a) and thus suffers from the same drawbacks as highlighted above.

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2.8.1.7 The method of Mavromatis & Kokossis (1998a, b)

According to Mavromatis & Kokossis (1998a), the models that had been developed at that time to describe steam turbines were simplistic, with a number of assumptions which included constant efficiencies and not taking into account the capacity constraints of the steam turbines. They thus proposed a more rigorous method for the estimation of power produced by steam turbines. The new method that they termed the turbine hardware model (THM), accounted for the variation of the efficiency of the turbine with size, load and operating conditions. They use the model in setting targets for utility system design and for selecting steam levels ahead of design. They later applied the THM and the conceptual understanding in the development and optimisation of utility networks (Mavromatis & Kokossis 1998b). A three step design methodology is applied which starts by developing a superstructure which is optimised in an MILP formulation.

The method of Mavromatis & Kokossis focuses mainly on the back pressure turbine providing steam for process use in addition to power. It thus, although it provides networks that minimise the use of external cooling water, does not look at the cooling water system in any detail and like all the previous models does not exploit the potential benefits of integrated water and steam system analysts.

The steam turbine model of Mavromatis & Kokossis (1998a) was later extended by Shang (2000), where the model included the effect of the equipment size on the performance of the system.

2.8.1.8 The method of Rodriguez-Toral *et al.* (2001)

Rodriguez-Toral *et al.* (2001) presented an equation oriented (EO) mathematical modelling approach to the optimisation of utility systems. The model is constructed based on balance and performance equations on heat and power systems following the earlier work of Morton *et al.* (1997, 2000). The equations represent the balances (mass, energy and momentum) and performance models for the unit operations in the utility system.

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They presented first EO models for process streams containing water/steam using thermodynamic equations for the physical properties involved in heat and power systems. They then presented two examples of utility systems optimisation (a power plant with two steam turbines and a combined heat and power plant incorporating turbines and compressors), which were built on EO models for each unit operation (heaters, coolers, turbines compressors etc.) and formulate these as a NLP's which were solved using a Sequential Quadratic Programming (SQP) solver.

In their analysis of the utility systems, Rodriguez-Toral *et al.* (2001) also used a fixed temperature for cooling water and thus do not include the cooling systems in any detail. As with the previous researchers their work thus suffers from the same drawback of not investigating the utility system in a comprehensive manner.

2.8.1.9 The method of Varbarnov *et al.* (2004)

Varbarnov *et al.* (2004) presented a MINLP method for the simulation and optimisation of utility cogeneration systems. They presented first improved methods for modelling of steam turbines and gas turbines from previous models (Mavromatis & Kokossis, 1998, Shang, 2000) and later applied these in optimising a utility system. Their models are based on thermodynamic principles and focus on the model accuracy is determining part load performance. The models particularly improved the shortcomings and assumptions in the method of Mavromatis & Kokossis (1998a) as extended by Shang (2000), in accounting for most of the factors influencing the performance of turbines (including turbine size in terms of maximum power load, pressure drop across turbine , turbine load and ambient conditions in gas turbines).
Varbanov *et al.* (2004) also presented a new approach for optimisation of the utility system where the detailed hardware models (MINLPs) were linearised and solved as a succession of MILPs. They applied their models to a case study and conclude that different trends are obtained for different scenarios depending on the marginal cost of steam, with a significant potential for cost savings.

The optimisation method of Varbarnov *et al.* (2004), like that of previous researchers, focuses only on the cogeneration systems as applied to steam and power

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generation. They do not look at the heat exchanger network that uses the generated utilities nor do they look into the cooling water network.

Other recent MILP utility system optimisation methods were by Aguilar *et al.* (2007) and Papandreou & Shang (2008). Aguilar *et al.* (2007) presented a multiperiod formulation for both grassroots and retrofit designs. Papandreou & Shang (2008) proposed a multiobjective optimisation method for sustainable design of utility systems that seeks both economic and environmental goals. They used the superstructure of Papoulias & Grossmann (1983a) and introduced the elements of Life Cycle Analysis (LCA) in the formulation to reduce undesirable emissions like SO_X, CO_2, CO and NO_X from utility generation systems.
Liu *et al.* (2009) also developed an MINLP model for the design and optimisation of polygeneration energy systems, where steam, electricity and other chemical products are produced (e.g. synthetic methanol and dimethyl ether, DME)

2.8.1.10 The method of Coetzee & Majozi (2008)

Coetzee & Majozi (2008) recently reduced the steam requirement for a process without compromising the duty requirements by following the work of Kim & Smith (2001). They presented a hybrid graphical and mathematical technique for targeting and heat exchanger network synthesis, where the series connection of heat exchangers is preferred to parallel connection, thus utilising some of the sensible heat in the steam (Coetzee & Majozi, 2008).
Traditional methods of heat exchanger design use a parallel configuration where each heat exchanger receives steam from the boiler and after condensation, the steam is returned to the boiler after some cooling to reduce cavitation in pumps. It is this residual energy in the condensate that Coetzee & Majozi (2008) exploited and thus the condensate could be used in other heat exchangers following the work of Kim & Smith (2001) and Majozi & Moodley (2007).

The authors first determined the target flowrate by matching the process composite curve with the horizontal and slanted lines for the steam representing the latent heat and sensible regions respectively. A pinch point was obtained when the sensible heat lines touch the process composite curve and thus gave the minimum steam

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requirements. The corresponding heat exchanger network was obtained from a MILP model based on a superstructure which encompassed the possible series/parallel configurations.

The effect of pressure drop in such a network was evaluated by Price (2010).

The work of Coetzee & Majozi (2008) focused on steam usage and minimisations and thus did not look into the cooling water network. It also had the disadvantage that cost was not used in the optimisation. A network that gives the minimum steam consumption is not necessarily the one that brings the best economic results. Their work also did not investigate the possible impact on the associated cooling water network. In a typical plant set up, cooling water is generally used to cool hot process streams (including hot condensate before reuse) and thus using the sensible heat as described by Coetzee & Majozi will reduce the heat load going to the cooling system which may affect the operation of the complete system. Thus, as with earlier researchers their work does not include a comprehensive treatment of the utility system

2.8.1.11 The method of Chen & Lin (2011)

Chen & Lin (2011) developed a steam distribution network for satisfying the energy requirements of a process. They use the transhipment model of Papoulias & Grossmann (1983b) and adapt it to simultaneously consider opportunities for steam usage and steam generation. In their work they use the steam turbine model of Aguiler *et al.* (2007) and modify it to determine the steam header levels and the operating conditions for varying energy demands, which were specified in the original model.

The method of Chen & Lin (2011) is similar to that of previous researchers and treats the cooling water temperature as a fixed parameter and thus does not look at the cooling water network in any detail and therefore also ignores the complementing and synergistic effect of treating the utility system in a comprehensive manner.

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2.8.2 Optimisation methods for cooling water systems

2.8.2.1 The Merkel method (1925)

Merkel developed the theory of evaluation of cooling towers as early as 1925. However according to Kloppers & Kroger (2005a), the work of Merkel was neglected for more than 15 years until 1941 when the work was translated into English. The Merkel method has already been highlighted above and it was indicated that it has widely been accepted as the evaluation method of cooling tower performance (Perry & Green, 1997).

A number of assumptions are made in the Merkel method, which greatly simplify the calculations and the solution for heat and mass transfer in cooling towers. The main assumptions are:

- The Lewis factor which relates heat and mass transfer is equal to 1
- The air exiting the tower is saturated with water and is characterized only by its enthalpy and
- The reduction of water flowrate by evaporation is negligible in the energy balance

Comparison of the predicted water temperature using the Merkel method to experimental values indicated that the Merkel method accurately predicts the cold water temperature (Bourillot, 1983; Bernier, 1994; and Perry & Green, 1997). However, Kloppers & Kroger (2005) indicate that the method is insufficient in calculating the characteristics of the warm air leaving the cooling tower and the changes in the water flowrate due to evaporation. They indicate that these are important considerations which determine the water consumption and also used to predict the behaviours of plumes exiting the cooling tower.

Another inaccuracy introduced by the assumption of not taking account of the water flowrate in the Merkel equation is that the effect of evaporation on the energy balance is ignored (Kloppers & Kroger, 2005a). Barker & Shyrock (1961) investigated the foregoing problem and conclude that the Merkel number increases by up to 4% when the evaporation losses are considered.

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The cooling tower model used in this work, detailed in chapter 3 is a variation of the Merkel method as outlined by Kim & Smith (2001) and indeed the work showed inaccuracies in the Merkel method when calculating the evaporated water despite having accurately predicted the outlet water temperature. As a result modifications were made to the method of Kim & Smith as used in this work. The details of the modifications are also presented in Chapter 3, which gives the mathematical formulation of the models used in this work.

2.8.2.2 The method of Lefevre (1984)

From as early as 1984, Lefevre (1984) investigated a number of ways of reducing water losses from wet cooling towers. He identified the major losses being those emanating from evaporation and blowdown and conceded that not much can be done to reduce the evaporation losses as these are dependent on the ambient conditions and thus recommended the use of a low air flowrate as is possible. He also pointed that in such a case a very efficient fill media would be required.
Lefevre (1984) also investigated the effect of the flow arrangement on the cooling tower performance, i.e. if the crossflow arrangement will have an advantage over the counterflow arrangement. He pointed that because of an increased air requirement in crossflow cooling towers for the same degree of cooling, the crossflow units will thus have more evaporation losses. He however ends by pointing that the difference in the evaporation losses is small and less than 1% is most cases, therefore making the type of flow arrangement not being that important in reducing evaporation losses.
He also investigated the combination of wet and dry cooling towers and concluded that when insufficient make up water supply is available, then a combination of wet and dry cooling towers may be the most practical way of reducing evaporation losses.

Lefevre (1984) in his work however looked only at the cooling water system in isolation of the heat exchanger network. It is the authors' understanding that a complete analysis of the utility system's components including even the heat source is needed to arrive at the best results and benefits in terms of operation efficiency.

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2.8.2.3 The e-NTU method of Jabber & Webb (1989)

Jabber & Webb (1989) developed the effectiveness-NTU method for cooling tower design, akin to that used in standard heat exchanger design. Their method is applicable to both counter flow and cross flow cooling towers. They showed that for a cooling tower:

$$\frac{d(i_{masw} - i_{ma})}{(i_{masw} - i_{ma})} = hd\left(\frac{di_{masw}/dT_w}{m_w c_{pw}} + \frac{1}{m_a}\right)dA \qquad (2\text{-}17)$$

which is similar to the heat exchanger e-NTU equation:

$$\frac{d(T_h - T_c)}{(T_h - T_c)} = -U\left(\frac{1}{m_h C_{ph}} + \frac{1}{m_c C_{pc}}\right) \qquad (2\text{-}18)$$

They define the effectiveness as:

$$\varepsilon = \frac{Q}{Q_{\max}} = \frac{m_w c_{pw}(T_{wi} - T_{wo})}{C_{\min}(i_{maswi} - \lambda - i_{mai})} \qquad (2\text{-}19)$$

and the number of transfer units being:

$$NTU = \frac{1}{1-C}\ln\frac{1-\varepsilon C}{1-\varepsilon} \qquad (2\text{-}20)$$

Where $C = \frac{C_{\min}}{C_{\max}}$ with $C_{\min}$ being the minimum of the air flowrate m_a and $\frac{m_w c_{pw}}{di_s/dT_w}$ and the maximum being $C_{\max}$. λ is a correction factor to improve the approximation of the enthalpy of saturated air i_{masw} and is given as:

$$\lambda = (i_{maswo} + i_{maswi} - 2i_{maswm})/4 \qquad (2\text{-}21)$$

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The Merkel number based on the e-NTU method of Jabber & Webb (1989) is

$$Me_c = \frac{c_{pw}}{di_{masw} / dT_w} NTU \tag{2-22}$$

if m_a is greater than $\frac{m_w c_{pw}}{di_s / dT_w}$ and

$$Me_c = \frac{m_a}{m_w} NTU \tag{2-23}$$

if m_a is less than $\frac{m_w c_{pw}}{di_s / dT_w}$

The e-NTU method of Jabber & Webb (1989) is also based on the same simplifying assumptions of Merkel (1925), but has the advantage over the Merkel method in that it simplifies the solution procedure as it avoids the unwieldy numerical integration necessary for the Merkel method (Kloppers & Kroger ,2005a).

Kloppers & Kroger (2005a) compared the Merkel and the e-NTU methods together with a third method of Poppe (1991) for both natural draft and mechanical draft cooling towers. Amongst their findings they concluded that both the Merkel and e-NTU methods inaccurately predict the outlet air temperatures. They also showed that the outlet water temperatures for all the methods were identical and with the heat rejected, they found that the Poppe method (which does not make any of the simplifying assumptions in the Merkel and Poppe methods) predicts higher heat rejection. This is expected given that the Poppe method does not ignore the water loss in the energy calculation. The impact of assuming a Lewis factor of 1 was also investigated by Kloppers & Kroger (2005a), where they calculated the amount of water evaporated for varying Lewis factors. They concluded that the Lewis factor is important when the ambient temperature is less than about 26 ^{o}C where, there are discrepancies which begin to be evident as the Lewis factor increases.

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Their final conclusion was that the Merkel and e-NTU methods should give good results if the water temperature is the most important parameter in design calculations.

Other authours that presented the effectiveness methods include Khan*et al.* (2003) and Soleymez (2004).

2.8.2.4 The method of Bernier (1994)

Bernier (1994) presented a one dimensional model for a countercurrent cooling tower and uses it to investigate the effect of a number of variables including the water and air flow rates and fill height on the performance of a cooling tower as indicted by the cooling tower characteristics.
He used a water droplet in the cooling tower as a control volume and developed equations for the change in the water temperature from an energy balance around the control volume. Some of Bernier’s findings from the analysis of the control value where that about 90% of the total heat rejection is from evaporation. Also the author found that lowering the wet-bulb temperature has a much increased effect on heat rejection than from lowering of the dry-bulb temperature. Bernier (1994) also showed that the outlet water temperature increases with an increase in the circulating water flowrate for a given heat load and that the cooling tower performance increases with an increase in the air flowrate, or with a reduction in the L/G ratio. This was also shown to be the case by Kim & Smith (2001) and by Papaefthimiou *et al.* (2006). An increase in the air flowrate however has the disadvantage of increasing the electrical costs in mechanical draft cooling towers.

2.8.2.5 The method of Castro *et al*. (2000)

Castro *et al.* (2000) presented a model for a cooling tower system incorporating the heat exchanger network. Their work was different in that it attempted to account for the pressure drop in the network, and that it was also based on correlations to predict the water properties and cooling tower performance. They used cost as an objective function in optimising the cooling tower for given climatic conditions. Apart from the

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general conclusions from previous researchers, they concluded that the humidity of the air and not the ambient temperature itself has the greater effect on the performance of a cooling water system

Other than considering the pressure drop in the heat exchanger network, the optimisation method of Castro *et al.* (2000) does not look in further detail at the heat exchanger network and the impact the changes in the cooling tower operating conditions will have on the process. They assume a fixed heat load from the heat exchangers and work with a fixed process. In practice, changes in the cooling system affect the process and thus complete optimisation of a cooling system can only be achieved by a comprehensive approach that considers the process as well.

2.8.2.6 The method of Kim & Smith (2001)

Kim & Smith (2001) also presented a one dimensional model for a cooling tower which they later used to optimise a cooling water system comprising of the cooling tower, the recirculation system and the heat exchanger network. The optimisation of the complete cooling water system and its associated heat exchanger network was based on the understanding that a reduction in the circulating water flowrate in a cooling tower improves the cooling tower performance for a fixed heat load.

Their method of optimising the HEN was based on Water Pinch principles and a graphical technique was used to arrive at the minimum water requirement for the HEN. In the optimisation they viewed the HEN and the cooling tower in a single approach having realised that the performance of the cooling tower affects that of the HEN and *vice-versa*. This holistic understanding and treatment of the system was applied in this work, where the power plant utility system was optimised in a single approach. The work of Kim & Smith (2001) was one of the first to treat the cooling water system and the associated heat exchanger network in a single approach, with most of the previous research having treated the cooling water network and the cooling tower in a discreet manner without consideration of the entire cooling system components (Kim & Smith, 2001). Such an individual analysis is likely to give suboptimal results.

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The cooling tower model of Kim & Smith (2001) is a version of the Merkel method and indeed makes the same assumptions as Merkel (1925). They developed the model from first principles by taking balances around control volumes, which are later integrated for the whole tower. According the Kim & Smith (2001) their model represents a balance between the simplistic models and the more detailed models which have the drawback of not allowing the system interactions to be investigated reliably. The cooling tower model of Kim & Smith was used in this work. Details of the models are presented in Chapter 3.

2.8.2.7 The method of Papaefthimiou *et al.* (2006)

Papaefthimiou *et al.* (2006) presented a thermodynamic model for a countercurrent cooling tower. Their model is based on energy balances in a control volume inside the cooling tower, ending up with a set of ordinary differential equations for the rate of change of the air temperature, humidity ratio, water temperature and water flowrate along the tower height, equations which are solved numerically along the length of the cooling tower using the Runge-Kutta method. They also investigated the effect of atmospheric conditions, including the inlet water and air temperatures on the performance of the cooling tower.

In their model, they make a number of simplifying assumptions based on (ASHRAE, 1976, Perez-Blanco & Bird, 1984) which are:

- That heat and mass transfer only occur in a direction perpendicular to the walls of the cooling tower.
- There is negligible heat and mass loss through the walls of the cooling tower to the surroundings.
- They ignore the heat that the fans give to the air and water as it passes the fans.
- Constant heat and mass transfer coefficients in the cooling tower.
- A constant Lewis number in the whole of the cooling tower.
- Drift losses are minimal.
- Uniform water temperature across any section in the cooling tower.

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- The cooling tower has a constant and uniform cross sectional area

These are the same assumptions as those made by Merkel (1925) and thus the method of Papaefthimiou *et al.* (2006) suffers from the same drawbacks as that of Merkel.

Like Castro *et al.* (2000), they conclude that the thermal performance of the cooling tower is greatly dependent on the humidity of the inlet air. They also obtained the same findings as Kim & Smith (2001) and Bernier (1994) that the outlet water temperature increases with an increase in the L/G ratio.

The cooling tower method and optimisation of Papaefthimiou *et al*. (2006) has the same setbacks as that of Castro *et al.* (2000) and other earlier researchers in that it only considers the cooling tower and does not mention in any way the heat exchanger network.

2.8.2.8 The method of Panjeshahi & Ataei (2008)

Following the cooling water network optimisation of Kim & Smith (2001), Panjeshahi & Ataei (2008), presented a method of optimising a cooling tower system with the associated heat exchanger network that incorporates ozone treatment of the water.

The authors thus use the same principle as Kim & Smith (2001) that series connection of heat exchangers allows for a reduction in fresh cooling water requirement and also accompanied by an increase in the cooling tower performance.

Unlike Kim & Smith (2001), who worked with a fixed COC value (cycles of concentration which was 3), the authors increased the cycles of concentration (to 15), due to the use of ozone which improved the quality of the circulating water by limiting the growth of mineral and microbial deposits on the heat transfer surfaces. Consequently the blow down and make up from the cooling tower decreased. They reported a 46% reduction in makeup, 93% reduction in blow down and 17% energy saving when their results where compared to those of Kim & Smith (2001).

Their cooling tower model was similar to that of Castro *et al.* (2000) and based on overall material and energy balances coupled with a number of correlations for the water and its properties as given by Kroger (2004). Constraints in the model included

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those obtained from the feasible regions in the graphical pinch analysis as was given by Kim & Smith (2001).

2.8.2.9 The method of Gololo & Majozi (2011)

Gololo & Majozi (2011) recently presented a mathematical technique for the optimisation of a heat exchanger network with multiple cooling water sources. Their work was an improvement of the earlier work by Majozi & Nyathi (2007) and Majozi & Moodley (2008), where instead of the work by Majozi & Moodley (2008), a detailed cooling tower model was incorporated in the minimisation of cooling water required for a fixed duty in the heat exchanger network.

Majozi & Nyathi (2007) had earlier presented a combined graphical and mathematical programming technique using a superstructure for the synthesis of cooling water systems consisting of multiple cooling sources.

Majozi & Gololo (2011) used a Merkel type cooling tower model as presented by Kroger (2004), together with a superstructure based approach for the heat exchanger network design. The minimum circulating water flowrate was found from simultaneous solution of the cooling tower model to determine the cooling tower performance together with the MINLP model for the heat exchanger network.

They showed in their work through different cases that the cooling tower system could be debottlenecked without compromising on the heat exchanger duties. In one of the cases a maximum decrease in circulating water flowrate of 22% was realised. As was the case for the method of Kim & Smith (2001), Gololo & Majozi (2011) showed that better system performance is obtained from a holistic system treatment and analysis as opposed to separate optimisation.

The method of Gololo and Majozi, focused only on the cooling water system and its associated heat exchanger network. In typical industrial utility systems incorporating cooling water networks, steam and hot utility networks are also available and thus, the method of Gololo & Majozi (2011), like those of previous researchers does not guarantee optimality of a complete plant utility system as it considers only the optimisation of the cooling water system.

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2.8.2.10 The work of Leffler *et al.* (2012)

Cooling towers are by far the most common method of heat rejection in process and power plants and thus the preceding sections focused on cooling tower optimisation in power plants. Other widely used methods include lakes and oceans although in recent decades these have met with continued opposition from environmental groups due to possible negative effects on marine life in these water bodies.

Leffler *et al*. (2012) presented five alternative heat rejection methods for power plants and these are; cooling canals, open water algae bioreactors, greenhouse heating, spray ponds and modified solar updraft towers. In the work, the water loss for each heat rejection method is compared to that of a cooling tower in a 500 MW coal power plant and the conclusion is that cooling canals are the worst requiring 22% more water than that evaporated in cooling towers. Spray ponds are comparable to cooling towers (94%) and the algae bioreactors require 76 % of the cooling tower water while the other two methods do not use evaporation and thus no water is lost. Apart from the fact that these alternative methods are not common in the power industry, partly due to the fact that they are climate dependent and again depend on a lot of other factors, the work only focuses on heat rejection and does not consider the impact on the complete power plant (capital and or operating cost, efficiency e.t.c.). The work thus like that of the previous researchers suffers from the same drawback of not considering the power plant utility system in an integrated approach.

2.8.3 Combined boiler and cooling systems

The preceding sections gave an overview of the methods that have been developed over the years for the optimisation of the both cold and hot utility systems. As outlined in the work, despite there being many studies that were conducted on such systems, virtually all treat the utility system components in a discrete manner. Focus is either on the steam system optimisation while ignoring the equally important cooling water system, or vice versa.

Very limited research work is available from the literature that presents a combined and holistic treatment of the utility systems and two notable examples are the work of

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Barigozzi *et al.* (2011) and Gharaiea *et al.* (2013), which are discussed in the following sections.

2.8.3.1 The work of Barigozzi et *al.* (2011)

Barigozzi *et al.* (2011) recently investigated the use of combined wet and dry cooling systems as applied to a combined heat and power plant that uses municipal and industrial solid waste as fuel. In the work, an air cooled condenser is used in parallel with a water cooled condenser with cooling water being supplied from a mechanical draft cooling tower in a closed loop. The work shows that for lower ambient temperatures the best way of heat rejection would be through the use of the air cooler whereas for higher ambient temperatures the condensation should be through the cooling tower as much as possible with any remaining heat being rejected through the air cooler. This conclusion reflects increased operating costs of the air cooler under high ambient conditions since more air is needed to achieve the same degree of cooling as that in the case when ambient conditions are low.
Although the work of Barigozzi *et al.* recognises the cooling system as having an effect on the performance of the power system, it does not look holistically at the performance of both the power and cooling systems. The focus is on the effect of rotational speeds of the cooling tower fan, air condenser fan and cooling water pump on the net power output from the power plant. Maximising the power output from the system does not guarantee efficient operation of the comprehensive system together with its subsystems. The performance of the power system needs to be understood together with how effective the cooling system is. A better and sustainable performance index that takes account of the operation of both the power and cooling systems is required. The work thus focuses on maximising the power output without analysing the effectiveness of the cooling water system.

2.8.3.2 The work of Gharaiea *et al.* (2013)

Gharaiea *et al.* (2013) integrates a site utility system with an IGCC power plant having precombustion CO_2 capture. The work initially looks at the IGCC and utility subsystems separately and later combines these in a comprehensive analysis.

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Although the work looks at the subsystems comprehensively and also addresses the important environmental aspect of CO_2 reduction in power plants, it does not look at the cooling system at all as part of the utility subsystem. The cooling water is accounted for as a cost item and thus the analysis does not give any insight into the performance of the cooling water system. The impact of the IGCC system on the cooling water and vice versa is thus ignored.

The present work is therefore one of the early attempts of combining the cold and hot utility optimisation methods in a single approach. Details of how this was done are presented in the following Chapter, Chapter 3 which gives the methods that were used in this work. The results are presented in Chapter 4.

2.9 Mass Integration

Akin to Energy Integration, which has as its focus on energy, particularly heat and power generation, movement and utilization, Mass Integration deals with the flow of mass in developing optimal process designs. El-Halwagi & Spriggs (1998), define mass as involving the creation and routing of chemical species in reactions and by-product and waste processing systems. They argue that although Mass Integration is analogous to Energy Integration in many ways, Mass Integration has the advantage that it tackles the core process and thus has a more direct and significant impact in determining process performance. They however conclude that mass and Energy Integration complement each other and together form the basis of a comprehensive process design methodology (El-Halwagi & Spriggs ,1998).
Presented here is an introduction to the extensive subject of Mass Integration and for further information and details the reader is referred to excellent reviews and texts on the subject which include Wang & Smith (1994;1995), El-Halwagi (1997), El-Halwagi & Spriggs (1998), Dunn & El-Halwagi (2003) and El-Halwagi (2006).

Figure 2-28 below shows the classes of Mass Integration. The main ones, which are huge subjects in their own right, are the water and the mass pinch methods which were developed following the methods of Pinch Technology described above. The other two, Hydrogen and Oxygen pinch are also further extensions of pinch analysis

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but will not be discussed in this work. More details on these can be found in literature for example in Alves (1999) and in Hallale & Liu (2001).

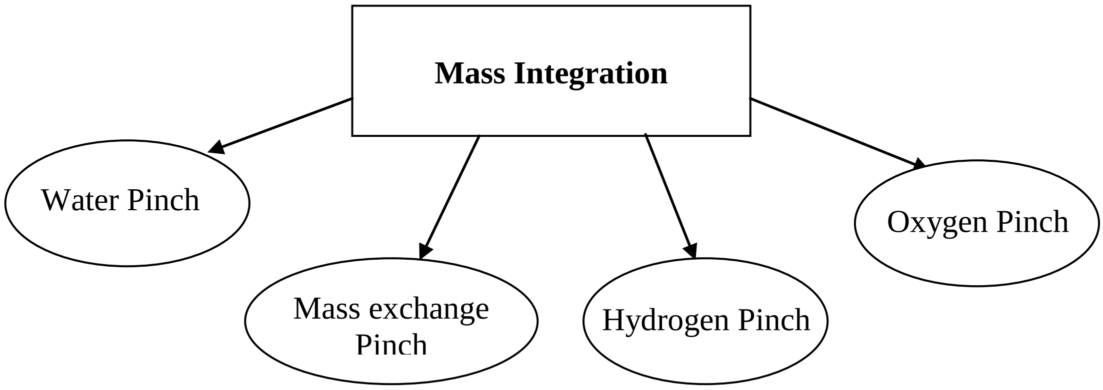

Figure 2 - 28, The classification of Mass Integration

Mass Integration begins with the focus being on the chemical species or components involved and not the unit operations. The first step in performing Mass Integration is representing all the species involved in the process as shown in Figure 2-29 below. For each species under consideration, there are sources and process sinks. Process sources are the streams that carry the respective components and process sinks refer to the units that accept these species and include reactors, bio treatment facilities and heaters and coolers. As seen in the Figure 2-29, streams that leave the sinks become sources and thus become generators of the species under consideration. The compositions of the sources are generally manipulated or changed to those required by the sinks and this is done in a network of separation units referred to as species interception network (SPIN) (El-Halwagi & Spriggs, 1998)

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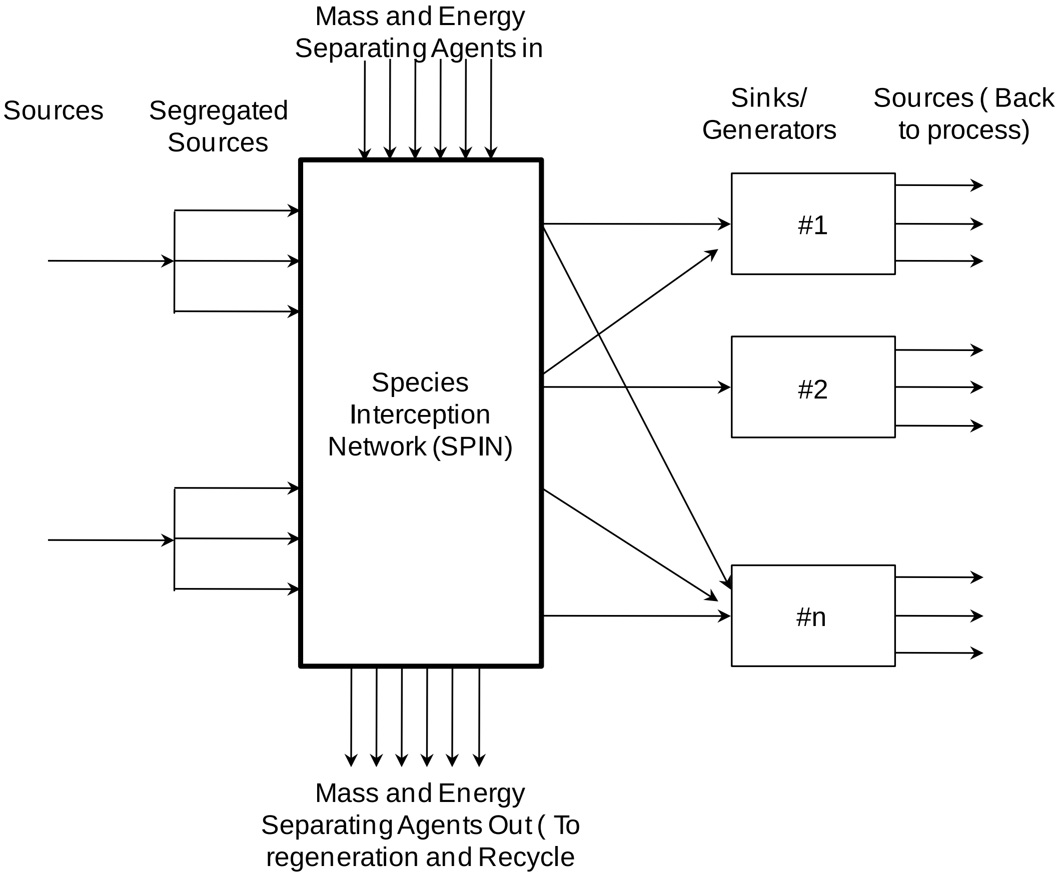

Figure 2 - 29, Mass Integration from a species perspective (El-Halwagi & Spriggs, 1998)

In preparing the sources for the sinks, Mass Integration thus involves a combination of stream manipulations which include:

- Segregation
- Mixing
- Recycle
- Interception and
- Sink/generator manipulation.

The principles involved in each of the above concepts are discussed in detail in the references provided above and will not be presented here.
Of importance in water and wastewater minimisation (Wang & Smith, 1994) are the concepts of reuse, regeneration reuse and regeneration recycle all falling under recycle in the above 5 bullets. These concepts are illustrated in Figure 2-30 below.

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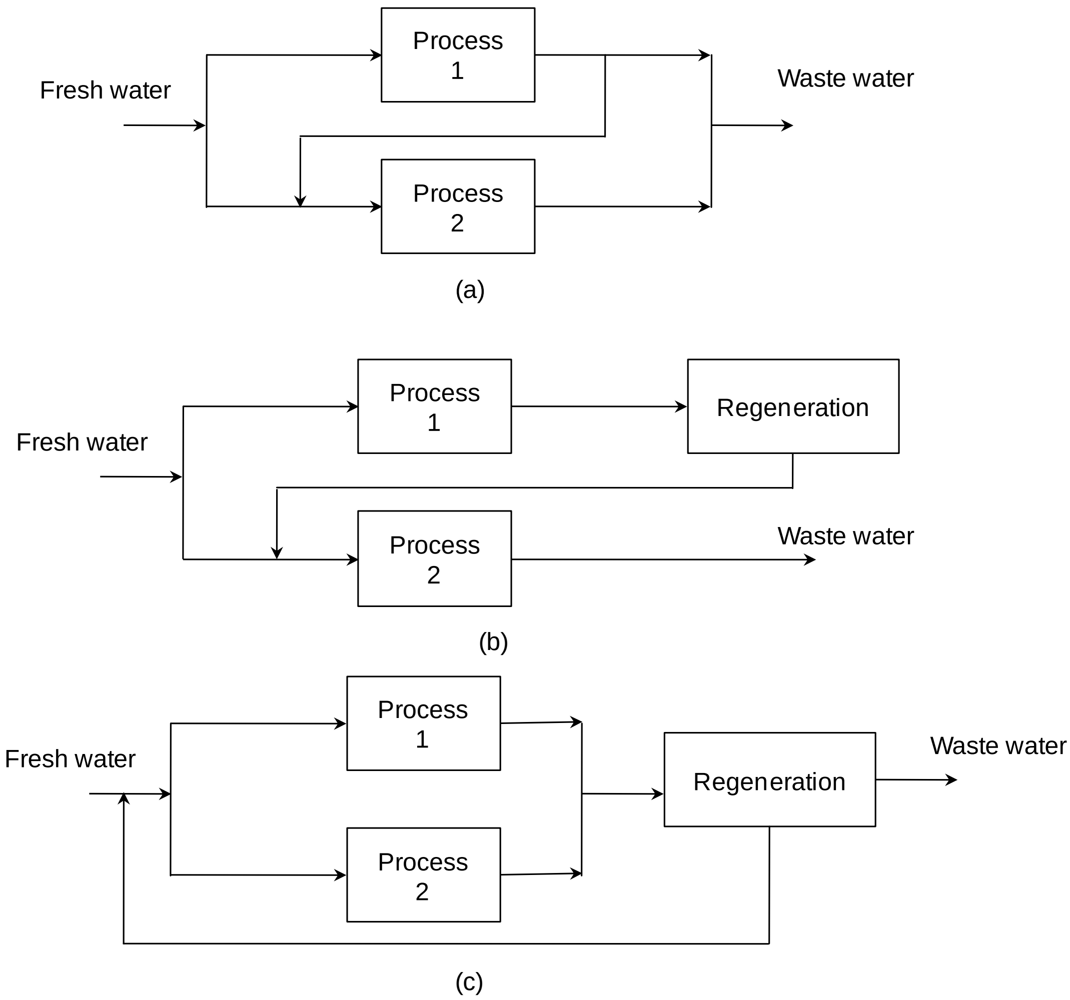

Figure 2 - 30, Water reuse strategies (a-reuse, b-regeneration reuse and c-regeneration recycle)

The Pinch concept of Linnhoff & Hindmarsh (1983), is used during targeting in Mass Integration is a similar way to heat integration. El-Halwagi & Manousiouthakis (1990), showed how such targeting is done in the reduction of contaminant load in a process that needed to be removed by mass separating agents by maximizing process-process Mass Integration as was illustrated in Figure 2-19 for heat integration. Figure 2-31 below shows the identification of the pinch point in Mass Integration.

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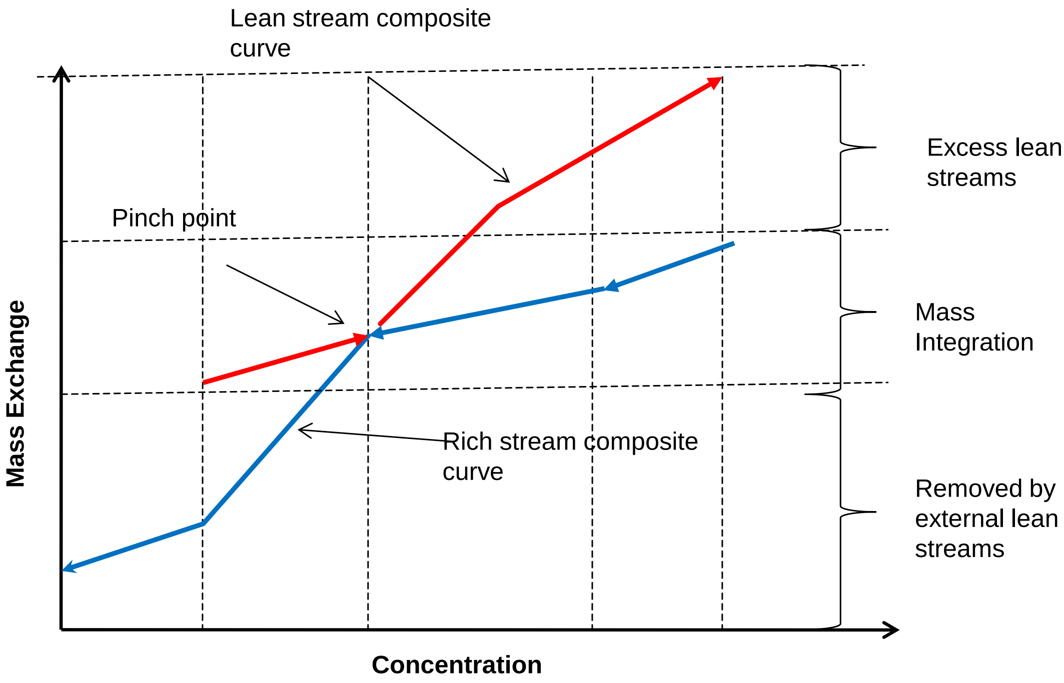

Figure 2 - 31, Pinch point in Mass Integration.

2.10 Property Integration

Property Integration is a relatively new field of Process Integration and only an overview of the concept is presented here. More information on the subject including areas of applicability is available in El-Halwagi (2006).

Property Integration in a new way of thinking about Process Integration as opposed to the traditional way in mass and Energy Integration and the concepts involved were pioneered by El-Halwagi and co-workers, El-Halwagi *et al.* (2004) and El-Halwagi (2006).

The motivation for the new area of Process Integration was that as opposed to Mass Integration techniques being centred on the individual chemical species, most material reuse problems are governed by stream properties or stream functionalities (El-Halwagi, 2006). Examples of such problems presented by El-Halwagi (2006) are:

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- In separation processes that are equilibrium dependent were parameters such as distribution coefficients, viscosity and volatility are used.
- In the analysis and design of heat exchangers where heat capacities and heat transfer coefficients are used.
- In emission control where the properties of pollutants which include volatility and solubility are used.
- Parameters which include, pH, TOC BOD as used in describing environmental pollutants.

Property Integration is thus defined as a functionality-based, holistic approach to the allocation and manipulation of streams and processing units, which is based on the tracking, adjustment, assignment, and matching of functionalities throughout the process (El-Halwagi *et al.,* 2004; Shelley & El-Halwagi, 2000).

Visualization techniques for identifying targets for direct reuse in property based applications are presented in El Halwagi (2006). Additional work on the concept of Property Integration and graphical techniques including algebraic methods are presented in literature (Kazantzi *et al.,* 2004, El-Halwagi *et al.,* 2004; Qin *et al.,* 2004 and Foo *et al.,* 2006).

2.11 Concluding remarks

The theoretical aspects of Process Integration were presented in this chapter. The chapter started with brief overviews of the principles of operation of both steam systems for power generation and cooling towers. This was followed by a discussion on Process Integration. Process Integration was described as having three main components which are process synthesis, process analysis and process optimisation and these were identified as transcending the individual processes and being generally applicable to process design. Energy Integration was presented in relatively more detail with particular focus being on Pinch Technology as applied to heat and power integration. Also presented were a selection from literature of the studies and optimisation methods for both steam and cooling water systems that illustrated the development of the subject over the years. The main conclusion from the review is that most of the previous research focuses on the individual components of plant utility systems. Few studies attempted a holistic treatment of the utility system with the background process but none included the treatment of the steam system and cooling water system for process plants, including power generation, in a single comprehensive approach. There is thus a need for a complete and holistic treatment of plant utility systems and this work aimed at providing such an analysis for a power plant. The methods used and the results obtained are presented in the following two chapters respectively, viz. Chapters 3 and 4.

The last sections presented overviews of Mass and Property Integration. The two main classes of Mass Integration were identified as Mass and Water Pinch Technologies. Finally the overview on Property Integration focused on spurring the new approach which continues to see active research in Process Integration.

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3

Chapter Three

3 Model formulation

3.1 Introduction

This chapter presents the models that were used in this work. It describes how the comprehensive utility system model was formulated. The discussion starts with a description of the modelling approach that was followed in the work. This is then followed by a description of the steam system models which in turn is followed by a description of the cooling water system model and finally the comprehensive system model is presented. The application of the models is presented in the next chapter, Chapter 4.

Figure 3-1 below, presented already in Chapter 1 is reproduced for reference purposes. The rest of this chapter presents the formulation of the mathematical model of the complete power plant as shown in Figure 3-1 including that for a variation of the power plant as was shown in Figure 1-3 of Chapter 1.

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3.2 Modelling approach

The comprehensive utility system model was developed in such a way that the individual models for the Rankine cycle and cooling tower systems were formulated first and later combined to give a single integrated system model.

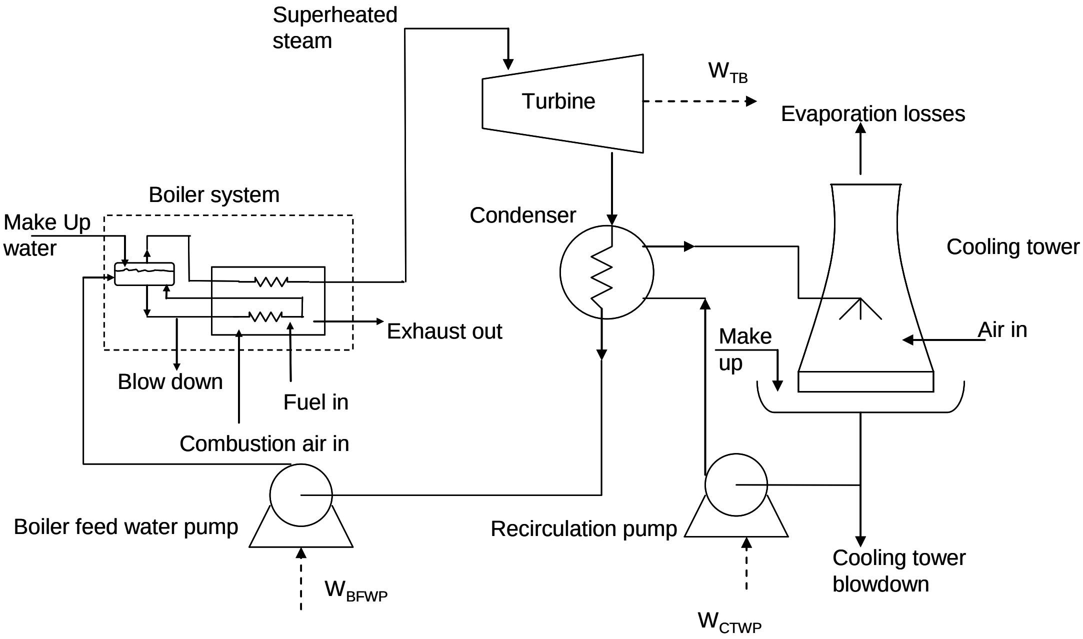

Figure 3 - 1, Simple Rankine cycle for power generation

The mathematical models formulated in this work gave nonlinear programming problems (NLP's) which were discussed in Chapter 2 as having the general structure:

$$\min Z(x), x = [x_1, x_2, ...x_n]^T \in R^n \tag{3-1}$$

or

$$\max Z(x), x = [x_1, x_2, ...x_n]^T \in R^n \tag{3-2}$$

subject to the constraints:

$$\begin{aligned} g_j(x) &\leq 0,\ j = 1,2......., m \\ h_j(x) &= 0,\ j = 1,2......., r \end{aligned} \tag{3-3}$$

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where $Z(x)$, $g_j(x)$ and $h_j(x)$ are scalar functions of the real column vector x^T and x is the vector of optimisation variables. The constraints in the models were the mass, energy and pressure equations and these together with the performance indices, presented in the following section, as objective functions, where solved in both the GAMS® and Matlab® programming environments. Details of the solution algorithm are presented in Sections 3.6 and 3.7 of the thesis.

3.3 Performance Indices

This section presents a discussion of the performance indices that were used in this work.

A performance index (*PI*) is a measure of how effective a process or system is. Presented first are the typical performance indices for the Rankine cycle (power block) and the cooling tower. Lastly, three *PI's* for the combined system are presented. These overall system *PI's* were used for the analysis of the comprehensive system in this work and the results are presented in Chapter 4.

3.3.1 Rankine cycle

A steam turbine is an energy conversion device and a general measure of the effectiveness of such an energy conversion device is the thermal efficiency defined as the ratio of the cycle net work to the heat supplied from external fuel sources.

$$n_{th} = \frac{\text{cycle net work}}{\text{heat supplied from external sources}} \tag{3-4}$$

In optimising a steam system with the thermal efficiency as the objective function, the aim will be to maximise the thermal efficiency.

The Carnot efficiency, which is the limiting value on the fraction of heat added which can be converted to useful work is given by:

$$\eta_{Carnot} = 1 - \frac{T_C}{T_H} \tag{3-5}$$

where T_H and T_c are the temperatures at which heat is added and rejected respectively. A Carnot engine was presented in Chapter 2 as a completely reversible engine operating between the two temperature extremes. It was also explained that the cycle efficiency can be increased by either lowering the cold reservoir (condenser) temperature or by increasing the superheated steam temperature (turbine inlet temperature). Efficiencies for real Rankine cycles are affected in a similar way by the change in the reservoir temperatures.

A second common measure of efficiency used in power plants is the heat rate defined as the ratio of the rate of heat addition to the power output. The rate of heat addition is proportional to the fuel consumption rate and therefore the heat rate is a measure of fuel utilisation rate per unit of power output. A low value of the heat rate represents a high thermal efficiency and hence desirable (Weston, 1992 and Rogers & Mayhew, 1997).

$$HR = \frac{\text{rate of heat addition (kJ/h)}}{\text{power output (kW)}} \tag{3-6}$$

Unlike, thermal efficiency, with the heat rate, the aim will be to minimise it in an optimisation program when used as the objective function.

3.3.2 Cooling water system

The performance of a cooling tower is generally measured by the effectiveness defined as the ratio of the actual heat removal to the maximum attainable heat removal:

$$\eta_{eff} = \frac{Q_{ACT}}{Q_{MAX}} \tag{3-7}$$

or

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$$\eta_{eff} = \frac{h_o - h_i}{h_{s,w,i} - h_i} \tag{3-8}$$

where *h* is the enthalpy of air and the subscripts *o* and *i* represent outlet and inlet conditions and $h_{s,w,i}$ is the enthalpy of saturated air evaluated at the inlet water temperature.

A high effectiveness is thus desirable as it represents better cooling and high heat removal and thus in a cooling tower optimisation model, the aim will be to maximise the effectiveness.

3.3.3 Comprehensive system performance index

The development of the comprehensive system *PI* was motivated by the understanding that an integrated system performance index should be one that caters for both the boiler and cooling tower performances. Thus the two measures of efficiencies, Equations 3-6 and 3-7 (or 3-8) were as a first approach, added to give a new performance index for the comprehensive utility system:

$$PI = \eta_{th} + \eta_{eff} \tag{3-9}$$

In this case, a high *PI* will be desirable as it represents better performance of both the steam and cooling water systems and thus the aim will be to maximise the *PI* when used as an the objective function in the comprehensive system model. During the optimisation process, in order to ensure that both the thermal efficiency of the optimised system is not less than that attainable prior to the optimisation process, a minimum bound is added as a constraint in the model, i.e.:

$$\eta_{th} \geq C_1^L \tag{3-10}$$

Where C_1^L is a constant less than unity and set equal to the initial system value before optimisation.

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The effect of the two subsystems performance indices on the overall system PI as shown in Equation 3-9, is understood by noting that the two systems above, that is both the boiler and cooling tower are linked at the condenser and thus a measure of the overall system performance will be a function of the condensation or cooling water temperature. This is because the temperature of the cooling water determines the condensation temperature and hence the pressure of the condensing steam. It is therefore clear from this and earlier discussions that the colder the cooling water temperature, the lower the minimum temperature and pressure of the steam cycle and consequently the higher the efficiency of the steam system. Thus an increase in the recooled water temperature results in a decrease in the net power output and hence thermal efficiency while on the other hand, the heat that is rejected by the condenser/cooling tower and hence effectiveness increases. The relative change on the overall system PI as given by Equation 3-9 is thus based on the relative changes in each of the two subsystems PI's.

It should be noted also that a good PI is one that incorporates the element of sustainability amongst the other important criteria that define the effectiveness of a process or system. In a power plant similar to the one shown in Figure 3-1 above, such a PI should be linked with both the feed to and effluent from the plant which include the makeup water, blow down, evaporation losses and boiler exhaust. Kroger (1996, 2004) indicates that in a modern fossil fuelled power plant equipped with a wet cooling system, an average of between 1.6 and 2.5 litres of make-up cooling water will typically be required for cooling per kWh or net generation to replace cooling tower evaporation losses alone. He further mentions that in such power plants, blowdown and drift losses combined will range from about 20 percent to 50 percent of evaporative losses corresponding to from 6 to 3 cycles of concentration. A reduction of the evaporative losses will therefore also result in a reduction of both the blowdown and make up water requirements resulting is a more sustainable plant operation.

Based on the preceding discussion, a second PI for the combined system was therefore defined as the heat rejection per kW (or kWh) of net power generation, i.e.:

$$PI = \frac{\text{heat rejected to atmosphere}}{\text{net power generation}} \qquad (3\text{-}11)$$

or

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$$PI = \frac{Q_c}{W_{Net}} \tag{3-12}$$

Better performance will correspond to a low PI. A close look at the PI defined by Equation (3-11) shows that minimising the PI in an optimisation model can either be obtained by increasing the net power generation or reducing the heat rejected to the atmosphere. As highlighted above, these changes will be attended by an increase in the thermal efficiency of the power block together with a reduction in the blowdown and correspondingly low make up water. A higher thermal efficiency for a fixed power output would mean a reduction in the fuel consumption and correspondingly a reduction in the flue gasses released by the boiler to the atmosphere. As in the previous definition of the comprehensive system PI (Equation 3-9), the cooling tower effectiveness and power block thermal efficiencies can be given lower bounds using constrains to prevent obtaining results worse off than those at the beginning of the optimisation (Equation 3-10) and also to reduce the search space.

The third PI that was used for the comprehensive utility system was cost. The ultimate measure for any optimisation effort should have a positive effect on any company's bottom line and thus cost was selected as a PI. The annual revenue from the power plant was defined as the difference between the revenue from selling of electricity and the total operating costs, Equation 3-13:

$$\text{Maximise } TR = ER - TOR \tag{3-13}$$

In the preceding Equation, *TR* is the total revenue, *ER* is the revenue from selling electricity and *TOR* is the total operating cost. The operating costs are determined from the fuel consumed, makeup water consumed and the power consumed by the fan and water circulation pumps of the cooling tower and the boiler. With the pump power (kW) given by Equation 3-14:

$$\text{Pump Power} = \frac{Q\Delta P}{\eta \times 3.6 \times 10^6} \tag{3-14}$$

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In Equation 3-14, Q is the volumetric flowrate in m^3/h, ΔP is the pump head in Pa, η is the pump efficiency (taken to be 0.6 in this work). The fan power in kW is given by Perry & Green, 1997 as shown in Equation 3-15:

$$\text{Fan Power} = 2.72 \times 10^{-5} Q \text{x} P \tag{3-15}$$

In Equation 3-15, *P* is the fan discharge head in cm of water. The following equations can thus be written:

$$TOR = c_1 \times \Delta H_{Boiler} + c_2 \times M + c_3 \times \left(\frac{Q_{ST} \Delta P_{STP}}{\eta \times 3.6 \times 10^6} + \frac{Q_{CT} \Delta P_{CTP}}{\eta \times 3.6 \times 10^6} + 2.72 \times 10^{-5} Q_{Fan} P_{Fan} \right) \tag{3-16}$$

$$ER = c_3 \times W \tag{3-17}$$

where c_1 is the unit cost of fuel ($/GJ), c_2 is the unit cost of makeup water ($/$m^3$), c_3 is the unit cost of electricity ($/kWh), ΔH_{Boiler} is the fuel energy input in the boiler (GJ/h), M is the makeup water (m^3/h), and the subscript “ST” is for steam turbine, “STP” is for steam turbine circulation pump, “CTP” is for the cooling tower circulation pump and “Fan” is for the cooling tower fan.

With cost as the PI, the aim in the optimisation program will be to maximise the cost objective function, hence improving the profitability.

The following sections present the detailed models for the steam system, cooling water system and finally the comprehensive utility system.

3.4 The steam turbine model

Presented first is the model for the single steam turbine system as shown in Figure 3-1. Section 3.5 presents the model for a regenerative steam turbine cycle with a single feed pre-heater.

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As part of the modelling approach outlined in Section 3.2, the boiler section was not modelled in a detailed way, instead a fixed boiler efficiency was used and that allowed for the calculation of the fuel energy required for given inlet boiler feed water and outlet superheated steam conditions going to the steam turbine. Boiler efficiencies are generally fixed parameters based on the type of fuel and boiler used, as long as the boiler is not operated at part load conditions. Gas fuelled boilers (e.g. natural gas) are generally more efficient than solid fuelled (e.g. coal) boilers. For the power industry, circulating fluidised bed boilers running on pulverized coal, are the almost invariably used technology and have efficiencies ranging from 72 to 95%. The use of the boiler efficiency as per this discussion is given in more detail below (see Section 3.4.7).

3.4.1 The steam turbine

Figure 3-2 below shows the steam system that was modelled in this work. This is an extract of only the steam turbine system from Figure 3-1 and shows the numbering convention used in the model.

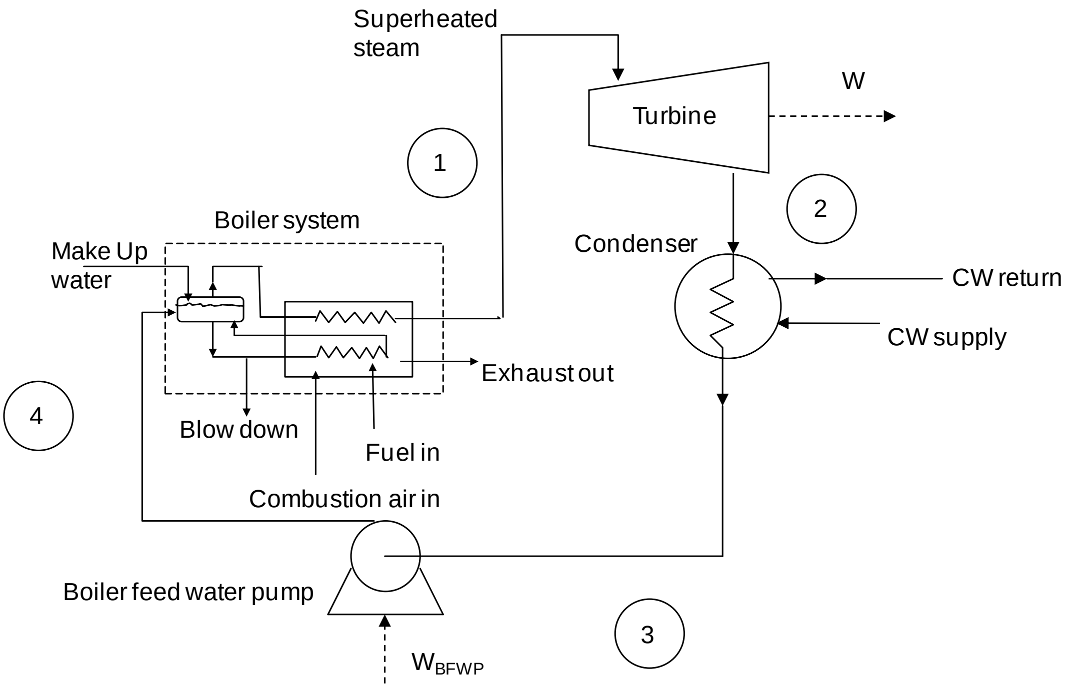

Figure 3 - 2, Model for a steam turbine system

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3.4.2 Variables

Below is the list of the variables used in the steam system model:

β	expansivity
c_P	specific heat capacity (J/kg.K)
F	mass flowrate of steam and water (kg/s)
h	specific enthalpy (kJ/kg)
H	enthalpy flow (kJ/s)
ϕ	vapour fraction
η_{th}	thermal efficiency
P	pressure (kPa)
ΔP	pressure drop across the turbine (kPa)
P_{Sat}	vapour pressure (bar)
Q	heat load (kW)
s	specific entropy (kJ/kg)
T	temperature (K)
T_r	reduced temperature
T_{sat}	saturation temperature (K)
V	molar volume (m^3/mol)
W	turbine work output (kW)

3.4.3 Parameters

A, B, C, D	constants in heat capacity equation
A', B', C'	Antoine coefficients
ε	conversion criterion for modelling
P_1	turbine inlet Pressure (K)
T_1	turbine inlet Temperature (K)
η_P	pump efficiency
η_M	turbine mechanical efficiency
η_{Iso}	turbine isentropic efficiency

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η_{Comb}	boiler combustion efficiency
M_W	molecular weight
R	universal gas constant (J/mol.K)
λ	latent heat of vaporisation

3.4.4 Subscripts and superscripts

0	reference conditions
$1,2,3 \; and \; 4$	refers to stream number as shown in Figure 1 above
c	critical property
$comb$	combustor
cyc	Cycle
Gen	generated
ig	Ideal gas
iso	isentropic
L	liquid phase
V	vapour phase
vap	vapourisation
$cond$	condenser
sat	saturated
Sup	superheated vapour
TB	turbine

3.4.5 Assumptions

The following assumptions are made in the model.

1. Stream 1 is superheated steam and thus the vapour fraction is 1.
2. Stream 3 is saturated liquid at the condenser pressure without any vapour present, i.e. total condensation takes place.
3. Stream 4 is complete liquid, i.e. the vapour fraction is 0.

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4. The temperatures of stream 3 and 4 are the same, i.e. there is negligible temperature increase due to the pump action.
5. The approach temperature in the condenser is 5 ^{o}C.
6. The steam in the system behaves as an ideal gas and the condensate as an incompressible liquid.
7. For the circulating fluid, the pressure drop across both the condenser and the boiler are zero.

The last two assumptions where added to simplify the analysis and it should be noted that in real systems the approach temperature is usually higher than 5^{o}C and the pressure drop not equal to zero. The inclusion of pressure drop analysis can as a first approximation be taken as a constant delta P across each of the condenser and the boiler. Empirical correlations are available in literature for estimating these.

3.4.6 Constraints

The model developed in this work follows the approach of Rodriguez-Toral *et al.* (2001). The constraints for the model are the balance equations (mass and energy) including pressure equations together with equations describing the performance of the system and those equations for estimating the physical properties. The constraints are given below.

3.4.6.1 Mass balance

A mass balance around the steam turbine system gives:

$$F_1 = F_2 = F_3 = F_4 \tag{3-18}$$

3.4.6.2 Energy balance

An energy balance around the turbine gives:

$$\eta_M \left(H_2 - H_1\right) = W \tag{3-19}$$

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where η_M is the mechanical efficiency. A heat balance around the condenser gives:

$$H_2 - H_3 = Q_{Cond} \tag{3-20}$$

3.4.6.3 Pressure balance

A pressure balance around the steam turbine gives:

$$P_2 - P_1 = \Delta P \tag{3-21}$$

where ΔP is the pressure drop across the turbine and is one of the variables.

3.4.6.4 Performance equations

This section gives the performance equations which form part of the constraints for the model. The turbine isentropic efficiency is defined by:

$$\eta_{iso} = \frac{(h_2 - h_1)}{(h_{2,iso} - h_1)} \tag{3-22}$$

where the subscript “iso” depicts an ideal isentropic turbine. h_{iso} is the outlet enthalpy for an isentropic turbine having the same pressure drop and the real unit.

Additional equations (3-23 to 3-25 below) are needed to define the isentropic outlet conditions:

$$h_{iso} = \phi_{iso} h_v(T_{iso}, P_2) + (1 - \phi_{iso}) h_L(T_{iso}, P_2) \tag{3-23}$$

$$s_{iso} = \phi_{iso} s_v(T_{iso}, P_2) + (1 - \phi_{iso}) s_L(T_{iso}, P_2) \tag{3-24}$$

$$s_{iso} = s_1 \tag{3-25}$$

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For each water or steam stream the following equations which describe the stream properties apply:

$$H - hF = 0 \tag{3-26}$$

$$h - \phi h_v(T,P) + (1-\phi)h_L(T,P) = 0 \tag{3-27}$$

$$s - \phi s_v(T,P) + (1-\phi)s_L(T,P) = 0 \tag{3-28}$$

$$P - P_{sat}(T_S) = 0 \tag{3-29}$$

$$\phi + aq - b = 0 \tag{3-30}$$

where *a* and *b* are constants which depend on the phase:

$a = 0, b = 0$ for liquid

$a = 1, b = 1$ for saturated two phase mixture

$a = 0, b = 1$ for vapour

Equation 3-30 above contains a non-smooth function with discontinuous first derivations close to saturated liquid and close to saturated vapour (Rodriguez-Toral *et al.* 2001). (See Figure 3-3 below).

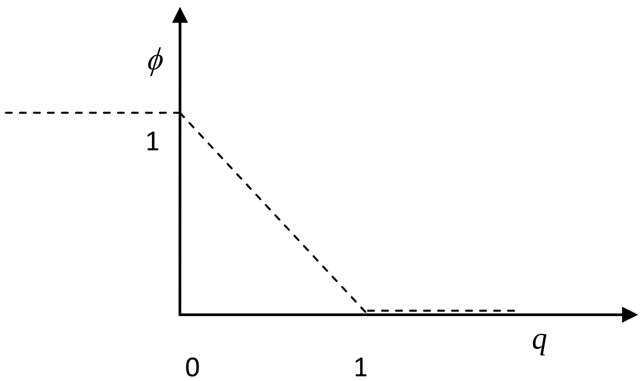

Figure 3 - 3, Diagram for ϕ as a function of q

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Nonlinear programming (NLP) requires continuous second derivations for all the functions and thus in the work Rodriguez-Toral *et al.* (2001) a polynomial was fitted to Equation 3-32 in the region close to the vapour ($q \approx 0.0$) to approximate $aq+b$:

$$\frac{1}{16}\frac{q^4}{\varepsilon^3}-\frac{3}{8}\frac{q^2}{\varepsilon}-\frac{1}{2}q+1-\frac{3}{16}\varepsilon \tag{3-31}$$

where ε is a small value that determines the region over which the polynomial is fitted. They use a value of $\varepsilon = 2\times10^{-5}$ which worked well in all their NLP problems. In the region close to saturated liquid ($q \approx 1.0$), they propose the use of a symmetric fit, taking [1 minus Equation (3-31)] and using $(1-q)$ instead of q:

$$\frac{1}{2}-\frac{1}{16}\frac{(1-q)^4}{\varepsilon^3}+\frac{3}{8}\frac{(1-q)^2}{\varepsilon}-\frac{1}{2}q+1+\frac{3}{16}\varepsilon \tag{3-32}$$

Equations 3-31 and 3-32 thus replace Equation 3-30 near $q=0$ and $q=1$

In this work, the equations describing the properties of the fluid streams where restricted to either liquid or vapour (see assumptions in Section 3.4.5) and thus the vapour fraction (or stream quality) was either 1 or 0, which thus obviated the use of the preceding two equations (3-31 and 3-32) from Rodriguez-Toral *et al*. (2001). Only the turbine exhaust is allowed to be a mixture of both the vapour and liquid phases.

Following the stream numbering convention in Figure 2 we thus have the following.

For the complete vapour stream (stream 1):

$$h_1 - h_v(T_1, P_1) = 0 \tag{3-33}$$

$$s_1 - s_v(T_1, P_1) = 0 \tag{3-34}$$

For the complete liquid stream, Streams 3:

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$$h_3 - h_L(T_3, P_3) = 0 \tag{3-35}$$

$$s_3 - s_L(T_3, P_3) = 0 \tag{3-36}$$

Similar expressions apply for Stream 4. For vapour and liquid stream (Stream 2)

$$h_2 - \phi_2 h_V(T_2, P_2) + (1-\phi_2) h_L(T_2, P_2) = 0 \tag{3-37}$$

$$s_2 - \phi_2 s_V(T_2, P_2) + (1-\phi_2) s_L(T_2, P_2) = 0 \tag{3-38}$$

The condenser pressure is given by:

$$P_3 - P_{sat}(T_3) = 0 \tag{3-39}$$

In the above equations, h_V , h_L , s_V , s_L and P_{Sat} are physical property correlations available for example in Smith *et al.* (2005). These expressions are presented in the following section.

3.4.6.5 Physical property correlations

In the models used in this work, it was assumed that the steam or water vapour behaves as an ideal gas and that the liquid phase is incompressible. The foregoing assumptions allowed the use of ideal gas property correlations together with those for incompressible liquids in determining these properties. It should be noted that these assumptions are reasonable and valid for low to moderate pressures for gasses and almost completely valid for the liquid phase (Smith *et al.,* 2005).

3.4.6.5.1 Ideal gas property equations

From the first and second laws of thermodynamics, it can be shown that:

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$$dh = c_P dT + \left[V - T\left(\frac{\partial V}{\partial T}\right)_P \right] dP \qquad (3\text{-}40)$$

and

$$ds = c_P \frac{dT}{T} - \left(\frac{\partial V}{\partial T}\right)_P dP \qquad (3\text{-}41)$$

where the coefficients of dT and dP are evaluated from heat capacity and *PVT* data. When the ideal gas equation is used for the *PVT* data, then from the following equation:

$$PV = RT \qquad (3\text{-}42)$$

we have

$$\left(\frac{\partial V}{\partial T}\right)_P = \frac{R}{P} \qquad (3\text{-}43)$$

Substituting the preceding equation in Equations 3-40 and 3-41 gives for an ideal gas the following equations:

$$dh = c_P^{ig} dT, \text{ and } \quad ds = c_P^{ig} \frac{dT}{T} - R\frac{dP}{P} \qquad (3\text{-}44)$$

where the superscript "*ig*" denote ideal gas. Equations 3-44 can be integrated from an ideal gas state at reference conditions T_o and P_o to the ideal gas state at *T* and *P* to give:

$$h_v = h_o + \int_{T_o}^{T} c_P^{ig} dT \qquad (3\text{-}45)$$

and

$$s_v = s_o + \int_{T_o}^{T} c_P^{ig} \frac{dT}{T} - R\ln\frac{P}{P_o} \qquad (3\text{-}46)$$

The temperature dependence of the ideal gas heat capacity is normally given in empirical equations of the form:

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$$\frac{c_P}{R} = A + BT + CT^2 + DT^{-2} \tag{3-47}$$

where *A, B, C* and *D* are constants. Smith *et al.* (2005) present these parameters for a number of common organic and inorganic gases. They also present similar parameters for selected solids and liquids.

Equation 3-47 thus allows for the analytic integration of Equations 3-45 and 3-46 to give the enthalpy and entropy of the steam respectively.

Equations 3-48 and 3-49 below are used to calculate the integrals in Equations 3-45 and 3-46 respectively (Smith *et al*., 2005)

$$\int_{T_0}^{T} \frac{c_P}{R} dT = AT_0(\tau - 1) + \frac{B}{2}T_0^2(\tau^2 - 1) + \frac{C}{3}T_0^3(\tau^3 - 1) + \frac{D}{T_0}\left(\frac{\tau - 1}{\tau}\right) \tag{3-48}$$

$$\int_{T_0}^{T} \frac{c_P}{R}\frac{dT}{T} = A\ln\tau + \left[BT_0 + \left(CT_0^2 + \frac{D}{\tau^2 T_0^2}\right)\left(\frac{\tau + 1}{2}\right)\right](\tau - 1) \tag{3-49}$$

where

$$\tau = \frac{T}{T_0} \tag{3-50}$$

The only remaining unknowns will be the enthalpy and entropy at the reference ideal gas state. According to Smith *et al.* (2005), applications of thermodynamics require only the differences in enthalpy and entropy and thus the reference conditions T_o and P_o are selected for convenience and values are assigned to s_o and h_o arbitrarily. For the present work, a reference temperature of 273.15 K and a reference pressure of 0.01 bar were used and the reference enthalpy and entropy at these conditions were both be taken to be zero. The foregoing equations (3-48 and 3-49) are used to calculate both liquid and vapour properties, the only difference

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being the values of the coefficients *A, B, C* or *D* in Equation 3-47. These are different for liquid and vapour phases.

3.4.6.5.2 Calculation of vapour enthalpy and entropy (h_v and s_v)

The use of Equations 3-45 and 3-46 for calculating the enthalpy and entropy assumes that the fluid under consideration remains in the gaseous state from the reference state to the final state. For our present case on the other hand, water exists as a liquid at the reference conditions and a gas at the final state. The equations for calculating the final steam properties thus need to take into account the change of phase that occurs at the saturation temperature.

Equation 3-45 is thus modified to be:

$$h_v = h_0 + h_L + h_{vap} + h_{\sup}$$

or

$$h_v = h_o + \int_{T_o}^{T_{sat}} c_P^L dT + h_{vap} + \int_{T_{sat}}^{T} c_P^{ig} dT \qquad (3\text{-}51)$$

Where h_0 - enthalpy at reference conditions

h_{vap} - is the enthalpy of vaporisation at the saturation temperature T_{sat}.

h_L - liquid enthalpy change from the reference temperature to T_{sat}

$h_{\sup}$ - steam enthalpy change due to superheating to temperature T

The two integrals, the first for calculating the liquid enthalpy due to sensible heat from T_o to T_{sat} and the second for calculating the steam enthalpy change from T_{sat} to T are both evaluated using Equation 3-48 with liquid and vapour heat capacity parameters respectively.

The enthalpy of vaporisation is calculated from the enthalpy of vaporisation at the normal boiling point using the Riedel equation (Smith *et al.*, 2005):

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$$\frac{h_{vap,n}}{RT_n} = \frac{1.092(\ln P_c - 1.013)}{0.930 - T_{rn}} \qquad (3\text{-}52)$$

where P_c is the critical pressure in bars and T_{rn} is the reduced temperature at the normal boiling point calculated from $T_{rn} = T_n / T_c$.

and correcting it for temperature using the Watson Equation (Smith *et al.*, 2005) for the temperature dependence of the latent heat of vaporisation:

$$\frac{h_{vap,2}}{h_{vap,1}} = \left(\frac{1 - T_{r2}}{1 - T_{r1}}\right)^{0.38} \qquad (3\text{-}53)$$

where T_r is the reduced temperature.

Equation 3-46 for the entropy calculation is also adjusted to cater for the phase change to give:

$$s_V = s_0 + s_L + s_{vap} + s_{\sup}$$

or

$$s_V = s_o + \int_{T_o}^{T_{sat}} c_P^{ig} \frac{dT}{T} + s_{vap} + \int_{T_{sat}}^{T} c_P^{ig} \frac{dT}{T} - R \ln \frac{P}{P_{sat}} \qquad (3\text{-}54)$$

The derivation of the liquid enthalpy and entropy equations is presented in Section 3.4.6.5.3 below. The integrals in the above equation are evaluated similarly to those for the enthalpy calculation this time using Equation 3-49.

s_{vap} is calculated from the definition of entropy:

$$s_{vap} \equiv \frac{h_{vap}}{T_{sat}} \qquad (3\text{-}55)$$

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The calculation of the right hand terms in the preceding equation has already been presented under the vapour enthalpy calculations.

3.4.6.5.3 Calculation of liquid enthalpy and entropy h_L and s_L), Incompressible liquid equations

Invoking the definition of expansivity (Equation 3-58 below) in Equation 3-40 and 3-41 gives alternative equations for calculating the liquid phase properties (Smith *et al.*, 2005):

$$dh = c_P dT + (1 - \beta T)VdP \qquad (3\text{-}56)$$

and

$$ds = c_P \frac{dT}{T} - \beta VdP \qquad (3\text{-}57)$$

where β is the volume expansivity defined as follows:

$$\beta \equiv \frac{1}{V}\left(\frac{\partial V}{\partial T}\right)_P \qquad (3\text{-}58)$$

For an incompressible fluid, β, the volume expansivity is zero and Equations 3-56 and 3-57 reduce to:

$$dh = c_P dT + VdP \qquad (3\text{-}59)$$

and

$$ds = c_P \frac{dT}{T} \qquad (3\text{-}60)$$

As was the case for the ideal gas, the above equations can be integrated from a reference state with conditions T_o and P_o to a new state at T and P to give:

$$h_L = h_o + \int_{T_o}^{T} c_P^L dT + V(P - P_o) \tag{3-61}$$

$$s_L = s_o + \int_{T_o}^{T} c_P^L \frac{dT}{T} \tag{3-62}$$

The same analytical expression, Equation 3-47, for the temperature dependence of the liquid heat capacity can be used to integrate Equations 3-61 and 3-62 analytically. The only remaining unknown is the liquid molar volume which is estimated from the critical parameters using the Rackett (1970) equation.

$$V = V_C Z_C^{(1-T_r)^{0.2857}} \tag{3-63}$$

The critical molar volume, V_c and compressibility factor, Z_c are given in Reid *et al.* (1988) for a large number of components and the reduced temperature T_r is calculated from T/T_c.

3.4.6.5.4 Saturation pressure

The saturation pressure is generally correlated by the Antoine equation:

$$\ln P_{Sat} = A' - \frac{B'}{T_{sat} + C'} \tag{3-64}$$

where A', B', C' are constants and available in a number of compilations for most species. Smith *et al*. (2005) tabulates these for selected components.

For elevated temperatures (up to the critical point and pressures up to 200 bar) Equation 3-65 from Polling *et al.* (2001) is used:

$$\ln P_{sat} = \ln P_C + \frac{T_C}{T_{sat}}\left(a\lambda + b\lambda^{1.5} + c\lambda^3 + d\lambda^5\right) \tag{3-65}$$

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Where a, b, c and d are constants, available in Reid *et al.* (1988), for a number of common organic and inorganic substances and

$$\lambda = 1 - T/T_C \tag{3-66}$$

3.4.7 Objective function

The *PI* for the steam turbine model was presented earlier (See Section 3.3.1 above) and thus the objective was to maximise the thermal efficiency of the steam turbine plant:

$$\text{Maximise } \eta_{th}^{Cyc} \tag{3-67}$$

The thermal efficiency of the cycle is defined as the ratio of the network output to the total rate of energy input to the system. For the steam plant shown in Figure 3-1 above, the thermal efficiency is given by:

$$\eta_{th}^{Cyc} = \frac{W}{Q_{comb}} \tag{3-68}$$

where *W* is steam turbine work output. In the comprehensive model described later in this chapter, the cycle thermal efficiency takes into account the power requirements of the cooling tower circulating pump and the boiler feed water pump. Thus the cycle thermal efficiency is given by Equations 3-69 and 3-70.

$$\eta_{th}^{Cyc} = \frac{W_{Gen}}{Q_{comb}} \tag{3-69}$$

where from Figure 3-1 above,

$$W_{Gen} = W_{TB} - W_{CTWP} - W_{BFWP} \tag{3-70}$$

The following definitions apply:

W_{Gen} is the net work output

W_{TB} is the steam turbine work output

W_{CTWP} is the power requirement for the cooling tower circulating water pump

W_{BFWP} is the power requirement for the boiler feed water pump.

Q_{comb} in Equations 3-69 is the energy from the combustion of fuel in the boiler and is related to the superheated steam enthalpy by the combustion efficiency η_{comb} through Equation 3-71 (see Figure 3-4):

$$\eta_{comb} = \frac{(H_{steam} - H_{BFW})}{Q_{comb}} \tag{3-71}$$

Where H_{steam} and H_{BFW} are the enthalpies of the superheated steam entering the steam turbine and boiler feed water, streams 4 and 1 respectively in Figure 3-1.

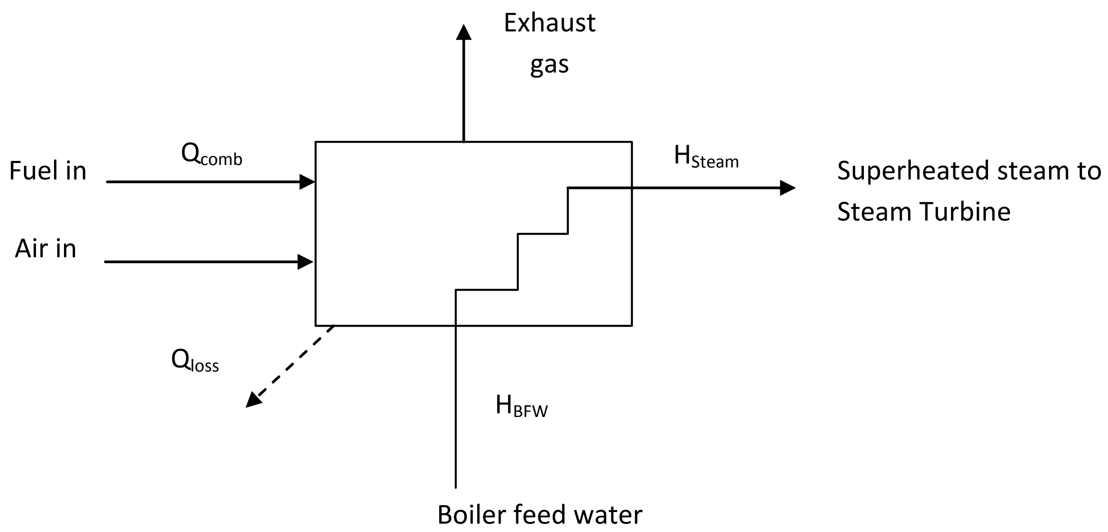

Figure 3 - 4, Energy flow around the boiler in a power plant

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3.4.8 Solution of the steam turbine model

For the steam system, the optimisation problem consisted of maximizing the thermal efficiency of the steam turbine, subject to specified initial conditions stated below:

- HP steam flowrate
- HP steam temperature
- HP steam pressure
- Turbine mechanical efficiency
- Turbine isentropic efficiency
- Boiler and condenser pressure losses which are assumed to be zero

An additional constraint added was a lower bound on the condenser temperature. In the comprehensive system optimisation presented later, this temperature will correspond to the cooling tower outlet water temperature.

3.4.9 Sizing of the steam turbine

The sizing or design of the steam system involved calculating the required high pressure steam flowrate and efficiency for a given work output and specified turbine inlet conditions. It should be noted that the performance of a steam turbine drops significantly at part load conditions and thus the sizing of the steam turbine should result in a single maximum steam flowrate including the pressure and temperature of the turbine inlet steam, conditions for which the turbine should run at. These variables are calculated as part of the solution of the steam turbine model. The results for an illustrative case are presented in Chapter 4.

3.5 Regenerative cycle steam turbine model

This section presents the formulation of a mathematical model of a steam turbine in a regenerative cycle with a single feed heater. Figure 3-5, already presented in Chapters 1 and 2, presents the regenerative cycle with the turbine shown in two

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parts, having a high pressure turbine and a low pressure turbine for modelling purposes. In real applications, an extraction/pass out turbine is used where high pressure steam is extracted from the turbine and used to preheat the feed going to the boiler. The steam turbine model used is an adaptation of the model by Rodriguez-Toral *et al.* (2001)

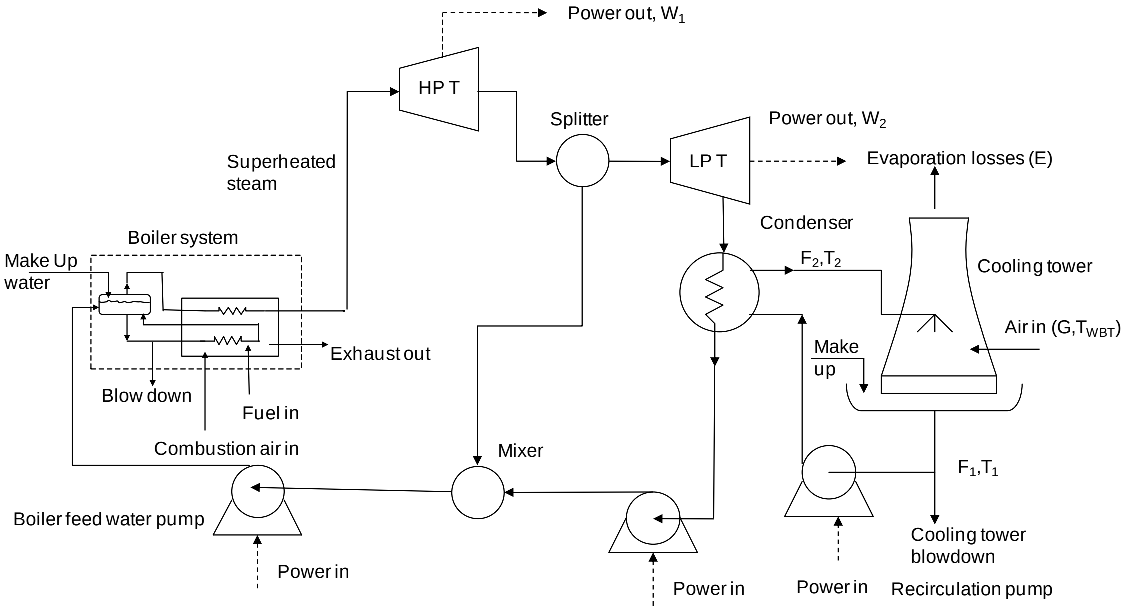

Figure 3 - 5, A comprehensive utility system having a regenerative cycle with a single feed water heater

3.5.1 Variables

Below is the list of the variables used in the regenerative steam system model. Most of the variables and parameters are similar to those used in the simple Rankine cycle above.

β	expansivity
c_P	specific heat capacity (J/kg.K)
F	mass flowrate of steam and water (kg/s)
h	specific enthalpy (kJ/kg)
H	enthalpy flow (kJ/s)
ϕ	vapour fraction

η_{th}	thermal efficiency
P	pressure (kPa)
ΔP	pressure drop across the turbine (kPa)
P_{Sat}	vapour pressure (bar)
Q	heat load (kW)
s	specific entropy (kJ/kg)
T	temperature (K)
T_r	reduced temperature
T_{sat}	saturation temperature (K)
V	molar volume (m^3/mol)
W	turbine work output (kW)
y	fraction of steam extracted

3.5.2 Parameters

A, B, C, D	constants in heat capacity equation
A', B', C'	Antoine coefficients
ε	conversion criterion for modelling
P_1	turbine inlet Pressure (K)
T_1	turbine inlet Temperature (K)
η_P	pump efficiency
η_M	turbine mechanical efficiency
η_{Iso}	turbine isentropic efficiency
η_{Comb}	boiler combustion efficiency
M_W	molecular weight
R	universal gas constant (J/mol.K)
λ	latent heat of vaporisation

3.5.3 Subscripts and superscripts

0	reference conditions

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1,2 3 4,5,6 7,8,9	refers to stream number as shown in Figure 3-6
c	critical property
$comb$	combustor
cyc	Cycle
Gen	generated
ig	Ideal gas
iso	isentropic
L	liquid phase
HPT	high pressure turbine
LPT	low pressure turbine
V	vapour phase
vap	vapourisation
$cond$	condenser
$PUMP1$	condensate pump 1 in Figure 3-5
$PUMP2$	condensate pump 2 in Figure 3-5
sat	saturated
sup	superheated vapour
TB	turbine

.

3.5.4 The steam turbine model

Figure 3-6 below shows the numbering convention that was followed in the development of the model.

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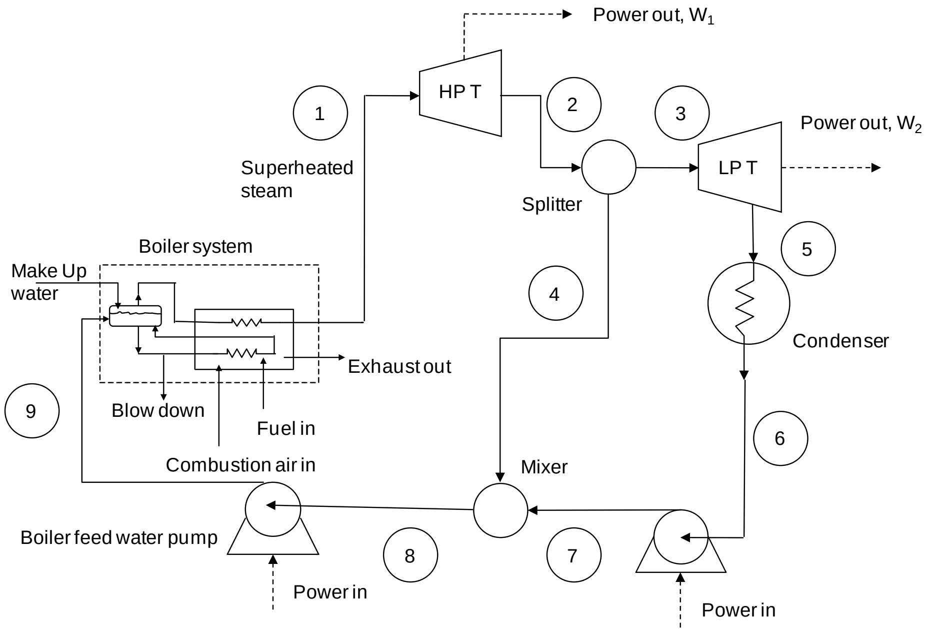

Figure 3 - 6, Regenerative steam turbine plant with a single feed heater

3.5.5 Assumptions

The following assumptions were made in the model:

(i) Streams 1 and 2 are both pure vapour streams and thus the vapour fraction is 1. It follows from Figure 3-6 that this is similarly with streams 3 and 4 which are also vapour streams.

(ii) Stream 6 is saturated liquid at the condenser pressure without any vapour present, i.e. total condensation takes place.

(iii) Streams 7, 8 and 9 are liquid streams and thus the vapour fraction is 0.

(iv) There are negligible temperature differences across the pumps.

(v) The minimum approach temperature between the cooling water supply or return and condenser outlet is 5^{o}C.

(vi) The blowdown from the steam is negligible.

(vii) The steam in the system behaves as an ideal gas and the condensate or liquid water as an incompressible liquid.

(viii) The pressure drop across the condenser and that across the boiler are both equal to zero.

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3.5.6 Constraints

As was the case for the single steam turbine model in Section 3.4 above, the constraints for the model were the balance equations (mass, energy and pressure) together with equations describing the performance of the system and those equations for estimating the physical properties. These constraints are described below.

3.5.6.1 Mass balance

The following mass balance equations can be written from a close look at Figure 3-6:

$$F_1 = F_2 = F_8 = F_9 \qquad (3\text{-}72)$$

$$F_3 = F_5 = F_6 = F_7 \qquad (3\text{-}73)$$

$$F_2 = F_3 + F_4 \qquad (3\text{-}74)$$

$$F_8 = F_4 + F_7 \qquad (3\text{-}75)$$

If y is the split fraction (by mass) at the splitter, then it follows that:

$$F_4 = yF_2 \qquad (3\text{-}76)$$

3.5.6.2 Energy balance

An energy balance around each of the system's components gives:

3.5.6.2.1 Boiler

$$H_1 = Q_1 + H_9 \qquad (3\text{-}77)$$

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3.5.6.2.2 High pressure turbine:

$$\eta_M \left(H_2 - H_1\right) = W_1 \tag{3-78}$$

where η_M is the mechanical efficiency.

3.5.6.2.3 Splitter

$$H_2 = H_3 + H_4 \tag{3-79}$$

Using the split fraction y gives:

$$H_4 = yH_2 \tag{3-80}$$

3.5.6.2.4 Low pressure turbine

A similar energy balance to that of the HP turbine can be written:

$$\eta_M \left(H_5 - H_3\right) = W_2 \tag{3-81}$$

3.5.6.2.5 Condenser

A heat balance around the condenser gives:

$$H_6 - H_5 = Q_{Cond} \tag{3-82}$$

3.5.6.2.6 Mixer

$$H_8 = H_4 + H_7 \tag{3-83}$$

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3.5.6.3 Pressure equations

The following pressure equations can be written from Figure 3-6, and after taking into account assumption number 8 presented in Section 3.5.5 above (i.e. that the pressure drop across the boiler is zero and similarly to the pressure drop across the condenser):

$$P_9 = P_1 \tag{3-84}$$

$$P_2 - P_1 = \Delta P_{HPT} \tag{3-85}$$

$$P_5 - P_3 = \Delta P_{LPT} \tag{3-86}$$

$$P_2 = P_3 = P_4 \tag{3-87}$$

$$P_5 = P_6 \tag{3-88}$$

$$P_8 = P_7 = P_4 \tag{3-89}$$

$$P_7 - P_6 = \Delta P_{PUMP1} \tag{3-90}$$

$$P_9 - P_8 = \Delta P_{PUMP2} \tag{3-91}$$

3.5.6.4 Performance equations

As was the case with the single steam turbine model, the HP turbine isentropic efficiency is defined by:

$$\eta_{iso} = \frac{(h_2 - h_1)}{(h_{2,iso} - h_1)} \tag{3-92}$$

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where the subscript "iso" depicts an ideal isentropic turbine. h_{iso} is the outlet enthalpy for an isentropic turbine having the same pressure drop as the real unit.

Similarly, additional equations are needed to define the isentropic outlet conditions:

$$h_{iso} = \phi_{iso} h_v(T_{iso}, P_2) + (1-\phi_{iso}) h_L(T_{iso}, P_2) \quad (3\text{-}93)$$

$$s_{iso} = \phi_{iso} s_v(T_{iso}, P_2) + (1-\phi_{iso}) s_L(T_{iso}, P_2) \quad (3\text{-}94)$$

$$s_{iso} = s_1 \quad (3\text{-}95)$$

The foregoing equations (Equations 3-93 to 3-95) can be written for the LP turbine.

For each stream (water or stream), the following equations which describe the stream properties applies

$$H - hF = 0 \quad (3\text{-}96)$$

For the vapour streams:

$$h_v - h_v(T, P) = 0 \quad (3\text{-}97)$$

$$s_v - s_v(T, P) = 0 \quad (3\text{-}98)$$

For the liquid streams:

$$h_L - h_L(T, P) = 0 \quad (3\text{-}99)$$

$$s_L - s_L(T, P) = 0 \quad (3\text{-}100)$$

For vapour or liquid mixtures:

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$$h - \phi h_v(T, P) + (1-\phi) h_L(T, P) = 0 \tag{3-101}$$

$$s - \phi s_v(T, P) + (1-\phi) s_L(T, P) = 0 \tag{3-102}$$

The condenser pressure is given by:

$$P - P_{sat}(T) = 0 \tag{3-103}$$

In the above equations, h_V, h_L, s_V, s_L and P_{Sat} are physical property correlations as was explained in Section 3.4.6.5

3.5.7 Objective function

The objective function for the optimisation was to maximise the thermal efficiency of the steam turbine plant:

$$\text{maximise } \eta_{th}^{Cyc} \tag{3-104}$$

For the regenerative steam cycle shown in Figure 3-5 above, with Q_{comb} being the heat load from fuel combustion, the thermal efficiency is given by:

$$\eta_{th}^{Cyc} = \frac{W_1 + W_2}{Q_{comb}} \tag{3-105}$$

Q_{comb} in Equations 3-105 is the energy from the combustion of fuel in the boiler and is calculated the same way as was the case for the single steam turbine model:

$$\eta_{comb} = \frac{(H_1 - H_9)}{Q_{comb}} \tag{3-106}$$

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where H_1 and H_9 are the enthalpies of the superheated steam to the turbine and boiler feed water respectively

3.5.8 Solution of the optimisation problem

The optimisation problem consisted of maximising the thermal efficiency of the steam turbine system, subject to specified initial conditions which were:

- HP steam flowrate
- HP steam temperature
- HP steam pressure
- Turbine mechanical efficiency
- Boiler combustion efficiency

An additional constraint was added, corresponding to a lower bound on the cooling water temperature. This and the assumptions listed in Section 3.5.5 give the following constraints:

$$-T_{CW}+30\leq 0 \qquad (3\text{-}107)$$

$$T_5-T_{CW}-5\leq 0 \qquad (3\text{-}108)$$

$$\phi_1=\phi_2=\phi_3=\phi_4=1 \qquad (3\text{-}109)$$

$$\phi_6=\phi_7=\phi_8=\phi_9=0 \qquad (3\text{-}110)$$

$$\Delta P_5=0 \text{ (Condenser pressure drop)} \qquad (3\text{-}111)$$

$$\Delta P_9=0 \text{ (Boiler pressure drop)} \qquad (3\text{-}112)$$

$$\Delta T_{PUMP1}=\Delta T_{PUMP2}=0 \qquad (3\text{-}313)$$

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$$-\phi_5 \leq 0 \qquad (3\text{-}114)$$

$$\phi_5 - 1 \leq 0 \qquad (3\text{-}115)$$

The constraint represented by Equation 3-114 above can be removed as a separate equation and taken into account by declaring all the variables to be positive in the optimisation program.

3.6 *Cooling tower model*

Figure 3-7 below shows the cooling tower system that was modelled in this work. The model as has already been highlighted is an adaptation of the model of Kim & Smith (2001).

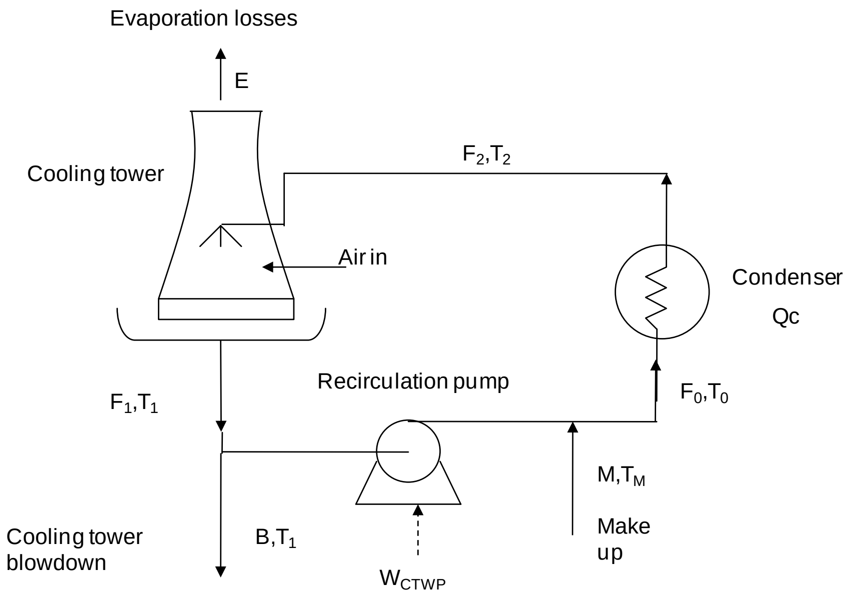

Figure 3 - 7, cooling water system

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3.6.1 Variables

Below is a list of the variables used in the model

B	blowdown flowrate (kg/s)
C_B	salt concentration in blowdown
C_M	salt concentration in makeup
c_P	specific heat capacity (kJ/kg.K)
c_{PA}	heat capacity of air (kJ/kg.K)
c_{PL}	heat capacity of water (kJ/kg.K)
c_S	humid heat capacity (kJ/kg.K)
dH	differential increment of enthalpy
dL	differential increment of water flowrate (kg/m^2.s)
dT	differential increment of Temperature (^{o}C)
dW	differential increment of air humidity
dZ	differential increment of cooling tower height (m)
E	evaporation (kg/s)
F	mass flowrate of steam / water (kg/s)
G	dry air mass flowrate (kg/m^2.s)
h_G	heat transfer coefficient of air (kW/m^2. ^{o}C)
h_L	heat transfer coefficient of water,(kW/m^2 .^{o}C)
H	enthalpy flow (kJ/s)
k_G	mass transfer coefficient of air (m/s)
M	make up water flowrate (kg/s)
η_{eff}	cooling tower effectiveness
P	pressure (bar)
P_{Sat}	vapour pressure (bar)
Q	heat load (kW)
s	specific entropy (kJ/kg. ^{o}C)
T	temperature (^{o}C)
W	humidity (kg water/kg air)
$\dot{V}$	volumetric flowrate (m^3/s)

z	cooling tower height (m)

3.6.2 Parameters

a	interfacial area(m^2/m^3)
a_1, b_1, c_1	constant value of heat transfer coefficient equation in h_G
a_2, b_2, c_2	constant value of heat transfer coefficient equation in h_L
a_3, b_3, c_3	constant value of mass transfer coefficient equation in k_G
B	blowdown flowrate (kg/s)
CC	cycles of concentration
c_s	humid heat capacity (kJ/kg.K)
ε	conversion criterion for modelling
M_{Air}	molecular weight of air
M_w	molecular weight
η_P	pump efficiency
R	universal gas constant (kJ/kmol.K)
$T_{\max}$	maximum cooling water return temperature (K)
$z_{\max}$	maximum cooling tower height (m)
λ_o	latent heat of vaporisation (kJ/kg)

3.6.3 Subscripts and superscripts

0	reference conditions/cold water going to the condenser
1	bottom of cooling tower
2	top of cooling tower
ACT	actual value
c	critical property
cal	calculated
$cond$	condenser
CT	cooling tower
CW	cooling water
DBT	dry bulb temperature
Fan	fan

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G	air
i	interface
in	inlet conditions
L	liquid phase
Max	maximum
Min	minimum
Out	outlet conditions
sat	saturated liquid
v	vapour phase
vap	vapourisation
WBT	wet bulb temperature
W	water
Air	air

3.6.4 Abbreviations

CW	cooling water
$CTWP$	cooling tower water circulation pump

3.6.5 Constraints

As was the case for the steam systems the constraints include the balance equations (mass, energy and pressure equations) together with property and performance equations. These constraints are presented below.

3.6.5.1 Mass balance

The overall mass balance around the cooling tower gives:

$$M = B + E \tag{3-116}$$

A local cold water mass balance (refer to Figure 3-7 above) gives:

$$F_o = F_1 - B + M \tag{3-117}$$

The evaporation loss is given by:

$$E = G\Delta W \tag{3-118}$$

With the cycles of concentration defined by:

$$CC = \frac{C_B}{C_M} = \frac{M}{B} \tag{3-119}$$

the following equations can be written:

$$B = \frac{E}{(CC-1)} \tag{3-120}$$

$$M = E\frac{CC}{CC-1} \tag{3-121}$$

3.6.5.2 Energy balance

The energy balance around the condenser can be written as:

$$Q_{Cond} = F_2 c_P (T_2 - T_0) \tag{3-122}$$

The preceding equation gives the energy given off in the condenser which will be equal to the energy released in the cooling tower.

3.6.5.3 Detailed cooling tower model

The detailed model predicts the exit conditions of water and air based on known inlet conditions of both streams. The model gives the cooling water return flowrate and temperature, evaporation loss and outlet air temperature for a specified cooling tower

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height and inlet conditions. The parameters obtained are functions of the air wet bulb temperature, cooling tower return temperature and the air and water flowrates:

$$T_1 = f\left(F_2, T_2, G, T_{WBT}\right) \tag{3-123}$$

$$F_1 = f\left(F_2, T_2, G, T_{WBT}\right) \tag{3-124}$$

$$E = f\left(F_2, T_2, G, T_{WBT}\right) \tag{3-125}$$

Several assumptions as outlined in Kim & Smith (2001) are employed in the model:

(i) The cooling tower is operated in an adiabatic mode
(ii) The flowrates of dry air and water are constant in the cooling tower
(iii) There are no drift and leakage losses
(iv) The fan location has no effect on the cooling process
(v) The Interfacial areas for heat and mass transfer are equal
(vi) The transfer coefficients are not influenced by temperature
(vii) The thermodynamic properties are constant across any section of the tower.

Figures 3-8 and 3-9 below respectively show the cooling tower and a control volume within the cooling tower, which form the basis of the analysis that follows. In order to find the interface temperatures, which are important in the evaluation of the interface mass and heat transfer, heat balances were set up for the overall control volume (I), water (II) and air (III) sides as presented in Kim & Smith (2001).

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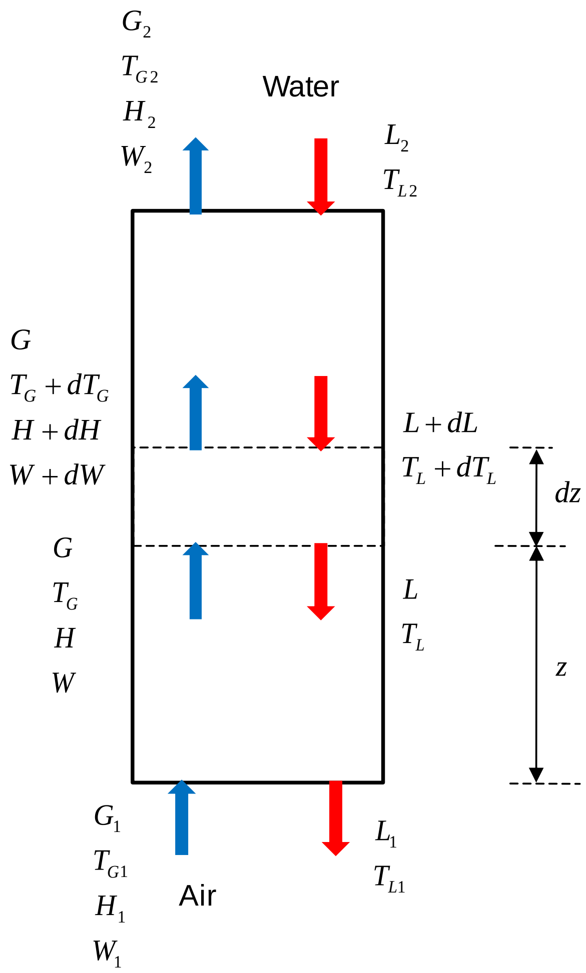

Figure 3 - 8, Cooling Tower (Kim & Smith, 2001)

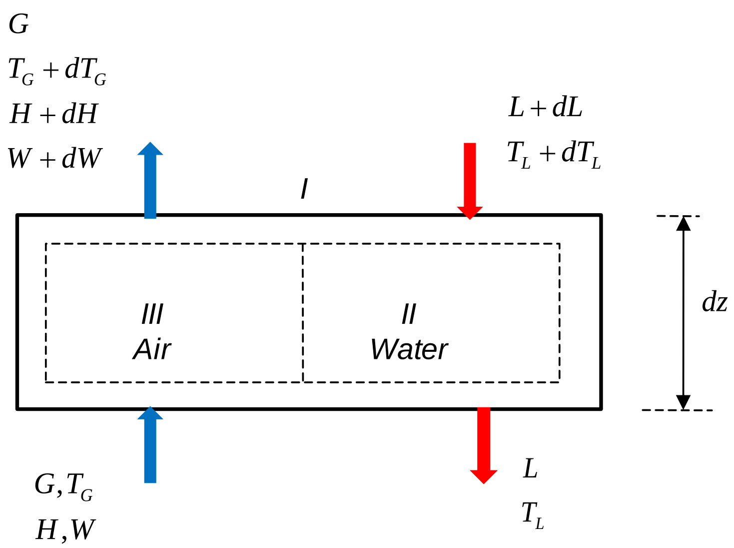

Figure 3 - 9, Control volume of cooling tower model (Kim & Smith, 2001)

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3.6.5.3.1 Control volume (I)

An enthalpy balance around the overall control volume *I* shows that:

enthalpy in =

$$GH + c_{PL}(L + dL)(T_L + dT_L - T_0) \tag{3-126}$$

enthalpy out =

$$Lc_{PL}(T_L - T_0) + G(H + dH) \tag{3-127}$$

where:

$$dH = c_S dT_G + (c_{PA}(T_G - T_o) + \lambda_o)dW \tag{3-128}$$

Equating the inlet and outlet enthalpies gives:

$$GH + c_{PL}(L + dL)(T_L + dT_L - T_o) = Lc_{PL}(T_L - T_o) + G(H + dH) \tag{3-129}$$

$$\Rightarrow GH + (Lc_{PL} + c_{PL}dL)((T_L - T_o) + dT)_L = Lc_{PL}(T_L - T_o) + GH + GdH \tag{3-130}$$

Expanding the left hand side gives:

$$GH + Lc_{PL}(T_L - T_o) + Lc_{PL}dT_L + c_{PL}dL(T_L - T_o) + c_{PL}dLdT_L = Lc_{PL}(T_L - T_o) + GH + GdH \tag{3-131}$$

Cancelling common terms in both the left and right hand sides of the equation and noting that the second order term $dLdT_L \approx 0$ gives:

$$Lc_{PL}dT_L = GdH - c_{PL}dL(T_L - T_o) \tag{3-132}$$

Substituting for dH in the preceding equation using Equation 3-128 and noting that $dL = GdW$ gives Equation 3-133:

$$Lc_{PL}dT_L = Gc_S dT_G + G\left(c_{PA}(T_G - T_o) - c_{PL}(T_L - T_o) - \lambda_o\right)dW \qquad (3\text{-}133)$$

3.6.5.3.2 Control volume (II)

An enthalpy balance around the water side control volume gives:

enthalpy in =

$$c_{PL}(L + dL)(T_L + dT_L - T_o) - GdWc_{PL}(T_L - T_o) \qquad (3\text{-}134)$$

enthalpy out =

$$Lc_{PL}(T_L - T_o) \qquad (3\text{-}135)$$

The heat transfer is modelled as heat transfer coefficient multiplied by the driving force based on interface temperature:

$$h_L a(T_i - T_L)dZ \qquad (3\text{-}136)$$

An overall enthalpy balance around the water side gives:

enthalpy out = enthalpy in – heat transferred to air stream (3-137)

Replacing each of the terms in the above equation gives:

$$(Lc_{PL} + c_{PL}dL)((T_L - T_o) + dT_L) - GdWc_{PL}(T_i - T_o) = Lc_{PL}(T_L - T_o) - h_L a(T_i - T_L)dZ \qquad (3\text{-}138)$$

$$\Rightarrow Lc_{PL}(T_L - T_o) + Lc_{PL}dT_L + c_{PL}dL(T_L - T_o) + c_{PL}dLdT_L - GdWc_{PL}(T_i - T_o)$$

$$= Lc_{PL}(T_L - T_o) - h_L a(T_i - T_L)dZ \qquad (3\text{-}139)$$

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Cancelling common terms in both the left and right hand sides of the preceding equation and noting that the second order term $dLdT_L \approx 0$ gives:

$$Lc_{PL}dT_L = GdWc_{PL}(T_i - T_o) - c_{PL}dL(T_L - T_o) - h_L a(T_i - T_L)dZ \qquad (3\text{-}140)$$

$$\Rightarrow Lc_{PL}dT_L = GdWc_{PL}T_i - GdWc_{PL}T_o - c_{PL}dLT_L - c_{PL}dLT_o - h_L a(T_i - T_L)dZ \qquad (3\text{-}141)$$

Noting that $dL = GdW$ gives:

$$Lc_{PL}dT_L = GdWc_{PL}(T_i - T_L) - h_L a(T_i - T_L)dZ \qquad (3\text{-}142)$$

This upon rearranging gives:

$$Lc_{PL}dT_L = (Gc_{PL}dW - h_L adZ)(T_i - T_L) \qquad (3\text{-}143)$$

3.6.5.3.3 Control volume (III)

enthalpy in =

$$GH \qquad (3\text{-}144)$$

enthalpy out =

$$G(H + dH) - GdW(c_{PA}(T_G - T_o) + \lambda_o) \qquad (3\text{-}145)$$

heat transfer out =

$$h_G a(T_i - T_G)dZ \qquad (3\text{-}146)$$

Again an overall energy balance around the air control volume gives:

enthalpy out = enthalpy in + heat gained from the water stream (3-147)

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and substituting the above terms with the respective equations we get:

$$GH + GdH - GdW\left(c_{PA}\left(T_G - T_o\right) + \lambda_o\right) = GH + h_G a\left(T_i - T_G\right)dZ \tag{3-148}$$

Eliminating the common terms in both the left and right hand sides of the preceding equation and substituting for dH using Equation 3-128 gives:

$$Gc_S dT_G + GdW\left(c_{PA}\left(T_G - T_o\right) + \lambda_o\right) - GdW\left(c_{PA}\left(T_G - T_o\right) + \lambda_o\right) = h_G a\left(T_i - T_G\right) \tag{3-149}$$

which simplifies to Equation 3-150:

$$Gc_S dT_G = h_G a\left(T_i - T_G\right)dz \tag{3-150}$$

Combining Equations 3-133 and 3-143 gives an equation for the interface temperature:

$$T_i - T_L = \frac{Gc_S\left({dT_G}/{dZ}\right) + G\left(c_{PA}\left(T_G - T_o\right) - c_{PL}\left(T_L - T_o\right) + \lambda_o\right)\left({dW}/{dZ}\right)}{Gc_{PL}\left({dW}/{dZ}\right) - h_L a} \tag{3-151}$$

The air humidity, which is a measure of the mass transfer of water vapour from the interface to the air, is given by Equation 3-152:

$$\frac{dW}{dz} = \frac{k_G a}{G}\left(W_i - W\right) \tag{3-152}$$

The air temperature is obtained from Equation 3-150 as:

$$\frac{dT_G}{dz} = \frac{h_G a}{Gc_S}\left(T_i - T_G\right) \tag{3-153}$$

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The water temperature is given by Equation 3-154, which represents the heat transfer from water to the interface:

$$\frac{dT_L}{dz} = \frac{h_L a}{Lc_{PL}}(T_L - T_i) \tag{3-154}$$

An iterative method is necessary for computing the interface temperature in the model, Kim & Smith (2001). Additional information needed for the model is the interface humidity which is given by Equation 3-155:

$$W_i = \frac{M_W P^{Sat}}{M_{Air}(P - P^{Sat})} \tag{3-155}$$

where P^{Sat} is the vapour pressure and is calculated from the Antoine equation. Heat and mass transfer coefficients for the air water system are represented by Coulson & Richardson (1996) as:

$$h_L a = a_1 G^{b1} L^{c1} \tag{3-156}$$

$$h_G a = a_2 G^{b2} L^{c2} \tag{3-157}$$

$$k_G a = a_3 G^{b3} L^{c3} \tag{3-158}$$

where, a, b and c are constants and for the air/water system correlations for these are given in Table 3-1 below from Coulson & Richardson (1996). Equations 3-151 to 3-158 above form the detailed cooling tower model.

Table 3 - 1, Parameters for heat and mass transfer coefficients

a_1	b_1	c_1
3.0	0.26	2.95
a_2	b_2	c_2
1.04×10^4	0.51	0.26
a_3	b_3	c_3
2.95	1	0.72

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Figure 3-10 below shows the flowchart (Kim & Smith, 2001) for the foregoing model. It is used to calculate the exit water and air conditions when the inlet water conditions are specified. Kim & Smith (2001) proposed the use of the Runge-Kutta method for solving the differential equations numerically from the bottom of the tower to the top.

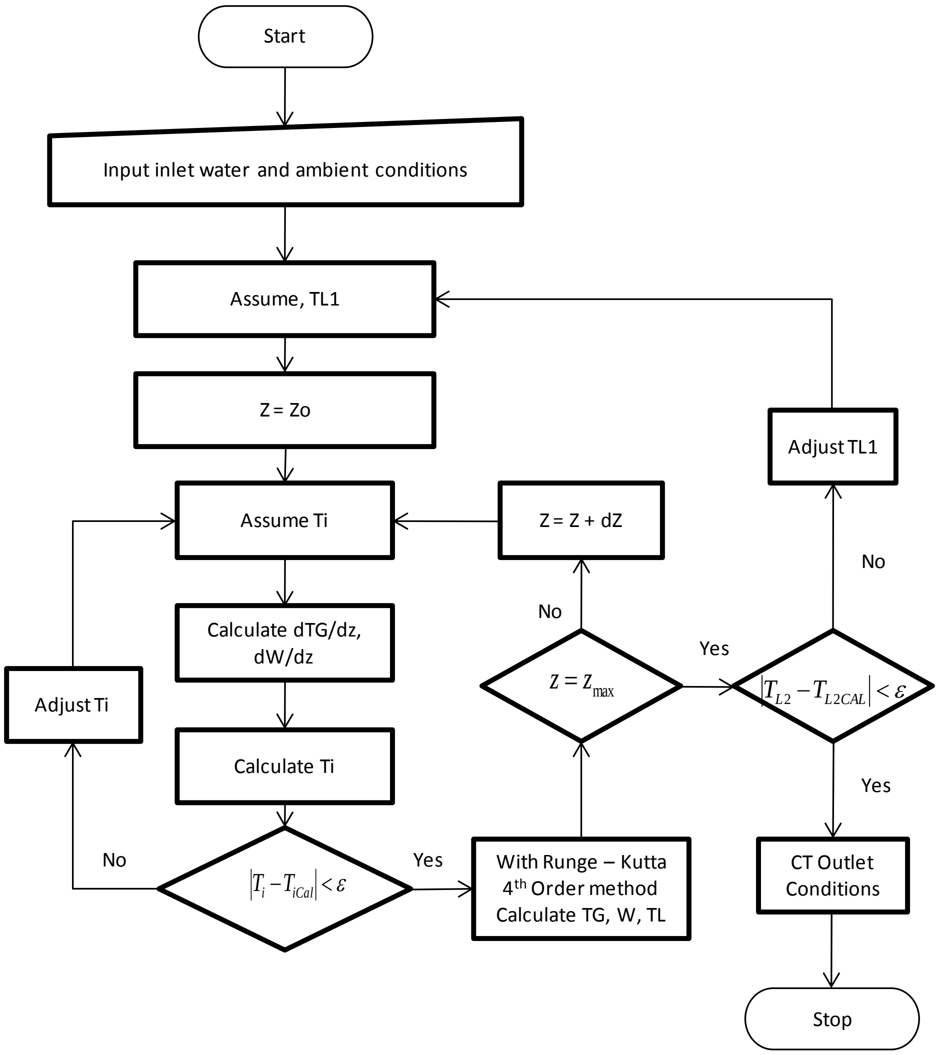

Figure 3 - 10, Flowchart for cooling tower modelling (Kim & Smith, 2001)

In this work, the detailed cooling tower model was solved in Matlab ® and the results were exported to the optimisation program in GAMS®.

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3.6.6 Modifications to the cooling tower model

Two modifications were made to the Kim & Smith (2001) model as was used in this work. The changes made were:

- To allow the model to be used to size a cooling tower
- To allow the model to accurately predict the evaporation loss.

These changes are discussed in turn below.

3.6.6.1 Cooling tower sizing

As was the case with the steam system, (See Section 3.4.9) the cooling tower model was also modified to allow the determination of the cooling tower size. The original Kim & Smith (2001) model is used to determine the exit conditions for a given cooling tower size. In this work, the cooling tower size was determined by solving the Merkel equation:

$$Me = \int_{T_{Wo}}^{T_{Wi}} \frac{C_{pw} dT_w}{\left(i_{masw} - i_{ma}\right)} = \frac{k_d a_{fr} L}{G_w} \tag{3-159}$$

where:

Me is the Merkel Number

L is the height of fill in the cooling tower

G_w is the water flux $\left(M_w \big/ A \right)$

A is the cooling tower area

a_{fr} is the wetted area per unit volume of fill

k_d is the mass transfer coefficient

i_{ma} is the enthalpy of saturated air evaluated at the local bulk water temperature per unit mass of dry air and

i_{masw} is the enthalpy of the air water mixture per unit mass of dry air

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Expressions for the calculation of the air and water enthalpies are available for example in Kroger (2004).

Correlations are available in literature for calculating the fill transfer characteristics, as was discussed in Chapter 2, for example as given in Kroger (2004) and these depend on the type of fill. For this work, a film type of fill was chosen and the mass transfer coefficient for the air water system was calculated using Equation 3-160 below (from Coulson & Richardson, 1996 and similar to Equation 3-158, presented above):

$$k_G a = a_1 G_a^{\ b1}\ G_w^{\ c1} \tag{3-160}$$

with a_1, b_1 and c_1 being constants and G_a being the air flux through the cooling tower. Equation 3-159 was integrated numerically using the Chebyshev method as per British standard 4485. In the four point form the integral is approximated as:

$$\int_a^b f(x)dx \approx \frac{(b-a)}{4}\left[f(x_1)+f(x_2)+f(x_3)+f(x_4)\right] \tag{3-161}$$

with

$$f(x_1) = \text{value of } f(x) \text{ at } x = a + 0.102673(b-a)$$
$$f(x_2) = \text{value of } f(x) \text{ at } x = a + 0.406204(b-a)$$
$$f(x_3) = \text{value of } f(x) \text{ at } x = a + 0.593796(b-a)$$
$$f(x_4) = \text{value of } f(x) \text{ at } x = a + 0.897327(b-a)$$

Apart from the obvious constraints added to prevent the exit air conditions being less than the inlet air temperature, outlet water temperature being less than the inlet water temperature, $\Delta T_{\min}$ between condensation and cooling water temperatures, additional constraints were used in the cooling tower sizing and these include Equations 3-162 to 3-166 below (Kloppers & Kroger, 2003, 2005b; Kroger 2004, Guitierrez *et al.*, 2010).

$$T_{wi} \leq 45.2 \tag{3-162}$$

$$i_{msaw} - i_{ma} > 0 \quad (3\text{-}163)$$

$$0.5 \le \frac{m_w}{m_a} \le 2.5 \quad (3\text{-}164)$$

$$2.9 \le \frac{m_w}{A} \le 5.6 \quad (3\text{-}165)$$

$$1.2 \le \frac{m_w}{A} \le 4.25 \quad (3\text{-}166)$$

3.6.6.2 Cooling tower evaporation loss

As discussed in Chapter 2, the Kim & Smith (2001) method, like that of Merkel (1925) makes simplifying assumptions to allow for the solution of the heat and mass transfer equations taking place inside the cooling tower. In this work, the method of Kim & Smith gave accurate cooling water temperatures when compared to literature values (see Chapter 4), but underestimated the evaporation loss. The amount of evaporated water in the model was thus calculated from the inlet and outlet water temperatures using Equation 3-167 from Perry & Green (1997).

$$E = 0.00085 \times W_C \times (T_{wi} - T_{wo}) \quad (3\text{-}167)$$

In the above equation, E is the evaporation loss in gal/min, W_C is the cooling tower inlet circulating water flowrate in gal/min and $(T_{wi} - T_{wo})$ is the difference in the water inlet and outlet temperature in ^oF.

3.7 Comprehensive utility system model

The combined utility system model was obtained by incorporating the cooling tower model constraint equations, as described above in the corresponding sections of the steam turbine model. The major change was that a single objective function for the comprehensive system model was used as highlighted in Section 3.3. There were also new overall balance equations that were generated together with new equations

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for integrating or linking the two sub systems' models which were based on the common variables in the two models.

3.7.1 Solution algorithm

The solution procedure for the model is shown in Figure 3-11 below. The solution procedure finds the optimum operating conditions for the cooling tower and steam turbine for a known system configuration in an iterative manner. The comprehensive system model is built in GAMS® with the cooling tower ordinary differential equations solved in Matlab® and values interchanged between the two platforms in an iterative manner using the method of Ferris (2005) until convergence.

In the flowchart below, (refer also to Figure 3-11), T_G is the air temperature, T_L is the water temperature and L the water flowrate. The subscripts *1* and *2* refer to the outlet and inlet to the cooling tower, respectively, with *cal* and *opt* referring to the calculated and optimum values. The solution process starts by guessing an estimate of the optimum inlet conditions to the cooling tower from which the cooling tower performance equations are solved from the bottom of the cooling tower to the top (z_o to z_{max}) giving the outlet conditions using the Runge-Kutta method (Kim & Smith, 2001). The cooling tower outlet conditions are then used as input parameters in the main system model to give the optimum conditions of the comprehensive system. The values of the optimum conditions are compared to the initially guessed values and if the required level of accuracy is not achieved, the guessed values are adjusted and the whole procedure repeated until the required convergence criterion has been satisfied.

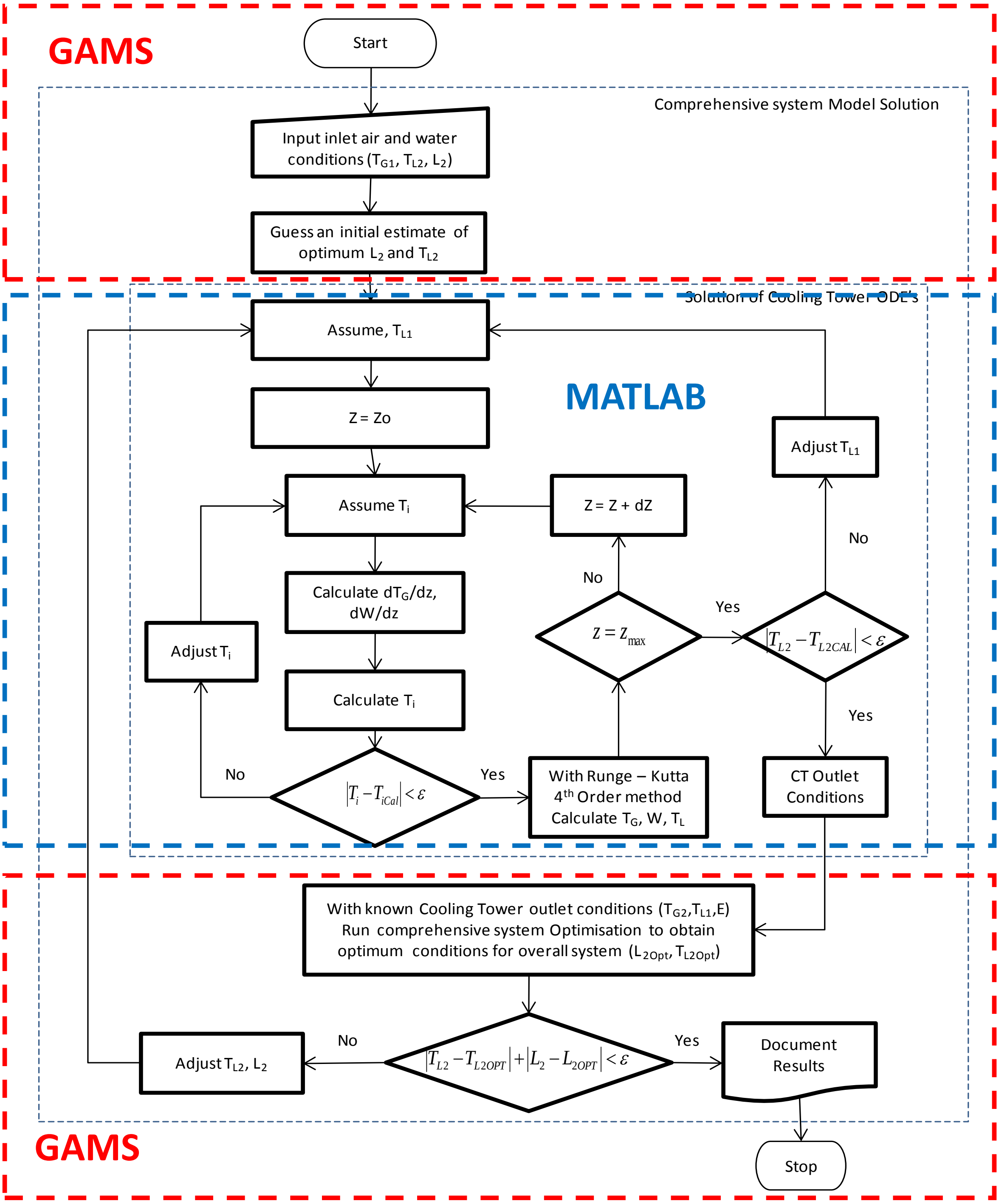

Figure 3 - 11, Solution algorithm for comprehensive utility system

3.7.2 Condensation temperature

The two utility systems are connected at the condenser and the common process variables are thus those of the condenser. The condensation temperature (and the

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condenser duty) is thus the main variable affecting both the cooling tower and steam turbine systems. The actual dependence is explained in Sections 3.3.3 and 4.3.3 for which the reader is referred to for more details.

3.7.3 Simultaneous use of Matlab® and GAMS®

The simultaneous use of the two modelling platforms, viz Matlab® and GAMS® in the optimisation of the systems used in this work and as highlighted in Figure 3-11 is evaluated in Section 4.2 of the next chapter. Comparison is made between the optimisation results of a cooling tower when the model is solely built in Matlab and when the two applications are used simultaneously.

3.8 Concluding remarks

This Chapter presented the models that were used in this work. Presented first were the performance indices that were used in the models as objective functions. These were then followed by the presentation of the steam system models, which were for both the simple Rankine cycle and for the regenerative cycle with a single feed pre-heater. The cooling tower model was then presented and finally the combined model for the comprehensive utility system. The solution procedures together with the algorithms were also presented.

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3.9 References

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4

Chapter Four

4 Model application, results and discussion

4.1 Introduction

The mathematical models that were used in this work were presented in Chapter 3. This chapter, already highlighted in the previous chapters, presents the application of the models in the optimisation of the comprehensive utility system as outlined in Chapter 1. Presented first is the illustration of the use of the individual systems' models by applying the models to test cases together with the validation of the models by comparing the model predicted results to literature values. After having shown that the individual models accurately predict the behaviour of the two utility systems in a simple power plant, and can thus be relied on, the optimisation results for the comprehensive system model are presented. The comprehensive system model results as presented are for the two cases as given in the Aims of the Study in Chapter 1, which is restated as:

Given the operating conditions of a simple power plant similar to that shown in Figure 1-2, together with the relevant sizes of the steam and cooling water systems:

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(i) Determine the optimum operating conditions for the overall comprehensive utility system

(ii) Determine the optimum sizes of the cooling tower and steam systems for a given power output.

The above two cases were also applied to the regenerative steam cycle with a single feed pre-heater as shown in Figure 1-3 in Chapter 1. This was carried out to illustrate the applicability of the work to modern power plants with advanced cycles incorporating regenerative and reheat cycles.

4.2 Individual system model validation

This section illustrates the application of the individual systems' models, particularly focusing on the validation of the models that were developed for the cooling tower and steam systems. The validation was done by comparing the model predicted results to data from real systems as found in literature and through using illustrative test cases.

4.2.1 The single steam turbine model

As an illustrative case, Table 4-1 below shows the input parameters for a single steam turbine model.

Table 4 - 1, Input parameters for single steam turbine model

Description	***Value***	***Units***
Inlet steam flowrate	10	kg/s
Turbine Inlet temperature	550	^{o}C
Turbine Inlet pressure	186	bar
Combustion efficiency	95	%
Mechanical efficiency	99	%
Turbine isentropic efficiency	80	%
Minimum cooling water return temperature	30	^{o}C

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The above values were used as input parameters in the model and the results in Table 4-2 were obtained. The model was solved as an NLP problem using the CONOPT solver in the GAMS ® software. Table 4-2 also shows some of the optimisation statistics.

Table 4 - 2, Single steam turbine model results

Description	***Value***	***Units***
Thermal efficiency	30.7	%
Number of iterations	25	
Variables	64	
Equations	69	
Turbine outlet pressure	0.074	bar
Condenser heat duty	-19931	kW

The model applicability was demonstrated by comparing the model results to those of a single steam turbine plant from a South African chemical company. The power plant is used to generate power for the company's internal consumption. Tables 4-3 and 4-4 show the input parameters for the power plant and a comparison between the plant data and the model results respectively.

Table 4 - 3, Input parameters for a single steam turbine from a South African chemical plant

Description	***Plant value***	***Units***
Power	60	Mwe
Vacuum	10	kPaa
Flow	258	ton/h
HP steam temperature	430	deg.C
Pressure	40	barg
Boiler efficiency	82.56	%
Turbine isentropic efficiency	85	%
Turbine mechanical efficiency	95	%

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Table 4 - 4, Comparison of model results with plant data

Description	*Plant value*	*Model value*	*Units*
Power output	60	60.428	Mwe
Condenser duty	140.435	146.064	MW
Thermal efficiency	24.34	24.2	%
% error in thermal efficiency		0.14	%

4.2.1.1 Discussion

The comparison of the model predicted data to the case data shown in Table 4-4 gave an error of 0.14% in the calculated turbine thermal efficiency. The small error thus indicated that the developed model was good enough and thus could be used in the modelling of the comprehensive utility system. The small difference in the thermal efficiency could be a result of inherent losses and inefficiencies in the real turbine that cannot be accurately modelled. The application of the model in optimising the complete utility system is presented in Section 4.3.

4.2.2 The regenerative cycle steam turbine model

The regenerative steam turbine cycle model was discussed in Section 3.5. Table 4-5 gives the model input data (parameters) from a case in Rodriguez-Toral *et al.* (2001) and the comparison to the results from the model used in this work is presented in Table 4-6.

Table 4 - 5, Input parameters for the regenerative cycle steam turbine model

Description	*Value*	*Units*
Inlet Steam flowrate	10	kg/s
Turbine Inlet temperature	550	^oC
Turbine Inlet pressure	20	bar
Turbine Isentropic efficiency	80	%
Mechanical efficiency	95	%
Combustion efficiency	100	%
Minimum condenser outlet temperature	50	^oC

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Table 4 - 6, Model validation results, two turbine model with feed pre-heater

	Rodriguez-Toral et al.(2001)	*This work*	
Description	**Value**	**Value**	**Units**
Thermal efficiency	28.22	28.6	%
Number of iterations	7	72	
Variables	121	118	
Equations	109	139	
HP turbine outlet pressure	2.48	2.129	bar
LP turbine outlet pressure	0.12	0.124	bar
Flowrate to LP turbine	8.868	8.827	
Split fraction to feed pre-heater	0.1132	0.1173	
Condenser heat duty	21397	19560	kW

4.2.2.1 Discussion

A close look at the model results and the literature results again shows good agreement and the model was thus also deemed acceptable to allow for use in the optimisation of the comprehensive utility system. Presented below is the validation of the cooling tower model which was later combined with the foregoing models to give the comprehensive system utility model.

4.2.3 The cooling tower model

The detailed cooling tower model presented in Section 3-6 of Chapter 3 was formulated in the Matlab® programming environment. The fourth order Runge-Kutta method was used to solve the ordinary differential equations from the bottom of the cooling tower to the top following the flow chart in Figure 3-10.

The cooling tower model was validated by comparing the model predicted results to the experimental data of Bernier (1994). Table 4-7 below compares the predicted cooling tower inlet and outlet temperatures to the experimental data. The experimental data are for four sets from the same cooling tower with differing circulating water flowrates and cooling tower inlet and outlet water temperatures. For

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the first dataset, the model predicted temperature values of 36.7483 °C and 19.801 °C against 36.7°C and 19.8°C for the cooling tower inlet and outlet temperatures.

Table 4 - 7, Cooling tower model validation

Case	***1***	***2***	***3***	***4***
Water flowrate (kg/s)	0.2	0.3	0.398	0.495
Air flowrate (kg/s)	0.67	0.656	0.664	0.658
CW inlet temperature (°C)	36.7	32	29.3	27.9
CW outlet temperature (°C)	19.8	20.4	20.7	20.8
Model result CW inlet temperature	36.7483	31.9985	29.3018	27.9832
Model result CW outlet temperature(°C)	19.801	20.708	21.033	20.52
Error (%) in CW inlet temperature	-0.08	0.00	0.01	0.30
Error (%) in CW outlet temperature	0.01	1.51	1.61	-1.35

Figure 4-1 and 4-2 also show the predicted cooling tower performance in graphical form.

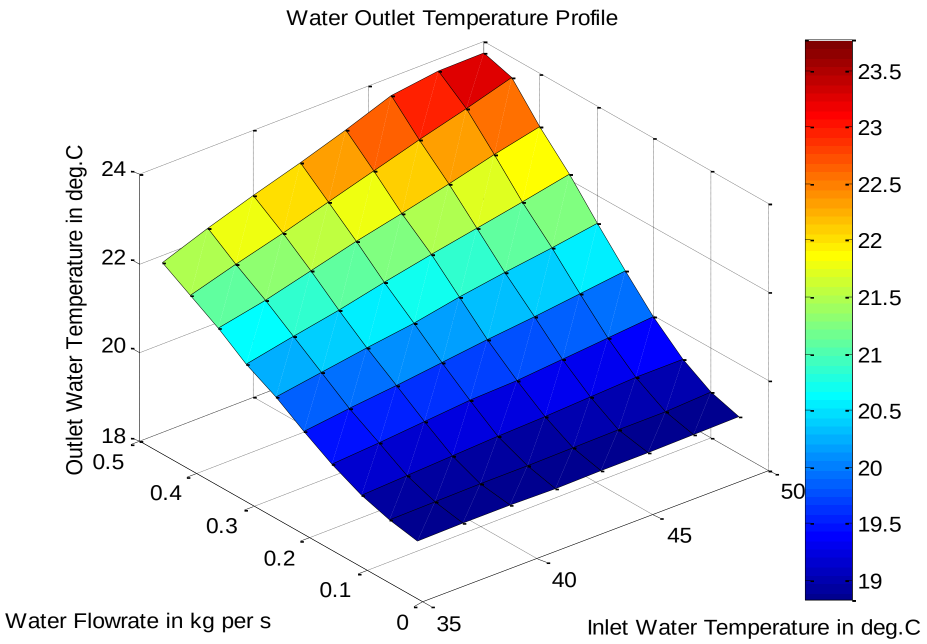

Figure 4 - 1, Cooling tower performance, outlet temperature profile

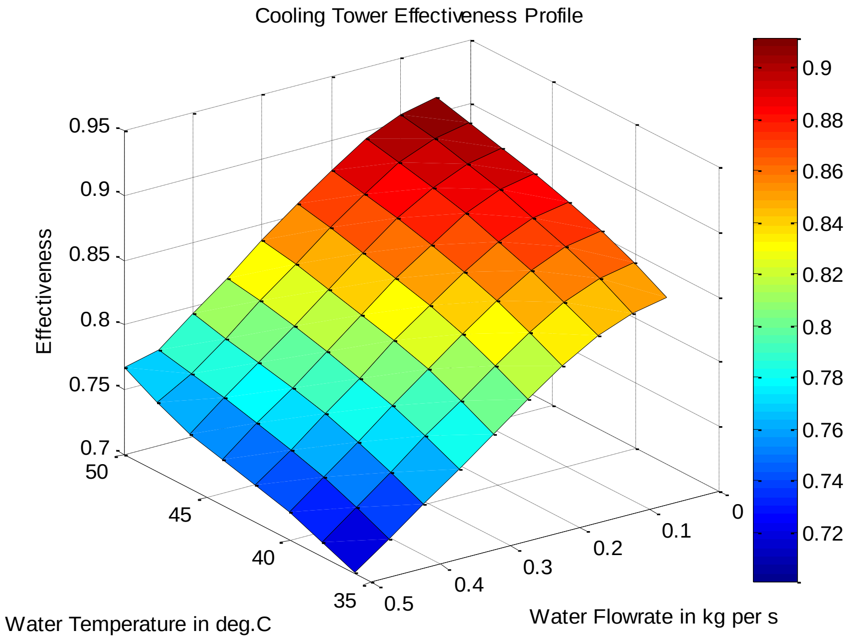

Figure 4 - 2, Cooling tower performance, effectiveness profile

4.2.3.1 Discussion

Table 4-7 above presented the comparison with experimental data, and as seen in the table, the proposed model is accurate to within 1.5 % of the experimental data. The effectiveness and cooling tower outlet water temperature profiles shown in Figures 4-1 and 4-2 indicate, as expected, that the cooling tower effectiveness increases with an increase in the inlet water temperature together with a decrease in the circulating water flow rate. This is the same behaviour as found in literature (Kim & Smith, 2001). The model was thus deemed accurate to allow for the modelling of the comprehensive utility system.

4.2.3.2 Evaporation loss

It was mentioned in Chapters 3 and 4 that the cooling tower model of Kim & Smith (2001) as used in this work was modified to predict the evaporation loss in the cooling tower. Table 4-8 below shows the calculated evaporation rates for the data of Bernier (1994). Unfortunately Bernier (1994) did not report the evaporation losses in his experimental data and thus there was no available literature data to compare the predicted losses to. However for the industrial scale cooling tower used as an

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illustrative case in this work (See following Section), the evaporation losses were available and thus comparison could be made. The results are also shown in Table 4-8.

Table 4 - 8, Evaporation losses from the cooling tower model

		Data of Bernier (1994)				*Industrial case*
Experimental data	**Unit**	**1**	**2**	**3**	**4**	**1**
Air flowrate	kg/s	0.670	0.664	0.665	0.656	60.000
water flowrate	kg/s	0.200	0.398	0.775	0.300	583.330
water inlet temperature	^oC	36.7	32	29.3	27.9	38
water outlet temperature	^oC	19.8	20.4	20.7	20.8	27
Calculated evaporation loss	kg/s	0.005171	0.007064	0.010197	0.003259	9.7195
Actual evaporation loss						9.514
% error						**2.16%**

As seen in the above table, a relative error of less than 2.2% in the evaporation loss was obtained. This was deemed acceptable for evaluating the performance of the comprehensive utility system, given the general accuracy of empirical equations similar to the one used in this work as given by Perry & Green (1997).

4.2.3.3 Application to industrial scale cooling tower

The versatility of the cooling tower model was also examined by modelling an industrial size cooling tower from a South African Chemical company. Details of the cooling tower are shown in Table 4-9 and the comparison with model data is shown in Table 4-10.

4.2.3.3.1 Discussion

The comparison results shown in Table 4-10 indicate an overall error in the calculated water outlet temperature of 0.4%. This result shows that the model used in the work is thus versatile, being able to predict the behaviour for very small cooling towers of the scale considered by Bernier (1994) together with that of big cooling tower as found in typical industrial applications.

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Table 4 - 9, Industrial cooling tower details

Description	***Value***	***Units***
Wet bulb temperature	21	^oC
dry bulb temperature	25	^oC
Water inlet temperature	38	^oC
Water outlet temperature	27	^oC
Design heat duty	53720	MW
Design water flow	4200	m^3/h
normal flow	3600	m^3/h
Fill	polypropylene (splash)	
Number of cells	2	
overall dimensions (L x W x H) , m	26.45 x 12.7 x 24.05	
drift loss	<0.02 of circulating flowrate	
circulating water	4170516	kg/h
Fan data		
Type	Variable speed drive	
Fan motor power	132	kW
Number of motors	2	
number of fans per cell	1	
maximum air volume per fan	1508400	m^3/h
Design air outlet	34.2	^oC
Total Fan power	202	kW
power per fan	101	kW
Efficiency	79%	
Fill data		
H x L x W	4.6 x 12.75 x 12	
Total volume	1407.6	
Design L/G	1.53	
E	68.5	m^3/h

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Table 4 - 10, Model results for industrial cooling tower details

Description	***Model results***	***Units***
Water flowrate	579.23	kg/s
Air flowrate	60	kg/s
CW inlet temperature	38	^{o}C
CW outlet temperature	21	^{o}C
Model result CW inlet temperature	37.9901	^{o}C
Model result CW outlet temperature	27.1	^{o}C
Error in CW inlet temperature	0.026	%
Error in CW outlet temperature	0.370	%

4.2.3.4 Cooling tower optimisation

The cooling tower model discussed in Chapter 3 and in the preceding section was used to optimise a cooling tower as discussed in Section 3.6. The optimisation was done in the GAMS® environment while the solution of the differential equations was done in Matlab ® as discussed above. The two programming applications were used concurrently following the method of Ferris (2005). Figure 4-3 below shows the flowchart for the optimisation program. The aim of the optimisation was to determine the optimum circulating water flowrate (and hence water return temperature) for a specified cooling tower configuration and fixed air flowrate. The objective of the optimisation was to maximise the cooling tower effectiveness as discussed in Section 3.3.

The cooling tower optimisation starts by guessing the optimum inlet water temperature and hence flowrate (from the knowledge of the cooling tower duty), from where the detailed cooling tower ordinary differential equations are solved in Matlab as discussed in Section 3.6.5 in Chapter 3. The cooling tower outlet conditions are then used as input into the GAMS® optimisation model from which new values of the inlet water temperature and flowrate are obtained. The new values are compared to the initially assumed values and if the calculated error is greater than a predetermining level of accuracy, the initially assumed values are adjusted and the whole sequence of operations repeated until convergence.

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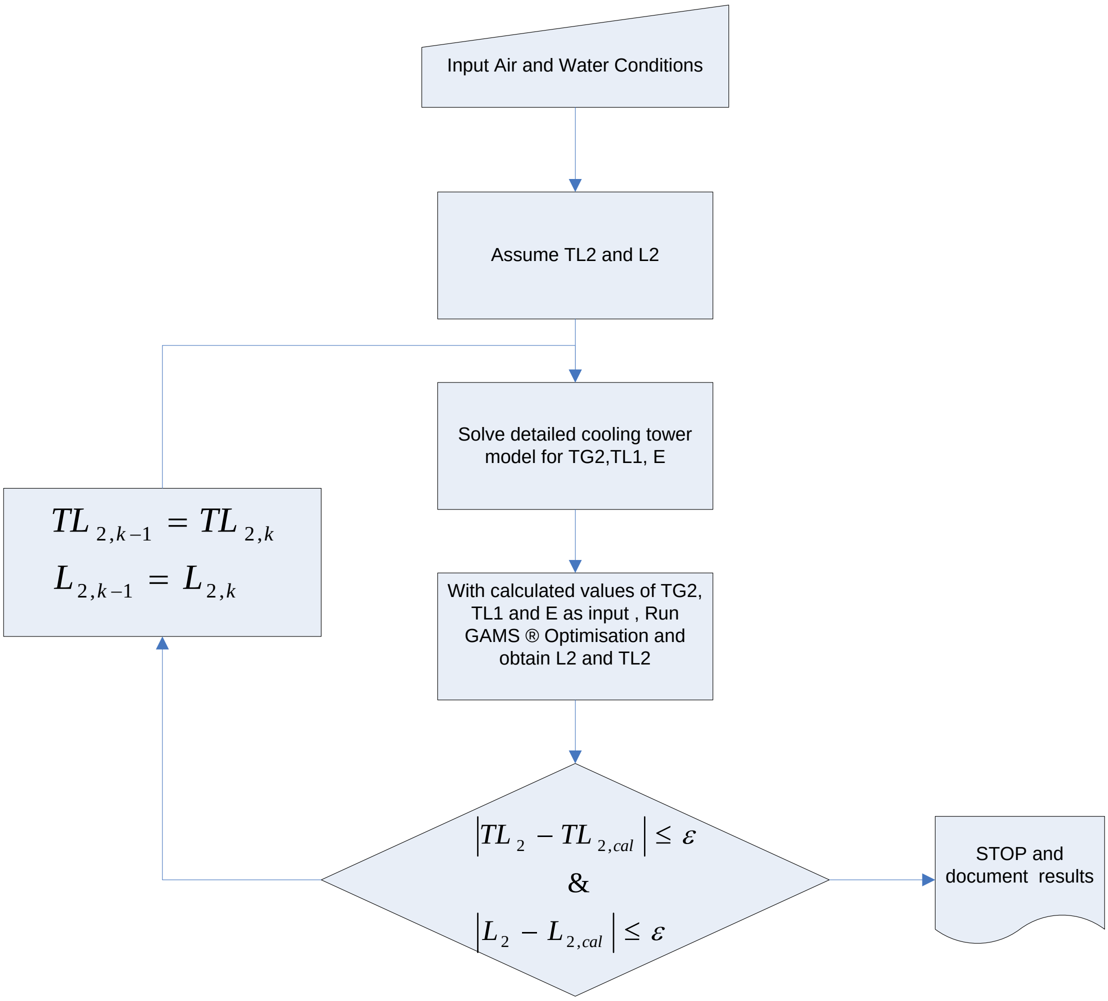

Figure 4 - 3, Cooling tower optimisation chart: Interfacing GAMS® and Matlab ®

Two additional constraints used in the optimisation model were that the cooling water return temperature be less than a certain maximum (in this case 45.2 ^{o}C), i.e.

$$TL_2 \leq T^{Max} \qquad (4\text{-}1)$$

and also above a certain minimum, which was 25 ^{o}C:

$$TL_2 \geq T^{Max} \qquad (4\text{-}2)$$

Table 4-10 below shows the optimisation results.

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Table 4 - 11, Optimisation results – cooling tower optimisation

	Before optimisation	***After optimisation***
Final value of $Z(x^*)$	0.8230	0.8853
Number of iterations	N/A	12
Variables	N/A	26
Equations	N/A	32
Circulating water flowrate (kg/s)	0.2	0.1298
Return Water temperature (^{o}C)	36.7	45.2

4.2.3.4.1 Discussion

The results show that the performance of the cooling tower improved, with the effectiveness increasing from 0.823 to 0.8853 and the circulating water flowrate reducing by almost 40 % to 0.1298 kg/s. In the same process the inlet water temperature increases from 36.7 to 45.2 ^{o}C.

The accuracy of the modelling approach in terms of the use of the simultaneous GAMS® and Matlab® modelling environments as described above was tested by developing the cooling tower optimisation model wholly in Matlab® and comparing the results to those from the combined approach. This was carried out so as to have faith in the interfacing approach of the two optimisation environments and Table 4-12 below shows the results using the first dataset from the experimental data of Bernier (1994) shown in Table 4-7.

As seen from the Table both modelling approaches give virtually the same results, thus the modelling approach used in this work where the two programming platforms were used simultaneously was deemed accurate to allow for the optimisation of the comprehensive utility system.

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Table 4 - 12, Comparison of the effectiveness of interfacing GAMS® and Matlab, cooling tower optimum results

	Initial system	***GAMS® with Matlab ®***	***Matlab ® optimisation***
Final value of $Z(x^*)$	0.8230	0.8853	0.8854
Number of iterations	N/A	12	
Variables	N/A	26	
Equations	N/A	32	
Circulating water flowrate (kg/s)	0.2	0.1298	0.1310
Return Water temperature (^{o}C)	36.7	45.2	45.2

Figure 4-4 below shows the variation of the cooling water temperature along the height of the cooling tower. Similar plots for the humidity and the air temperature profiles as predicted by the model are also shown in Figure 4-5 and 4-6.

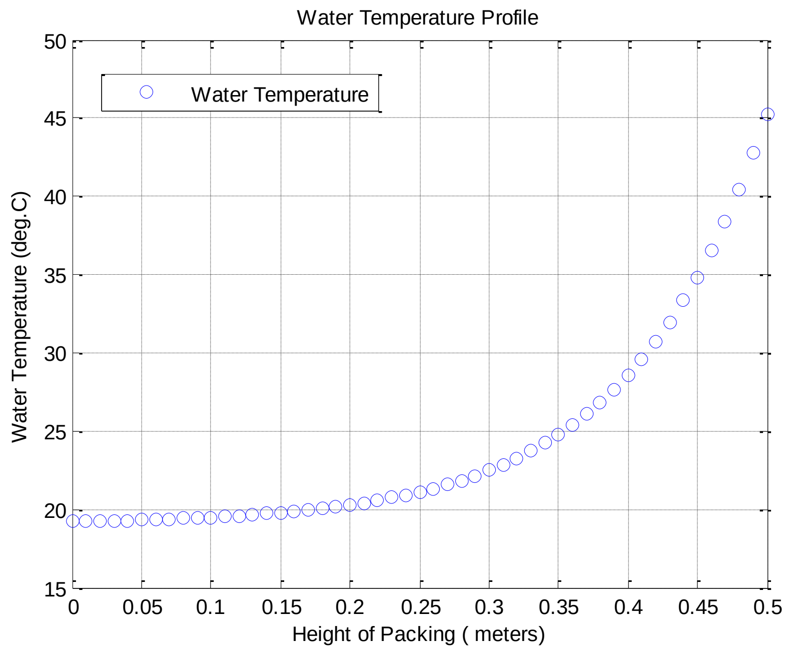

Figure 4 - 4, Cooling water temperature profile inside the cooling tower

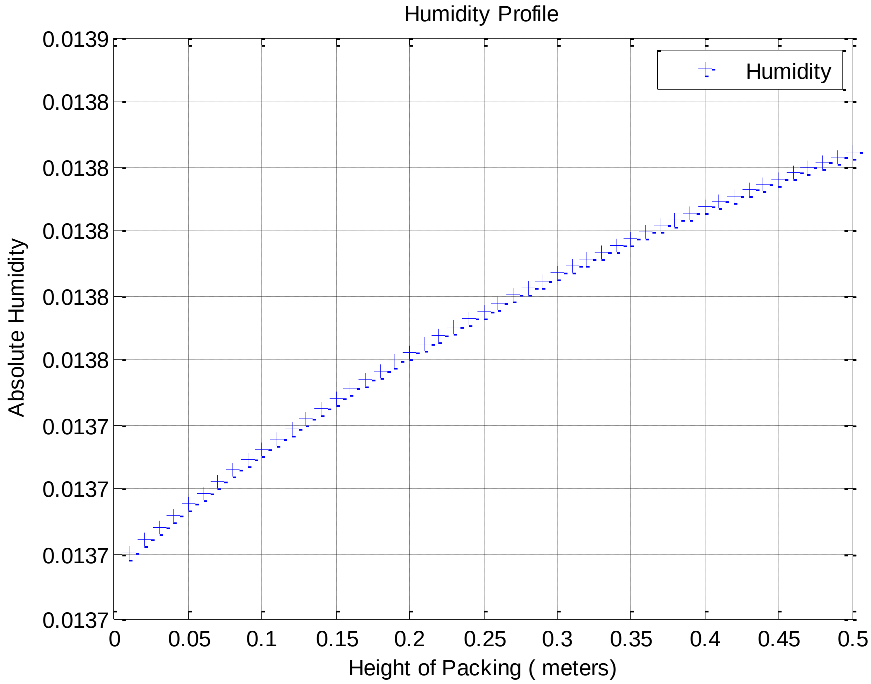

Figure 4 - 5, Absolute humidity profile inside the cooling tower

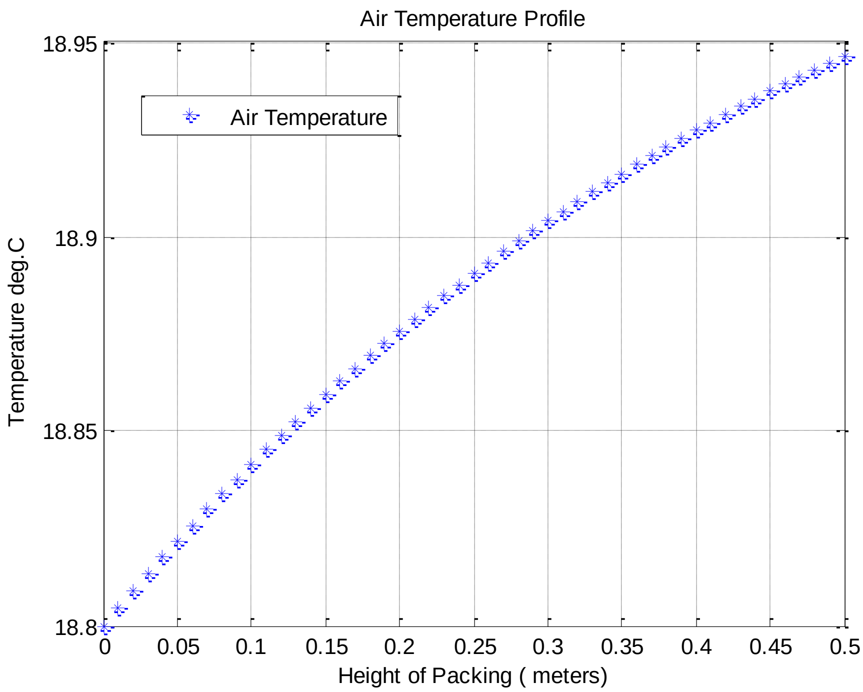

Figure 4 - 6, Air temperature profile inside the cooling tower

The foregoing profiles show as expected that the water temperature decreases down the cooling tower, with the air temperature and humidity also decreasing. Figure 4-4 , however shows that the water temperature decreases considerably in the top half of the cooling tower, which could be explained as being a result of the high temperature driving force between the water and air in that section. Down the cooling tower, as the water gets colder the driving force for heat transfer decreases, hence the drop in the cooling rate.

4.3 Comprehensive system optimisation - simple cycle with single steam turbine (Case 1a)

In this section the results from the optimisation of the comprehensive utility system are presented. For reference purposes, Figure 4-7 below, already presented earlier, is reproduced showing the comprehensive system.

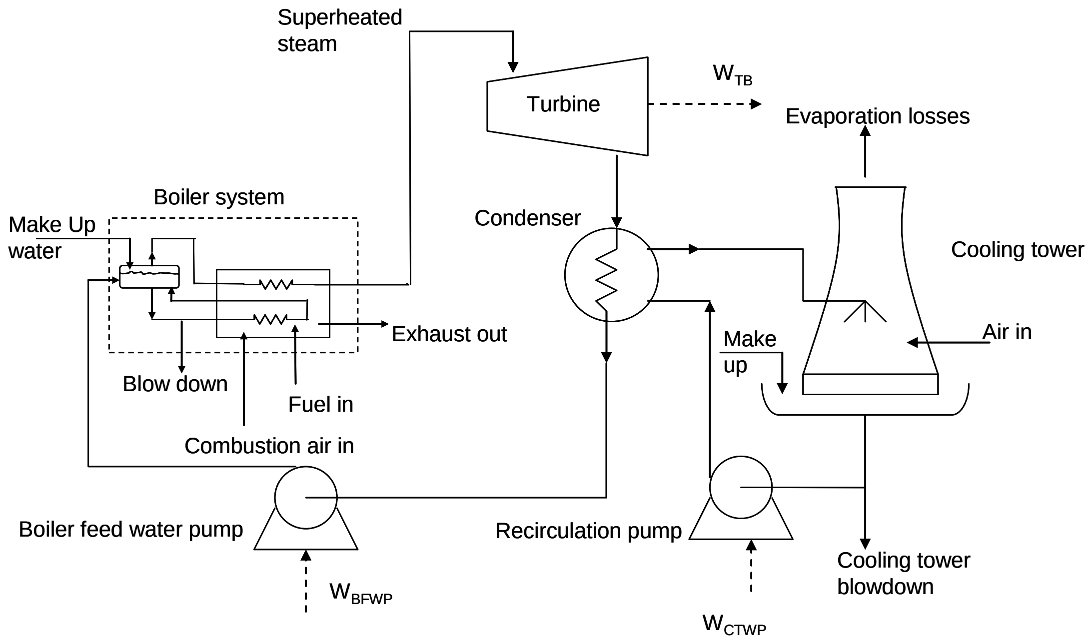

Figure 4 - 7, Simple Rankine cycle for power generation

4.3.1 Problem statement

The problem statement for this case is:

Given the prevailing operating conditions of a simple power plant similar to that shown in Figure 4-7, together with the relevant sizes of the steam and cooling water systems:

(i) Determine the optimum operating conditions for the overall comprehensive utility system.

4.3.2 Illustrative case

As an illustrative case, the cooling tower experimental data of Bernier (1994) were matched with the corresponding steam system data to give a comprehensive system dataset which was used as a basis for the analysis. Table 4-13 below shows the input data set for the comprehensive utility system. Four data sets are shown differing in the cooling tower circulating water flowrates and hence condensation temperatures.

Table 4 - 13, Comprehensive system input data set for simple steam cycle

Cooling tower system data					
Experimental data	**Unit**	**1**	**2**	**3**	**4**
Air flowrate	kg/s	0.669	0.656	0.664	0.658
Water flowrate	kg/s	0.200	0.300	0.397	0.495
Water inlet temperature	^oC	36.7	32	29.3	27.9
Water outlet temperature	^oC	19.8	20.4	20.7	20.8
Make up flowrate	kg/s	0.012	0.014	0.011	0.013
Blowdown flowrate	kg/s	0.007	0.006	0.007	0.005
Effectiveness		0.8230	0.7404	0.6692	0.6067
Heat duty	kW	14.3	14	12.5	11.6
Calculated heat duty	kW	14.1284	14.5464	14.2793	14.6906
Wet bulb temperature	^oC	15.8	16	16	15.9
Steam turbine system data					
Turbine inlet steam flowrate	x 10^3kg/s	7.089	7.298	7.164	7.37
Turbine Inlet temperature	K	773.15	773.15	773.15	773.15
Turbine outlet temperature	K	314.85	310.15	307.45	306.05
Condenser outlet temperature	K	314.85	310.15	307.45	306.05
Cooling water temperature	K	292.95	293.55	293.85	293.95
Work output	kW	6.273	6.603	6.564	6.797
Turbine inlet pressure	bar	86	86	86	86
Turbine outlet pressure	bar	0.081	0.063	0.054	0.05
Condenser outlet pressure	bar	0.081	0.063	0.054	0.05
Thermal efficiency		0.309	0.313	0.316	0.317
Condenser duty	kW	-14.128	-14.546	-14.279	-14.69

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The above data was used as input to optimise the comprehensive utility system first by following the traditional individual system optimisation approach and then by optimising the system in an integrated approach as presented in this work. In the discreet modelling approach, the cooling tower was optimised first to give the optimum cooling tower conditions, which were then used to optimise the power block (as would typically be done in individual optimisation). The results from the two approaches are presented in the following sections.

4.3.3 Results and discussion

Figure 4-8 below shows the performance of the comprehensive system.

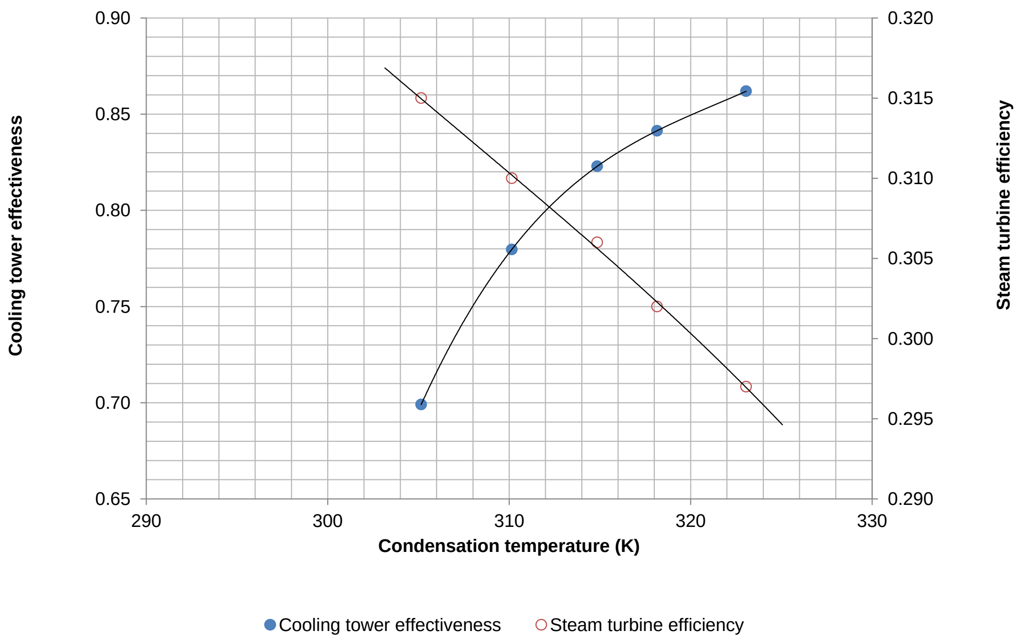

Figure 4 - 8, Comprehensive utility system performance

The data shows the effect of treating the system as a single integrated utility system as opposed to individual system component models. The comprehensive system model accurately predicts that the steam turbine efficiency decreases with increasing cooling tower effectiveness (Figure 4-8), a situation which is well borne out in practice as an increase in the cooling tower effectiveness occurs when conditions at the inlet of the cooling tower are high temperature and low flowrate. Consequently,

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the resultant high cooling tower inlet temperature translates to a high condensation temperature in the Rankine cycle and hence a reduction in the steam turbine efficiency. Figure 4-9 below illustrates this point on a temperature – enthalpy diagram.

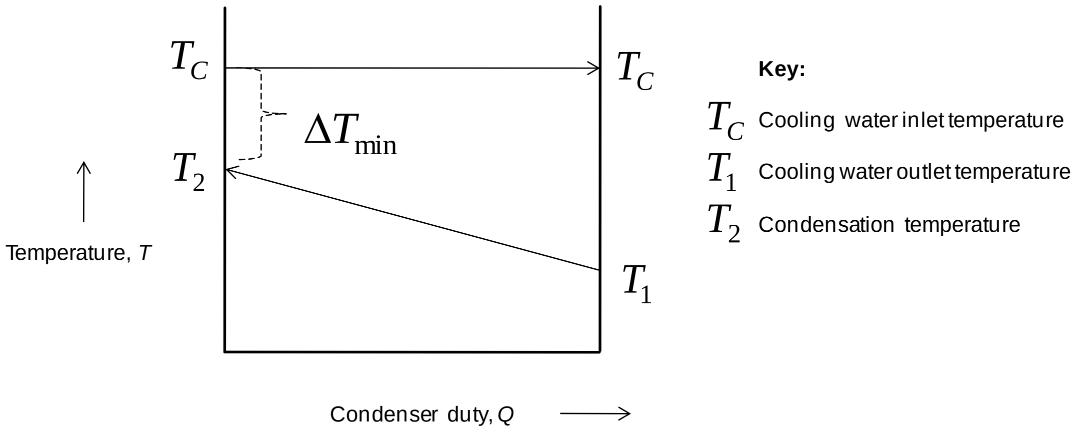

Figure 4 - 9, Condenser temperature profile

Table 4-14 below shows the results from the optimisation of the comprehensive system obtained with the cost performance index (Cost function as presented in Section 3.3.3) for the four data sets as presented in Table 4-13.

Looking at the first dataset, the net annual profit before optimisation was 9254 (unit cost). By optimising the components of the comprehensive utility system separately the overall profitability decreases to 8949, although accompanied with an increase in the cooling tower effectiveness. On the other hand, an integrated optimisation approach as outlined in this work gave an overall profitability of 13146 which corresponds to 42.06 % improvement when compared to the un-optimised system. In the process, the cooling tower effectiveness decreases by 19.8% as opposed to that from separate optimisation (increases by 7.47%) while the thermal efficiency increases by 1.4 %.

Comparing the two optimisation approaches shows that the integrated approach overall gives about 44% more profitability when compared to the separate optimisation approach. The corresponding result in terms of the performance of the two sub utility systems is that the cooling tower effectiveness decreases by 25.4 % with the thermal efficiency increasing by 4.5 percentage points.

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Similar behaviour is obtained from the other data sets and thus the conclusion is that even though the effectiveness of the cooling tower increases and thus the cooling tower operates optimally, the overall profitability of the system decreases when the individual parts of the utility system are optimised separately. This confirms the point that individual components optimisation does not guarantee overall system optimality.

It should be noted that the observed behaviour is dependent on the values of the unit costs used in the objective function, values c_1, c_2 and c_3 in Equation 3-16. In this work the values 0.001 \$/MJ, 0.01 \$/m^3 and 0.2 \$/kWh were used respectively for the unit costs.

Table 4 - 14, Comprehensive system performance

Comprehensive system PI (Unit cost)				
Dataset	**1**	**2**	**3**	**4**
Before optimisation	9 254	9 634	9 448	9 675
Separate optimisation	8 949	9 214	9 044	9 305
This work (total optimisation)	13 146	13 139	13 144	13 141
% Change in *PI* for separate optimisation	-3.30	-4.36	-4.28	-3.82
% change in *PI* for total optimisation	**42.06**	**36.38**	**39.11**	**35.83**
% relative change in *PI* between separate and Total system optimisation	46.91	42.60	45.33	41.22
Cooling tower effectiveness				
Dataset	**1**	**2**	**3**	**4**
Before optimisation	0.823	0.740	0.669	0.607
Separate optimisation	0.884	0.882	0.883	0.881
This Work (total optimisation)	0.660	0.656	0.658	0.657
% Change in *PI* for separate optimisation	7.47	19.11	32.02	45.28
% change in *PI* for total optimisation	**-19.78**	**-11.42**	**-1.61**	**8.23**
% relative change in *PI* between separate and Total system optimisation	-25.36	-25.63	-25.47	-25.50
Steam turbine efficiency				
Dataset	**1**	**2**	**3**	**4**
Before optimisation	0.3090	0.3130	0.3160	0.3170
Separate optimisation	0.3000	0.3000	0.3000	0.3000
This work (total optimisation)	0.3134	0.3133	0.3134	0.3133
% Change in *PI* for separate optimisation	-2.91	-4.15	-5.06	-5.36
% change in *PI* for total optimisation	**1.42**	**0.10**	**-0.83**	**-1.16**
% relative change in *PI* between separate and Total system optimisation	4.47	4.43	4.45	4.44

Changing the above values will thus change the profitability, which thus stresses the point that in an operating plant, the unit costs need to be updated regularly, more so when used as input in determining operating conditions for the plant.
Overall, the results showed that it is more profitable to operate a dedicated cooling tower system that serves a steam turbine in a power plant at low effectiveness in favor of improved efficiency and low operating costs. This depends on the unit costs for the different utilities that are supplied to and also produced by the comprehensive utility system.

4.3.4 Other objective functions

The comprehensive system was also modelled using the other two performance indices presented in Section 3.3.3, as objective functions. The results are shown in Table 4-15 for one of the datasets that was used (Data Set 1 in Table 4-13).

A close look at the results in Table 4-15 shows that the calculated optimum performance of the comprehensive utility system depends on the objective function chosen in the optimisation. Using cost as the objective function, an improvement of 42.06% in the overall system PI is obtained with the total optimisation approach. The other two performance indices gave improvements in the PI's of 3.03 % and 5.03 % respectively. (It should be noted that with the *Q/W* PI, which was the ratio of the heat released by the cooling tower to the net power produced by the steam system, the aim of the optimisation was to minimise the PI, hence a negative change as shown in Table 4-15 indicates improvement). Based on the observed changes in the PI's an initial conclusion would be that the best Pi is Cost, followed by *Q/W* and lastly the sum of the efficiency and effectiveness.

However a close look at the calculated profitability based on each of the overall subsystem PI (see Table 4-15) shows that the second PI (the sum of the effectiveness and thermal efficiency has a better net revenue of 12588 Unit costs compared to 12174 Unit costs for the third PI (36.02% vs. 31.56 improvement based on the unoptimised system).

Table 4 - 15, Optimisation results using different PI's

Comprehensive system performance index	$PI = ER - TOR$	$PI = \eta_{th} + \eta_{eff}$	$PI = \frac{Q_R}{W_{Net}}$
Before optimisation	9 254	1.1320	2.252
Separate optimisation *PI*	8 949	1.1844	2.345
This work (total optimisation)	13 146	1.1663	2.139
% Change in *PI* for separate optimisation	-3.30	4.64	4.11
% change in *PI* for total optimisation	**42.06**	**3.03**	**-5.03**
% relative change in *PI* between separate and total system optimisation	46.91	-1.53	-8.78
Calculated profitability based on comprehensive system PI			
Before optimisation	9 254	9 254	9 254
Separate optimisation	8 949	8 943	8 947
This work (total optimisation)	13 146	12 588	12 174
% Change in revenue for separate Optimisation	-3.30	-3.36	-3.32
% change in revenue for total optimisation	**42.06**	**36.02**	**31.56**
% relative change in revenue between separate and total system optimisation	46.91	40.76	36.08
Cooling tower effectiveness			
Before optimisation	0.823	0.823	0.823
Separate optimisation	0.884	0.882	0.883
This work (total optimisation)	0.660	0.867	0.357
% Change in *PI* for separate optimisation	7.47	7.16	7.35
% change in *PI* for total optimisation	**-19.78**	**5.38**	**-56.67**
% relative change in *PI* between separate and total system optimisation	-25.36	-1.66	-59.64
Steam turbine efficiency			
Before optimisation	0.3090	0.3090	0.3090
Separate optimisation	0.3000	0.3000	0.3000
This Work (total optimisation)	0.3134	0.2990	0.3192
% Change in *PI* for separate optimisation	-2.91	-2.91	-2.91
% change in *PI* for total optimisation	**1.42**	**-3.23**	**3.31**
% relative change in *PI* between separate and total system optimisation	4.47	-0.33	6.41

The initial 5.03 % improvement in the third overall system *PI*, thus does not translate into the same improvement when costs are considered, and this stresses the point that costs should always be considered in any optimisation process as some optimum designs based on certain objective functions may not necessarily give the lowest cost. The reduction in the profitability for the third *PI* can be explained in terms of a reduced cooling tower effectiveness which translates to increased circulating water flowrate in turn resulting in increased pumping costs, hence a reduction in overall profitability.

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The results show that the best *PI* for the comprehensive system is the cost. Where cost is not important, the other two PIs can be used to gain more insight into the operation of the system. Overall, the conclusion, as was shown previously, is that better performance of the system is achieved when the system is modelled as a single utility system as opposed to treating the system components individually

4.4 Comprehensive system optimisation - regenerative cycle with a single feed pre-heater (Case 1b)

This section presents the same analysis as that presented in the preceding section (Section 4.3) as applied to the regenerative cycle with a single feed pre-heater that was presented in Figure1-3 of Chapter 1. Figure1-3 is reproduced below for easy reference.

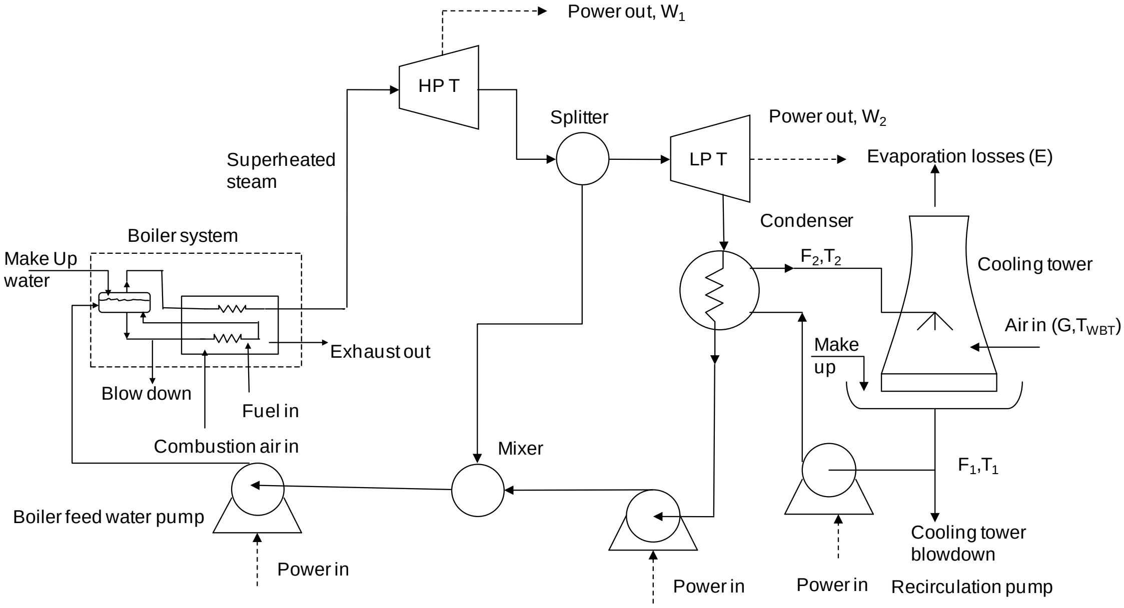

Figure 4 - 10, Regenerative steam turbine cycle with a single feed pre-heater

4.4.1 Problem statement

The same problem statement as given in Section 4.3.1 is applicable, the aim being to determine the optimum operating conditions for the power plant shown in Figure 4-10
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4.4.2 Illustrative case

As was the case with the simple steam turbine model, as an illustrative case, the cooling tower experimental data of Bernier (1994) was matched with the steam system data to give a comprehensive system dataset which was used as a basis for the analysis. Table 4-16 below shows the input data set for the regenerative steam cycle system.

Table 4 - 16, Comprehensive system input data set for regenerative steam cycle

Cooling tower system data						
Experimental data		***Unit***	***1***	***2***	***3***	***4***
Air flowrate		kg/s	0.66944	0.655833	0.663889	0.657778
water flowrate		kg/s	0.2	0.3	0.397222	0.495
water inlet temperature		^{o}C	36.7	32	29.3	27.9
water outlet temperature		^{o}C	19.8	20.4	20.7	20.8
make up flowrate		kg/s	0.01167	0.013889	0.011111	0.013056
Blowdown flowrate		kg/s	0.00731	0.00575	0.006944	0.005472
Effectiveness			0.82297	0.740373	0.669173	0.606723
Heat duty		kW	14.3	14	12.5	11.6
Calculated heat duty		kW	14.1284	14.5464	14.27934	14.69061
Wet bulb temperature		^{o}C	15.8	16	16	15.9
Steam turbine data						
HP turbine inlet steam flowrate	x 10^3	kg/s	9.084	9.396	9.249	9.529
LP Turbine inlet steam flowrate	x10^3	kg/s	7.042	7.256	7.128	7.335
Cooling water supply temperature		K	292.95	293.55	293.85	293.95
HP turbine inlet temperature		K	773.15	773.15	773.15	773.15
LP turbine inlet temperature		K	457.073	451.148	447.739	446.044
LP turbine outlet temperature		K	314.85	310.15	307.45	306.05
Feed preheater outlet temperature		K	441.101	438.649	437.236	436.534
HP Turbine Inlet pressure		bar	86	86	86	86
LP Turbine inlet pressure		bar	7.553	7.11	6.864	6.744
LP turbine outlet pressure		bar	0.081	0.063	0.054	0.05
Steam turbine thermal efficiency		-	0.344	0.35	0.354	0.355
HP turbine work output		kW	-4.422	-4.655	-4.628	-4.792
LP turbine work output		kW	-2.752	-2.932	-2.936	-3.051
Total work output		kW	-7.174	-7.587	-7.564	-7.843
Condenser duty		kW	-14.13	-14.546	-14.28	-14.69
fraction split to feed pre-heater		-	0.225	0.228	0.229	0.23

The rest of the steam system parameters are as shown in Table 4.5 (Section 4.2.2)

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The above dataset was used in the optimisation of the comprehensive system and the results in terms of the comparison between the discrete approach and the combined approach are shown in the following section.

4.4.3 Results and discussion

Table 4-17 below shows the model results following the above discussion. As seen from the table, the combined treatment of the system results in up to 13.8 % improvement in revenue when compared to the discrete optimisation where the cooling tower is optimised first and then the steam system.

Table 4 - 17, Model results for comprehensive system optimisation with a regenerative steam cycle.

Comprehensive system PI (Unit cost)				
Dataset	**1**	**2**	**3**	**4**
Before optimisation	10 630	11 113	10 923	11 193
Separate optimisation	10 190	10 491	10 298	10 596
This work (total optimisation)	12 094	12 090	12 092	12 091
% Change in *PI* for separate Optimisation	-4.14	-5.59	-5.72	-5.33
% change in *PI* for Total optimisation	13.77	8.79	10.70	8.02
% relative change in *PI* between separate and total system optimisation	**18.68**	**15.23**	**17.42**	**14.10**
Cooling tower effectiveness				
Dataset	**1**	**2**	**3**	**4**
Before optimisation	0.8086	0.7250	0.6466	0.5917
Separate optimisation	0.88445	0.88188	0.88342	0.88147
This work (total optimisation)	0.71746	0.71314	0.71582	0.71406
% Change in *PI* for Separate optimisation	9.38	21.64	36.62	48.98
% change in *PI* for total optimisation	-11.27	-1.64	10.70	20.69
% relative change in *PI* between separate and total system optimisation	**-18.88**	**-19.13**	**-18.97**	**-18.99**
Steam turbine efficiency				
Dataset	**1**	**2**	**3**	**4**
Before optimisation	0.344	0.350	0.354	0.355
Separate optimisation	0.333	0.333	0.333	0.333
This work (total optimisation)	0.352	0.352	0.352	0.352
% Change in *PI* for separate optimisation	-3.20	-4.86	-5.93	-6.20
% change in *PI* for total optimisation	2.22	0.43	-0.68	-0.97
% relative change in *PI* between separate and total system optimisation	**5.59**	**5.56**	**5.58**	**5.57**

An average of 16.4 % improvement in profitability is obtained by the total optimisation approach followed in this work when compared to the separate optimisation approach. The improvement is accompanied by an average of 19% reduction in effectiveness and 5.6 % improvement in the thermal efficiency.

When the regenerative cycle optimisation results are compared to those of the simple cycle it is seen that the simple cycle giver better profitability. Looking at the first dataset in Table 4-14, the total optimisation approach gave net profitability of 13 146 unit costs which compares to 12 094 for the regenerative cycle. The main reason for the difference is that less power is produced by the regenerative cycle although at a higher efficiency (35.2% compared to 31. 34 for the two systems respectively).

As was the case for the single steam turbine model, the results also show that an increase in the cooling tower effectiveness does not result in an increase in the overall plant profitability. Instead, the operation of the power plant is more profitable if the effectiveness is reduced. This result further illustrates the point that individual system optimisation does not necessarily give optimum results.

4.4.4 Other objective functions

The regenerative cycle was also modelled using the other two performance indices presented in Section 3.3.3, as objective functions, including the thermal efficiency and effectiveness as objective functions. The results are shown in Table 4-18 for the first dataset in Table 4-16.

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Table 4 - 18, Optimisation results for other objective functions

Objective function PI		$ER-TOR$	$\eta_{th}+\eta_{eff}$	Q_R/W_{Net}	η	ε
Parameter	**Unit**					
Overall system PI		**1.465**	**1.230**	**1.855**	**0.358**	**0.895**
Relative profitability	**%**	**11.00**	**4.71**	**7.15**	**7.36**	**0.00**
Performance Index		1.46488	1.36824	1.40419	1.40732	1.30378
Effectiveness		0.71746	0.89809	0.50471	0.50471	0.89500
Thermal efficiency		0.35163	0.33182	0.35799	0.35800	0.30993
Cooling tower flowrate	kg/s	0.38006	0.14313	0.80924	0.81105	0.15279
make up	kg/s	0.00767	0.00766	0.00731	0.00730	0.00819
Blowdown	kg/s	0.00192	0.00192	0.00183	0.00183	0.00205
Steam flowrate	* 10^3kg/s	10	10	10	10	10
regeneration steam fraction	-	0.2164	0.2192	0.2332	0.2314	0.1704
Cooling tower inlet temperature	^oC	29.8	45.2	25.0	25.0	45.2
Condenser duty	kW	15.6990	15.6925	15.3503	15.3847	16.6942
Cooling tower outlet temperature	^oC	293.0443	292.1258	293.6076	293.6076	292.2159
Total work output	kW	-8.2381	7.5399	-8.2849	-8.3034	-7.2433
HP turbine Work output	kW	-5.2661	-4.6854	-5.0188	-5.0622	-5.2661
LP turbine work output	kW	-2.9719	-2.8545	-3.2661	-3.2411	-1.9771
HP turbine inlet pressure	bar	86	86	86	86	86
LP turbine inlet pressure	bar	5.67650	8.55201	6.79367	6.58648	5.67650
LP turbine outlet pressure	bar	0.05911	0.13161	0.04523	0.04523	0.27246
HP turbine inlet steam temperature	K	773.15	773.15	773.15	773.15	773.15
LP turbine inlet steam temperature	K	429.78	469.49	446.75	443.78	429.78
LP turbine outlet steam temperature	K	308.93	324.35	304.15	304.15	340.00

As seen in the above table, the best comprehensive system performance index is cost, with a relative improvement in profitability of 11 % with the least being maximising the cooling tower effectiveness. Maximizing the cooling tower effectiveness in the comprehensive system gives the least benefits in terms of the total revenue from the facility. The conclusion again is that better overall system performance is obtained when the utility system is optimised as a single integrated system as opposed to individual treatment. Optimising either system does not give optimum results in the comprehensive system.

4.5 Industrial case study for simple cycle with a single steam turbine

In this section of the thesis, an industrial scale steam power plant producing 60 MW electricity is optimised following the methods described in the previous sections.

4.5.1 Description of system

The power plant is based on the industrial scale turbine and cooling tower already presented in Section 4.2 above. Table 4-19 below shows the operating conditions of the power plant.

Table 4 - 19, Operating conditions of an industrial scale power plant

Description	***Plant value***	***Units***
Power	60	Mwe
vacuum	10	kPaa
Flow	258	ton/h
HP steam temperature	430	^oC
Pressure	40	barg
Boiler efficiency	82.56	%
Condenser duty	161.16	MW
Turbine isentropic efficiency	85	%
Turbine mechanical efficiency	95	%
Cooling tower flowrate	12600	m^3/h
Cooling tower inlet temperature	38	^oC
Cooling tower outlet temperature	27	^oC
Evaporation	205.5	m^3h

The aim of the optimisation, as presented previously, was to determine the optimum operating conditions for the entire power plant. The results and discussion are presented in the following section.

4.5.2 Results and discussion

Table 4 -20 gives the model results for the industrial scale power plant optimisation using cost as the objective function.

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Table 4 - 20, Model results for the industrial scale power plant

Parameter	*Unit*	*Individual optimisation*	*Comprehensive optimisation*
Overall system PI (Unit cost)		**86 369 076**	**88 512 257**
% Change in PI	**%**		**2.48**
Total power output	*Mwe*	59.017	61.752
Cooling tower water inlet temperature	^{o}C	45.2	36.3
Cooling tower water outlet temperature	barg	26.632	26.968
Condenser duty	MW	161.16	145.864
Thermal efficiency	%	0.238	0.246
Effectiveness	%	0.767	0.610

The above results show that the comprehensive treatment of the industrial system resulted in 2.5 % improvement in profitability, with the simultaneous reduction in the cooling tower effectiveness and corresponding increase in the steam turbine thermal efficiency. It can thus be concluded that individual system optimisation does not guarantee optimality for the overall system.

4.6 Comprehensive system optimisation - Design of a simple steam cycle and a regenerative steam cycle (Cases 2a and 2b)

This section presents the results from the second part of the research problem, viz the design of both a simple steam cycle and a regenerative steam cycle.

4.6.1 Problem statement

The problem statement for this case is:

Given the input specifications and parameters of both simple and regenerative power plants similar to those shown in Figures 4-7 and 4-10 respectively:

(i) Determine the optimum sizes of the cooling tower and steam systems for a given power output.

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4.6.2 Illustrative cases

Presented first are the results from the sizing of the cooling tower and steam systems for 50MW heat load and 50 MWe electricity systems (with and without feed preheating). This was done to illustrate the applicability of the individual system design models that were used in this work. The systems were sized following the approach presented in Sections 3.4.9 and 3.6.7 in Chapter 3.

Table 4-21 gives the input parameters for the design models.

Table 4 - 21, Comprehensive system design input parameters

	Value	***Units***
Ambient dry temperature	18.3	^oC
Wet bulb temperature	15.8	^oC
Atmospheric pressure	1.01325	bar
Turbine Inlet pressure	100	bar
Turbine Inlet temperature	500	^oC
Turbine mechanical efficiency	100	%
Turbine isentropic efficiency	80	%
Combustion efficiency	95	%
Type of fill	Splash	-
Cycles of concentration	4	
Minimum cooling water return temperature	50	^oC

The input information in Table 4-21 was used in the models to give the results shown in Tables 4-22 and 4-23 for the steam turbine system and cooling system respectively. The thermal efficiency was maximised in the steam system design while the effectiveness was maximised in the cooling system design

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Table 4 - 22, Design results of a 50 MWe simple Rankine cycle and a 50 MWe regenerative Rankine cycle

Description	Simple Rankine cycle	Regenerative Rankine cycle	Units
HP Turbine inlet steam flowrate	62.312	61.494	kg/s
LP turbine inlet steam flowrate		55.221	kg/s
Cooling water supply temperature	313.15	313.15	K
HP Turbine inlet steam temperature	773.15	773.15	K
LP turbine inlet steam temperature		444.61	K
HP turbine inlet pressure	20	20	bar
LP turbine inlet pressure		1.545	bar
Condenser Pressure	0.124	0.124	bar
thermal efficiency	0.258	0.288	
HP Turbine power output	-50	-31.055	MW
LP turbine power output		-18.945	MW
condenser duty	-143.343	-124.716	MW
fraction extracted		0.102	

Table 4 - 23, Cooling system results for a 50 MW duty

Description	***Value***	***Units***
Effectiveness	0.870014	
Circulating water flowrate	471.2492	kg/s
Cooling tower inlet temperature	45	^oC
Water outlet temperature	19.59	^oC
Make up flowrate	24.413	kg/s
Air flowrate	734.94	kg/s
Fill cross sectional area	314.167	m^2
Fill height	1.335	M

The above results show that for a 50 MWe simple Rankine cycle system, with the parameters shown in Table 4-21 , the HP steam requirement would be 62.312 kg/s with a thermal efficiency of 25.8%. These values compare to 61.494 kg/s and 28.8 % for the regenerative cycle. The 3% improvement in thermal efficiency is a result of 10.2 % steam extracted from the turbine and used in the boiler feed water preheater at a pressure of 1.55 bar. The cooling system results on the other hand show that for a 50 MW duty cooling tower, the required water and air flowrate would be 471 kg/s

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and 735 kg/s respectively. The associated cooling tower effectiveness would be 87% and the cooling tower dimensions would be a total fill area of 314.2 m^2 and a fill height of 1.33 m.

As a further illustrative case, a comprehensive power plant utility system was designed for two cases, the first a 50 MW regenerative power cycle and the second 60 MW simple power cycle. The results are presented and discussed below. Design results for a 50 MWe regenerative steam cycle with the same operating conditions as those shown in Table 4-5 above are given below (Table 4-24). The table shows that cooling tower cross sectional area of 931.25 m^2 is required together with air and water flowrates of 1396.9 kg/s and 1303.7 kg/s respectively. The required high pressure steam flowrate is 60.91 kg/s while the fraction of steam extracted from the turbine is 9.22 % wt.

Table 4 - 24, regenerative power cycle design results for a 50 Mwe plant

Parameter	***Value***	***Unit***
Performance Index	80 174 989	$/year
Steam turbine efficiency	0.2876	-
Cooling tower Effectiveness	0.8534	-
Cooling Water flowrate	1396.9	kg/s
Make up water flowrate	210.2	kg/s
Blowdown flowrate	52.6	kg/s
Cooling tower inlet water temperature	45	^oC
condenser duty	-124.783	MW
Cooling tower outlet temperature	24.52	^oC
High pressure steam flowrate	60.91	kg/s
HP Turbine inlet pressure	20	bar
LP turbine inlet pressure	1.23	bar
LP turbine outlet pressure	0.124	bar
HP Turbine inlet temperature	773.15	K
LP turbine inlet temperature	423.64	K
LP turbine outlet temperature	323.15	K
Air flowrate	1303.7	kg/s
cooling tower volume	1420.47	m^3
HP turbine power output	-32.619	MW
LP turbine power output	-17.381	MW
fraction extracted	0.0922	-
Cooling tower area	931.25	m^2

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The industrial scale simple power plant presented in Section 4.5 was redesigned following the approach presented in this work and the results are compared in Table 4-25.

Table 4 - 25, Simple steam cycle design results for a 60 Mwe plant

Parameter	***Unit***	***Traditional design***	***New Design (This Work)***
Overall system PI		**88 335 192**	**95 812 198**
% Change in PI	**%**	**-**	**8.46**
Cooling Tower Water Inlet temperature	^oC	38	45.0
Cooling tower water outlet temperature	barg	27.000	24.514
High Pressure steam flowate	kg/s	71.667	72.777
Condenser Duty	MW	-145.962	-148.455
Thermal Efficiency	%	**0.248**	**0.238**
Effectiveness	%	**0.648**	**0.854**
Cooling water flowrate	m^3/h	12 600	5 982
Air flowrate	m^3/h	9 050 400	4 655 867
Evaporation	m^3h	205.5	312.3
Cooling tower area	m^2	918.00	1 107.74
Cooling tower height	m	4.60	1.53
fill volume	m^3	4 223	1 690

The above table shows that the new design approach results in 8.46% improvement in profitability when compared to the traditional design approach. The corresponding required high pressure steam flowrate is 72.8 kg/s with a cooling area of 1107.74 m^2 and air and water flowrates of 4 655 867 m^3/h and 5982 m^3/h respectively. The air flowrate is 48.6 % less than for the traditional design and the water flowrate is 52.6 %. However, the fill crossectional area increased by 21% from 918 m^2 to 1107.74 m^2

Also evident is the improvement in thermal effectiveness which is accompanied by a reduction in the thermal efficiency. It is important to note that when the new design approach is followed in the synthesis of new power plants one is able to end up with a design giving higher profitability, even if that occurs at reduced steam turbine thermal efficiency.

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In conclusion, a new approach has been presented for the design of new power plants which takes into account both the cooling tower and steam system in a single approach. In designing the comprehensive utility system, the cooling tower and steam systems sizes are determined to maximise the net profit from the operation of the power plant. The model gives for a required power output and for given inlet conditions and ambient conditions the optimum steam turbine circulating flowrate, the circulating water and air flowrate for the cooling tower together with cooling tower dimensions. Both the optimisation and design approaches presented in this work give better results when compared to traditional approaches.

4.7 Concluding remarks

This chapter looked at the application of the models that were developed in this work. The chapter began by validating the models through comparison of model predicted values to literature values. The comparisons indicated good agreement with the literature data. This fact, together with the ability of the models to describe real industrial plant equipment indicated that the models were reliable and versatile to be able to be used in the optimisation of the comprehensive utility system. The validated and verified models were then used to optimise first a simple steam cycle and then a regenerative steam turbine cycle in a power plant. The results from both cases indicated that better system performance is obtained by treating the system as an integrated utility system. In the cases considered up to 42% improvement in profitability was obtained for a single steam turbine cycle in the combined treatment as opposed to individual treatment. For a regenerative cycle, up to 13.8 % improvement was obtained.The relevance of the findings was further demonstrated in the evaluation of a real power plant producing 60 Mwe. The analysis predicted that 2.5% improvement in profitability is found from a holistic treatment of the system. Lastly the comprehensive system model was used to size both the cooling tower and steam systems for a required power output. Such a way of sizing the system thus indicated that the research findings are not only important in optimising existing facilities but can also be applied in the synthesis of new power plants. When compared to traditional approaches, both the optimisation and design approaches presented in this work give better results.

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4.8 References

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5

Chapter Five

5. Conclusions and recommendations

This section presents a summary of the work done in the study together with a summary of the findings and recommendations for further study.

5.1 Conclusions

The objective of this study was to develop a method of analysis for a comprehensive power plant utility system taking into account the integrated nature of the system.

After the introductory chapter, Chapter 1, which outlined the research problem and aims of the study, the complete details of the work done in the study were presented. The work started by conducting a comprehensive literature survey, as was presented in Chapter 2 which focused on the general principles of Process Integration and the development of methods and studies for the optimisation of both steam and cooling water systems.

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In the discussion on the general principles of Process Integration, the three elements of process integration were discussed as Synthesis, Analysis and Optimisation and the application of these principles in the design and operation of process equipment was also highlighted. The celebrated Pinch Analysis technique as applicable to Energy Integration and the design of heat exchanger networks was also presented and the discussion ended with a focus on the extension of the pinch principles to Mass Integration and the recently developed Property Integration.

The development of methods and studies for the optimisation of both steam and cooling water systems was then discussed through a selection of studies from literature that were conducted over the years. The review indicated that virtually all of the previous studies did not take into account the combined nature of the utility systems and only focused either exclusively on steam and power systems or on cooling water systems. The few studies that increased the analysis scope beyond the basic utility systems and therefore included some aspects of the background process indicated that such an approach gave better results. The literature study thus pointed a need for the development and analysis of utility systems in a holistic manner and thus the work presented in this report aimed at addressing this oversight by taking advantage of the complementing and synergistic effect of total system analysis as applied in a typical comprehensive utility system found in power plants.

Chapter 3 gave the details of the models that were used in this work and the application of the models was discussed in Chapter 4. The modelling approach followed was to develop individual system models first and later combining these to give a model of the comprehensive utility system. The individual system models were first validated by comparing the model predicted results to literature values and also to industrial scale data. The comparison indicated good fit of the data which gave faith and confidence in the models that were developed. The models were then combined as outlined above and used to optimise two comprehensive utility systems; the first one based on a simple Rankine cycle and the second on a regenerative cycle with a single feed pre-heater. For the single steam turbine system the results showed that up to 42% improvement in profitability is found when the system is treated as a single unit as opposed to discrete analysis. The results for the regenerative cycle showed a similar behaviour with up to 13.8% improvement in overall system profitability.

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The usefulness of the approach developed in this study was further demonstrated in the analysis of an industrial scale power plant producing 60 Mwe. The analysis showed that up to 2.5% profitability is obtained from an integrated system treatment.

Apart from optimising existing systems, the study went on to look into the design and synthesis of the comprehensive utility system, also taking into account the integrated nature of the system. Models were developed for the design of the comprehensive utility systems and these were used in an illustrative case and gave the equipment sizes and different parameters for a required power output. The new design approach outlined in this work gave 8.46% improvement in profitability when compared to traditional methods.

Overall, the work showed that better system performance is obtained from integrated optimisation and analysis. For the power systems considered, optimising the cooling tower and thus increasing the effectiveness resulted in lower performance of the steam system. In conclusion, it is thus advisable to include as much as is possible within existing and operating constraints, of the background process in optimising utility systems. Cooling water and steam systems should not be seen as different and diverse utility systems but should be optimised in a single and total approach.

5.2 Recommendations

The modelling and analysis of the comprehensive utility systems as presented in this thesis revealed to the researcher areas where further work and modifications can be done which could improve both the versatility and applicability of the optimisation methods presented in this work. These areas, presented not in any priority are:

- The use of more robust steam and water physical property estimation methods. In this work ideal gas and ideal liquid behaviours were used. The use of more rigorous thermodynamic equations to describe the water and steam properties is thus recommended, particularly those by the International Association for the properties of water and steam (IAPWS), (Harvey & Parry (1999), Wagner *et al*. (2000).

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- Use of more rigorous and detailed cooling tower models, other than the Merkel and related methods to accurately predict all the cooling tower parameters.

- Inclusion of the actual equipment configurations in the models to allow for the prediction of pressure drops within the model.

- Inclusion of capital cost in the cost objective function for use in designing new plants

- Inclusion of water chemical analysis to assess the effect of increasing the effectiveness of the cooling tower on scale formation and deposition on heat transfer surfaces.

- Extension of the principles outlined in this study to regenerative and reheat cycles for industrial scale power plants including the investigation of the effect of the number and types of feed pre-heaters (viz. Open and closed feed pre-heaters) on the performance of the power plant.

- Extension of the work to include consideration of the use of multiple cooling towers in a power plant, supplying water to a number of steam turbines.

5.3 References

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Nomenclature

1.	ΔH	change in total stream enthalpy (kJ/kg)
2.	$\Delta T_{\min}$	minimum temperature difference (^{o}C)
3.	ΔP	pressure drop across (kPa)
4.	a	area per unit volume (m^2/m^3), parameter in Equation 2-6
5.	A', B', C'	Antoine coefficients
6.	a_{fi}	cooling tower fill cross sectional areal per unit volume
7.	b	parameter in Equation 2-6
8.	B	blowdown flowrate (kg/s)
9.	c_p	specific Heat capacity (kJ/kg.K)
10.	c_{PA}	heat capacity of air (kJ/kgK
11.	c_{PL}	heat capacity of water (kJ/kgK
12.	c_S	humid heat capacity (kJ/kgK)
13.	a_1, b_1, c_1	constant value of heat transfer coefficient equation h_G
14.	a_2, b_2, c_2	constant value of heat transfer coefficient equation, h_L
15.	a_3, b_3, c_3	constant value of mass transfer coefficient equation in k_G
16.	CC	cycles of concentration
17.	CP	heat capacity flowrate (kW/K)
18.	dH	differential increment of enthalpy
19.	dL	differential increment of water flowrate (kg/s m^2)
20.	dT	differential increment of temperature (^{o}C)
21.	dW	differential increment of air humidity
22.	dz	differential increment of cooling tower height (m)
23.	e	cooling tower effectiveness
24.	E	evaporation (kg/s)
25.	F	mass flowrate (kg/s)
26.	G	dry air mass flowrate (kg/m^{2}s)
27.	H	enthalpy flow (kJ/s)
28.	h	specific enthalpy (kJ/kg)
29.	$\mathcal{H}$	humidity ratio (kg/kg dry air)
30.	h_d	mass transfer coefficient
31.	h_f	enthalpy of saturated fluid

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32.	h_g	enthalpy of saturated vapour
33.	h_G	heat transfer coefficient of air (kW/m^2.^{o}C)
34.	h_L	heat transfer coefficient of water,(kW/m^2.^{o}C)
35.	i	enthalpy (J/kg)
36.	K	loss coefficient, mass transfer coefficient
37.	k_G	mass transfer coefficient of air (m/s)
38.	L	water flowrate (kg/s)
39.	M	make up (kg/s)
40.	m_a	air flowrate (kg/s)
41.	m_w	water flowrate (kg/s)
42.	M_{Air}	molecular weight of air
43.	M_e	Merkel Number
44.	M_W	molecular weight
45.	NTU	number of transfer units
46.	P	pressure (kPa)
47.	PI	performance index
48.	P_s	vapour pressure (kPa)
49.	q	stream quality (identical to the liquid mole/mass fraction)
50.	Q	Heat Load (kW), volumetric flowrate in m^3/h
51.	R	universal gas constant (8.314 J/mol.K)
52.	s	specific entropy (kJ/kg)
53.	T	temperature (^{o}C, K)
54.	T^*	shifted temperature
55.	T_r	reduced temperature
56.	T_s	saturation Temperature (^{o}C)
57.	T_H	hot reservoir temperature (^{o}C)
58.	T_C	cold reservoir temperature (^{o}C)
59.	V	volume (m^3)
60.	W	work (kW), humidity (kg water/kg air)
61.	W_{TB}	turbine work
62.	W_1	high pressure turbine work output (kW)
63.	W_2	low pressure turbine work output (kW)
64.	W_c	cooling tower circulating flowrate as in Equation 3-169
65.	W_{BFWP}	power for boiler feed water pump

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66.	W_{CTWP}	power for cooling tower water pump
67.	x	optimisation variable
68.	y	discrete optimisation variable, regenerative cycle fraction
69.	z	fill height (m), Objective function

Superscripts

1.	*	optimum value
2.	‘	equilibrium value as in Equation 2-4
3.	cyc	cycle
4.	ig	ideal gas
5.	L	lower bound
6.	max	maximum
7.	U	upper bound

Subscripts

1.	o	outlet
2.	0	reference conditions
3.	1	cooling tower outlet conditions
4.	2	cooling tower inlet conditions’
5.	a	air
6.	act	actual
7.	$BFWP$	boiler feed water pump
8.	c	cold, condenser, critical property
9.	cal	calculated
10.	$cold$	cold stream
11.	$comb$	combustion
12.	CT	cooling tower
13.	$CTWP$	cooling tower water pump
14.	cw	cooling water
15.	DBT	dry bulb temperature

16.	eff	effectiveness
17.	fan	cooling tower fan
18.	Fil, fi	fill
19.	g	gas phase
20.	G	air
21.	Gen	generated
22.	h	hot
23.	H	high pressure
24.	hot	hot stream
25.	HPT	high pressure turbine
26.	i	interface, inlet, counter, enthalpy
27.	iso	isentropic
28.	j	counter
29.	L	liquid phase, low pressure
30.	LPT	low pressure turbine
31.	m	mean.
32.	ma	air water mixture
33.	$masw$	saturated air water mixture
34.	M	make up
35.	max	maximum
36.	min	minimum
37.	net	net
38.	opt	optimum
39.	out	outlet conditions
40.	s, sat	saturation
41.	SL	saturated liquid
42.	SV	saturated vapour
43.	ST	steam turbine
44.	STP	steam turbine pressure
45.	sw	saturated water
46.	TB	turbine
47.	th	thermal
48.	v	vapour phase
49.	w	water

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50.	WB	wet bulb
51.	WBT	wet bulb temperature
52.	x	liquid phase
53.	y	vapour or gas phase

Greek letters

1.	β	volume expansivity
2.	ε	effectiveness , convergence criterion
3.	η_M	turbine mechanical efficiency
4.	η_{th}	thermal efficiency
5.	ϕ	vapour fraction
6.	λ	latent heat of vaporisation

Abbreviations

1.	BOD	biological oxygen demand
2.	CFB	circulating fluidised bed boiler
3.	CHP	combined heat and power
4.	COC	cycles of concentration
5.	CT	cooling tower
6.	CW	cooling water
7.	DDB	Dortmund databank of thermophysical properties
8.	DFO	derivative free optimisation
9.	EO	equation oriented
10.	GCC	grand composite curve
11.	GA	genetic algorithms
12.	HEN	heat exchanger network
13.	HR	heat rate
14.	IGCC	integrate gasification combined cycle
15.	LCP	linear complimentary problem
16.	MILP	mixed integer liner problem

17.	MINLP	mixed integer nonlinear problem
18.	NLP	nonlinear problem
19.	PI	performance index
20.	QP	quadratic problem
21.	SA	simulated annealing
22.	SQP	sequential quadratic programming
23.	TOR	Total operating revenue/cost
24.	UCG	underground coal gasification