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STEAM SYSTEM OPTIMISATION USING PROCESS INTEGRATION: A FOCUS ON BOILER EFFICIENCY AND PRESSURE DROP

by

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Synopsis

The use of steam in heat exchanger networks (HENs) can be considerably reduced by the application of heat integration and optimisation with the intention of debottlenecking the steam boiler and indirectly reducing the water requirement (Coetzee and Majozi, 2008). The reduction of steam flowrate in a HEN affects the operation of the steam boiler. By reducing the steam flowrate the return condensate temperature to the boiler is compromised which adversely affects the operation of the boiler. A means of maintaining the efficient operation of the boiler whilst still reducing the overall steam flowrate to the HEN is to reheat the return flow to the boiler to a sufficiently high temperature. One means of achieving this is by utilising the sensible heat from the superheated steam leaving the boiler to preheat the boiler feed.

Reusing condensate also has the effect of increasing the pressure drop throughout the HEN. The pressure drop is dependent on the size of the heat exchanger or pipe, the flowrate through the exchanger or pipe and also the HEN layout. This creates an intricate situation that must be accounted for.

Steam systems typically employ turbines to use energy from superheated high pressure steam to generate shaft work. The exhaust of these turbines is usually saturated steam which is frequently used as a utility in the background process. Since turbines operate at various steam levels, a means for incorporating these steam levels into the HEN optimisation framework is necessary. By reducing the amount of steam required for these HENs an opportunity arises for the use of this steam in further preheating the boiler feed water in an attempt to maintain boiler efficiency.

This work involves modelling of the steam system in a mathematical framework and finding the global minimum steam flowrate for the system. The pressure drop throughout the HEN is then also calculated and minimised. The boiler efficiency is maintained by restructuring the HEN and making provision to preheat the boiler feed using synthesis and optimisation. In the event of there not being enough excess heat in the steam system to maintain the boiler efficiency a slight compromise in either boiler efficiency or the minimum flowrate must be made. The initial targeting for the minimum steam flowrate is based on a graphical technique of Coetzee and Majozi, 2008.

Consequently, this dissertation concerns the optimisation and restructuring of all steam system heat exchangers using conceptual and mathematical analysis to create a series/parallel HEN with the aim of reducing the overall steam flowrate, whilst maintaining boiler efficiency and minimising the HEN pressure drop.

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1. Introduction

1.1 Background and Problem Statement

Heat integration has the ultimate goal of reducing external utilities by maximising process to process heat exchange. Linnhoff and Hindmarsh (1982) have developed an extensive methodology to optimise process to process heat exchange. Cooling water systems comprising a cooling tower and associated HEN have been examined in detail and optimised by the likes of Kim and Smith (2001), Majozi and Moodley (2008) and Majozi and Nyathi (2007). They considered the cooling system in a holistic fashion and improved the cooling tower operation by reducing the cooling water flowrate to the HEN. A consequence of this is increased return temperature which is beneficial to cooling tower operation.

Kim and Smith (2003) realised that restructuring the HEN of cooling system as necessitated by the reduction of the cooling water flowrate had the added effect of increasing the overall network pressure drop. They developed an MINLP to find and then minimise the network pressure drop for the previously determined cooling water flowrate.

Coetzee and Majozi (2008) developed graphical targeting technique on a T/H diagram to minimise the steam flowrate to steam systems. In the context of this work a steam system comprised of a HEN and a steam boiler. Using this minimum steam flowrate an appropriate HEN is then designed accordingly. The whole process, including the targeting, can also be done simultaneously using an MILP model. A consequence of this is reduced return temperature.

By incorporating multiple steam pressure levels the graphical targeting technique becomes impractical as it cannot cater for the extra dimension of various steam pressure levels. Thus the MILP model derived by Coetzee and Majozi (2008) is altered to accommodate multiple steam pressure levels and is used to target for a minimum flowrate as well as design the network.

The effects of minimising the steam flowrate on the entire steam system have not as yet been considered. The efficient operation on the steam boiler is dependent on the condensate return flowrate and temperature. Reducing the steam flowrate reduces both of these stream variables and as such affects the steam boiler performance.

A holistic steam system optimisation is required that includes the aspects of boiler efficiency and pressure drop. If possible the boiler efficiency should be maintained and the pressure drop minimised while still reducing the steam flowrate.

1.2 Motivation

Heat integration in steam utility systems can yield tremendous savings in both capital and operating costs for grassroot and retrofit heat exchanger networks by reducing the steam flowrate (Coetzee and Majozi, 2008; Peters and Timmerhaus, 1991). While this idea may seem attractive, the effects on the steam system are as yet undetermined. By affecting the boiler efficiency any savings made by the reduction of the steam flowrate could be compromised by poorer boiler performance. An increased pressure drop may also introduce the need for additional fluid movers which would add to the capital cost of the system.

1.3 Aim

The objective of this work was to create an extension to the steam flowrate minimisation MILP developed by Coetzee and Majozi (2008). This extension will represent a holistic view of the heating system by incorporating the steam boiler as well as other common pieces of equipment such as turbines. Using this model the effects of minimising steam flowrate on the steam boiler are observed and a means of maintaining the boiler efficiency are proposed.

Multiple steam pressure levels are then incorporated into the design framework so that the model can be more flexible. Once again the objective is to maintain the boiler efficiency while minimising the steam flowrate.

The final aspect is ultimately to consider pressure drop in the HEN. The minimum flowrate is determined and then used as a parameter in the design of a network that exhibits the minimum pressure drop.

1.4 Structure

This dissertation is structured as follows. Chapter 2 follows the Introduction and presents a literature review on heat integration, heat exchanger network design and utility system design and optimisation. Chapter 3 presents a brief explanation of the system in hand; a formal definition of boiler efficiency and a background to calculating pressure drop in HEN elements. Chapter 4 contains the mathematical formulations developed to achieve the aims of this study. Chapter 5 presents a case study used to demonstrate the various means of minimising steam flowrate while maintaining the boiler efficiency as well as minimising network pressure drop. Chapter 6 gives conclusions and relevant recommendations of areas for further investigation.

Thereafter follows a list of the nomenclature and an Appendix which completes the dissertation.

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2. Background to the System of Interest

This section is intended to give a background to the mathematical models developed in this dissertation. It contains information on the steam system in the context of the investigation, a derivation of the boiler efficiency as well as a means of calculating pressure drop for various elements found within a HEN.

2.1 System Description

The steam system in the context of this investigation is shown in Figure 2.1. The boiler produces superheated high pressure steam, shown as stream 1. This steam then proceeds to a let down valve as stream 2, or is taken to a high pressure turbine as stream 3. The steam lost during let down is typically collected in flash drums which have been excluded for simplicity. Table 2.1 shows typical steam pressure levels, temperatures and saturation temperatures used for this chapter.

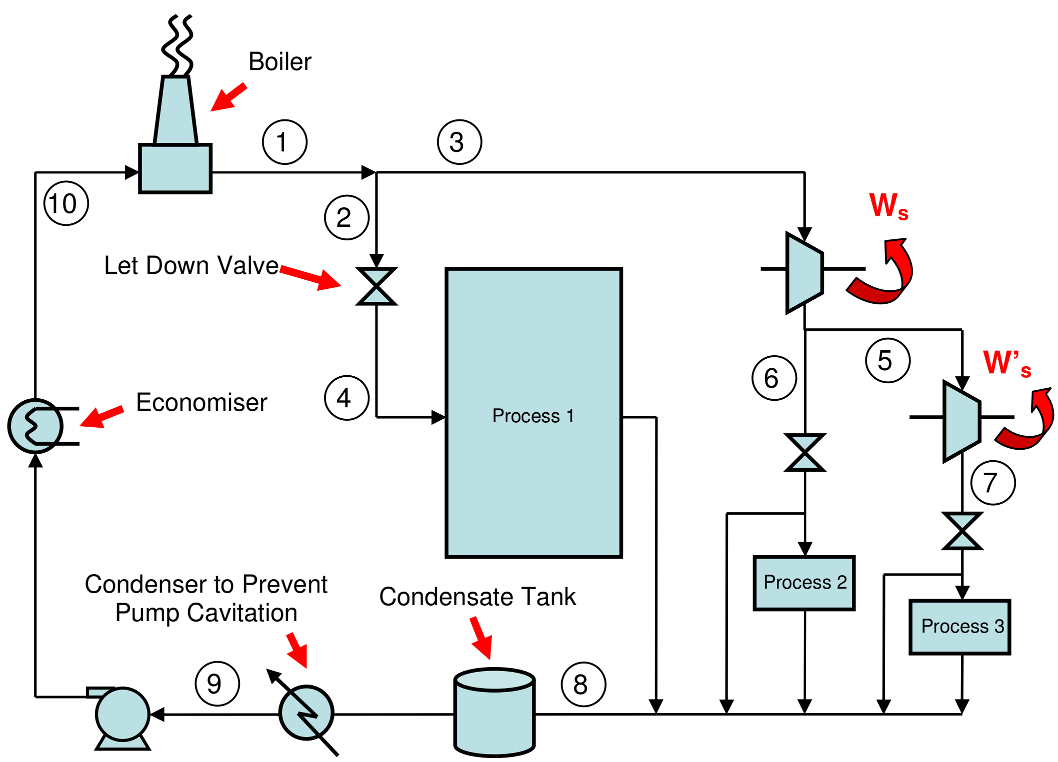

Figure 2.1: Steam system used as in a typical plant.

Table 2.1: Steam system data (Harrel, 1996).

Description	T (°C)	P (kPa)	T_{sat} (°C)
High Pressure (HP) from boiler	399	4 238	254
Medium Pressure (MP)	327	1 480	197
Process Pressure (an example)	225	2 550	225
Intermediate Pressure (IP)	209	377	141
Low Pressure (LP)	221	164	113
Deaerator Outlet	113	164	113
Feed Pump outlet to the boiler	116	6 310	277

Stream 1 is superheated high pressure steam, whereas the heating utility for processes is typically saturated steam. This is because heat exchanger networks are often arranged in parallel where each heat exchanger receives saturated steam and the latent energy is used to heat the appropriate process stream. Thus the let down valve provides a means to reduce the pressure of stream 2 such that it becomes saturated. Also, the latent energy of steam tends to decrease as temperature and pressure increase. Figure 2.2 (taken from steam tables) shows this tendency. As such, the temperature and pressure of the saturated steam used as a heating utility should, if at all possible, be as close to the required heating utility temperature so as to take as much advantage of the latent heat as possible. Thus the steam leaving the let down valve as stream 4 is at a lower pressure than stream 2. This is referred to as the process pressure from here onwards.

Stream 3 passes through the turbine where energy is recovered in the form of shaft work W_S, which is typically used to drive other process equipment. The exhaust from the high pressure turbine is generally at medium pressure. This steam then either passes through a medium pressure turbine as stream 5 where shaft work W'_S is recovered or passes through a let down valve and proceeds to a process as a heating utility (stream 6). The purpose of this let down valve is the same as that mentioned above. The exhaust of the medium pressure turbine is at low pressure and proceeds as a further heating utility to background processes as stream 7. A let down valve is indicated in stream 7, however, as seen in Table 2.1, low pressure steam is extremely close to saturation and as such this valve may not be necessary. The bypass streams around Process 2 and Process 3 have been included for completeness.

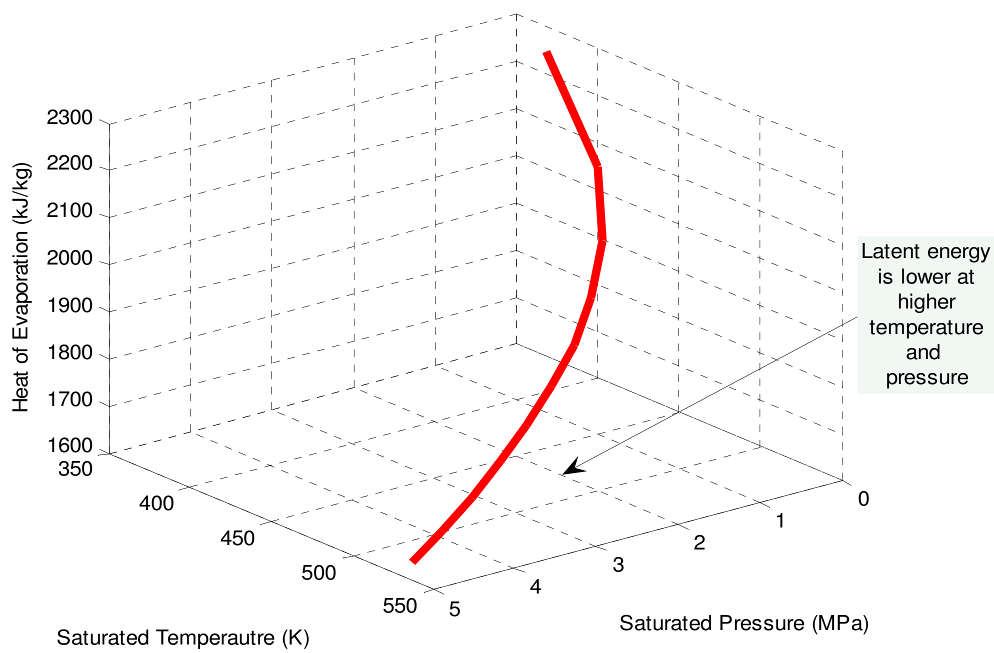

Figure 2.2: Tendency of latent heat to decrease as P and T increase.

The outlet from the hot utility using processes is typically saturated condensate. These streams then combine to form stream 8 and proceed to a condensate tank. The condensate then passes through a condenser to ensure the condensate is far away from the saturation temperature so as to prevent the risk of cavitation during pumping. The outlet from the condenser is stream 9. The risk of cavitation is mostly associated with centrifugal pumps which are the most prevalent in boiler systems. Make up water is also typically added in this region although this has been omitted for simplicity. After being pumped the return stream to the boiler passes through a preheater or economiser to heat the boiler feed, shown as stream 10.

By combining the processes that require a heating utility into a single process that utilises steam at multiple pressure levels the problem can be somewhat simplified. Figure 2.3 shows this modification. A HEN that utilises multiple steam pressure levels has a few more considerations than that of a single pressure level steam supply, but this will be expanded upon in the next section.

The streams 1 to 6 in Figure 2.3 are identical in description to those of Figure 2.1. Stream 7 however is the combined outlet of the HEN. Stream 8 has passed through the condensate tank, condenser and pump and is about to be preheated, while stream 9 is the preheated return to the boiler.

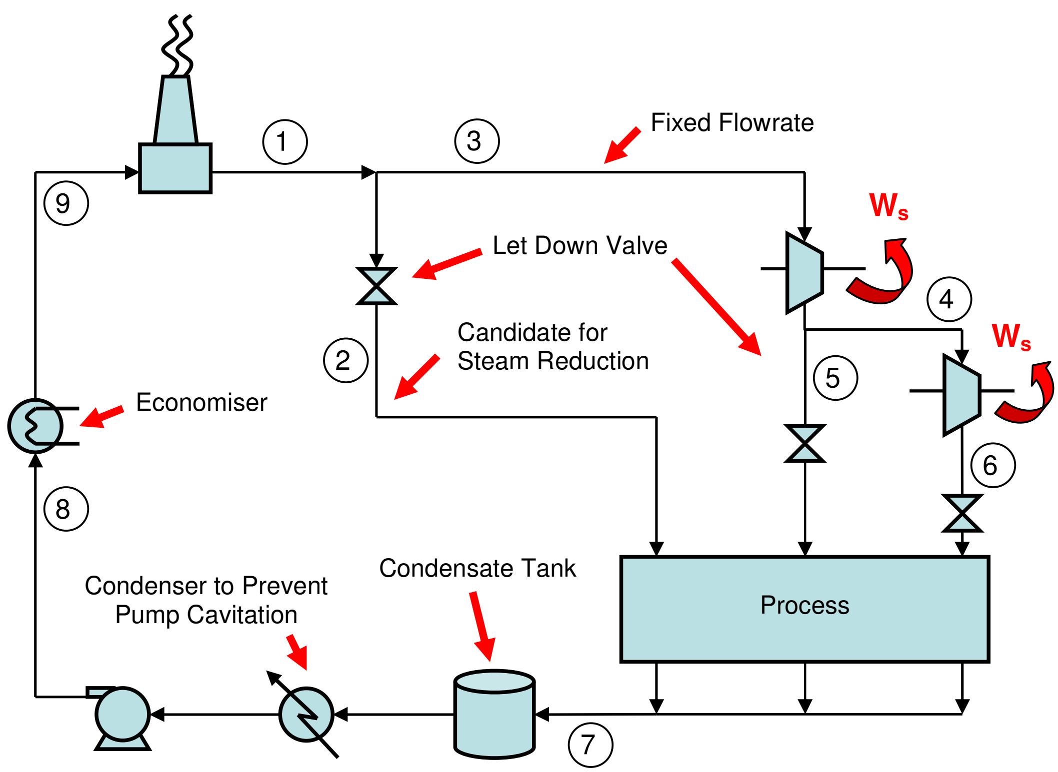

Figure 2.3: Steam system where HEN receives steam at multiple pressures.

2.2 Boiler Efficiency

Boiler efficiency in its simplest form is a ratio of the energy content of the steam produced by the boiler to the energy content of fuel used in the boiler, shown in Constraint (2.1). Since energy is lost in the process of transferring energy from the fuel to the steam, the energy of the fuel can often be expressed as the energy gained by the steam in addition to the energy lost by the boiler, shown in Constraint (2.2). Thus by substituting this relationship into the definition of the boiler efficiency a simple expression for the boiler efficiency can be created, shown in Constraint (2.3).

$$\eta_b = \frac{Q_{steam}}{Q_{fuel}} \tag{2.1}$$

$$Q_{fuel} = Q_{steam} + Q_{loss} \tag{2.2}$$

$$\eta_b = \frac{Q_{steam}}{Q_{steam} + Q_{loss}} \tag{2.3}$$

In the constraints above, η_b is the boiler efficiency, Q_{steam} is the energy gained by the steam, Q_{fuel} is the energy contained in the fuel and Q_{loss} represents heat losses in the boiler. The energy gained by the steam is shown in Constraint (2.4), where c_P is the specific heat capacity of the boiler feed water, ΔT_{sat} is the difference between the saturated temperature and the temperature of the boiler feed water, q is the sum of the latent and superheated sensible energy and F is the mass of steam raised by the boiler (Shang and Kokossis, 2004).

$$Q_{steam} = F\left(c_P \Delta T_{sat} + q\right) \tag{2.4}$$

The Q_{loss} term originates in the most part from two areas; the boiler surface losses and the flue gas losses. Pattison and Sharma (1980) performed a study on heat losses for British Gas. Using the data from their study they were able to form a correlation relating heat losses to the steam load percentage of the boiler, F/F^U and the amount of energy gained by the steam, Q_{steam}, using two regression parameters, *a* and *b*. Constraint (2.5) shows this relationship.

$$Q_{loss} = Q_{steam}\left(a\frac{1}{F/F^U}+b\right) \qquad (2.5)$$

Using Constraints (2.4) and (2.5) in the expression for boiler efficiency, Constraint (2.6) is formed. Simplifying this constraint gives Constraint (2.7) which accounts for the variation of boiler efficiency with changing load and capacity. Shang and Kokossis (2004) then define η_b as the ratio of the heat load of the steam to the heat of fuel, shown in Constraint (2.8).

$$\eta = \frac{F(c_P\Delta T_{sat}+q)}{F(c_P\Delta T_{sat}+q)+F(c_P\Delta T_{sat}+q)\left(a\frac{1}{F/F^U}+b\right)} \qquad (2.6)$$

$$\eta = \frac{F/F^U}{(1+b)(F/F^U)+a} \qquad (2.7)$$

$$\eta_b = \frac{q(F/F^U)}{(c_P\Delta T_{sat}+q)[(1+b)(F/F^U)+a]} \qquad (2.8)$$

Constraint (2.8), while not a strict definition of boiler efficiency, can be used as a comparative tool to indicate the effects of changes in the amount of steam raised by the boiler, *F*, and the subsequent changes in boiler feed temperature in the term ΔT_{sat}.

The effect of reducing the steam flowrate in the steam system will reduce *F* in Constraint (2.8). One means of maintaining the efficiency η_b is to increase the temperature of the stream returning to the boiler, which effectively reduces the value of ΔT_{sat}. Figure 2.4 shows the percentage increase in return temperature necessary to retain the boiler efficiency for a specified decrease in the return mass flowrate. Typical values of the other parameters were used to create Figure 2.4 and these are shown in Table 2.2.

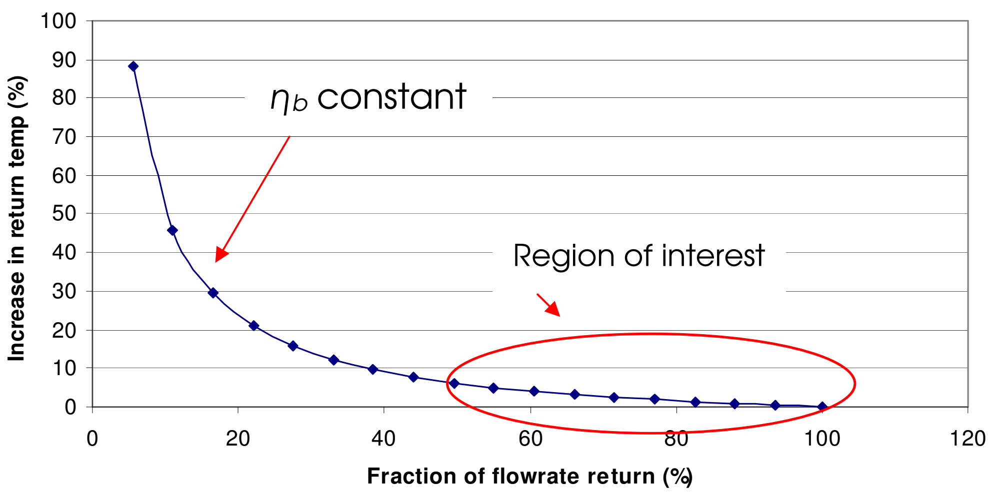

Figure 2.4: Sensitivity of Constraint (2.8) to changes in *F*.

Table 2.2: Steam system data (Harrel, 1996).

Parameter	
q (sum of the latent and superheated energy)	2110 (kJ/kg.K)
F^U (maximum steam load of boiler)	20.19 (kg/s)
c_P (specific heat capacity of boiler feed water)	4.3 (kJ/kg)
a (regression parameter)	0.0126
b (regression parameter)	0.2156
T_{sat} (saturated steam temperature at boiler pressure)	253.20 (°C)
T_{ret} (initial return temperature for 100% nb calculation)	116.10 (°C)
F (initial return flowrate for 100% nb calculation)	18.17 (kg/s)

In Figure 2.4 it can be seen that for a fairly substantial decrease in steam flowrate a relatively small increase in the return temperature is required to maintain the efficiency defined by Constraint (2.8). The ΔT_{sat} value is calculated by subtracting T_{ret} from T_{sat}.

2.3 Pressure Drop through HEN Elements

In HENs pressure can be lost through pipes, condensate heat exchangers and condensers. Pressure drop correlations are therefore needed in the form of constraints so as to incorporate this into the optimisation framework. Many correlations exist in literature, however those used by Kim and Smith (2003) are the most appropriate as they are readily integrated with flowrate which is already a variable in the current HEN model. The correlations are derived from the work of Nie (1998) where he represents pressure drop for conventional shell and tube heat exchangers. Thus it will be assumed that these heat exchangers are used in the investigation.

2.3.1 Heat exchanger pressure drop

The tube side pressure drop for a heat exchanger is much more easily determined than the shell side pressure drop, according to the rigorous derivations by Nie (1998). As such it will be assumed that the steam and condensate pass through the tube side of the heat exchanger for simplicity, as this will help integrate pressure drop into the HEN optimisation problem.

The pressure drop function presented by Kim and Smith (2003) is shown in Constraint (2.9).

$$\Delta P_t = N_{t1} V_t^{1.8} + N_{t2} V_t^{2} \tag{2.9}$$

This constraint was derived from the work of Nie (1998), where the tube side pressure drop is calculated with respect to the tube side fluid velocity. Kim and Smith (2003) then convert this velocity constraint to one with respect to volumetric flowrate, as seen in Constraint (2.9). According to Nie (1998) the two terms account for the pressure drop as a result of the friction loss in the tubes and the loss as a result of sudden contractions, expansions and flow reversals. The two factors N_{t1} and N_{t2} are shown in Constraints (2.10) and (2.11) below.

$$N_{t1} = \frac{1.115567}{\pi^{2.8}} \frac{\rho^{0.8} \mu^{0.8} n_{tp}^{2.8} A}{N_t^{2.8} d_o d_i^{4.8}} \tag{2.10}$$

$$N_{t2} = \frac{20}{\pi^2} \frac{n_{tp}^3 \rho}{N_t^2 d_i^4} \tag{2.11}$$

In Constraints (2.9), (2.10) and (2.11), ΔP_t is the tube side pressure drop, V_t is the tube side volumetric flowrate, ρ is the fluid density, μ is the fluid viscosity, n_{tp} is the number of tube passes, A is the heat transfer area, N_t is the number of tubes, d_o is the outside tube diameter and d_i is the inside tube diameter.

Many of the terms in these constraints such as the heat transfer area, the number of tubes and the tube dimensions are very much interrelated in the heat exchanger design. As such Kim and Smith (2003) utilise certain industrial design guidelines to demonstrate pressure drop in their case studies. These guidelines, as well as those used for this work are discussed in Appendix A.

The pressure drop for condensers must also be catered for as they also appear in the networks developed thus far. According to Sinnot (2005), the pressure drop through condensers where total condensation occurs can be approximated by calculating the pressure drop in the conventional fashion using the inlet vapour conditions and multiplying this by a factor. Two factors are put forward, the first by Kern (1950) who suggests a 50% factor and the second by Frank (1978) who suggests 40%. Since 50% will be the more conservative option this will be used throughout the model. Thus the condenser pressure drop will be approximated by Constraint (2.12).

$$\Delta P_{t,c} = 0.5\left(N_{t1}V_t^{1.8} + N_{t2}V_t^{2}\right) \tag{2.12}$$

In Constraint (2.12) N_{t1} and N_{t2} are equivalent to those for Constraint (2.9).

2.3.2 Piping pressure drop

Kim and Smith (2003) define the piping pressure drop according to Constraint (2.13). This is derived from commonly used pressure drop correlations, as well as a friction factor by Hewit, *et al.* (1994) to approximate the fanning friction factor.

$$\Delta P_P = N_P^{EX} V_P^{1.8} \tag{2.13}$$

This is remarkably similar to the first term of the tube side pressure drop correlation shown before, as tubes and pipes are very similar. In Constraint (2.13), N_P^{EX} is a factor to relate fluid properties and the pipe structure as shown in Constraint (2.14).

$$N_P^{EX} = \frac{1.11557}{\pi^{1.8}} \frac{\rho^{0.8}\mu^{0.2}L}{D_i^{4.8}} \tag{2.14}$$

In Constraint (2.14) L is the pipe length and D_i is the pipe inside diameter. Since the diameter is a design choice Kim and Smith (2003) use an economic trade-off of the optimal pipe size suggested by Peters and Timmerhaus (1991) where the optimal pipe diameter is given as a function of volumetric flowrate and fluid density. Using this relation Constraints (2.13) and (2.14) are rewritten as Constraints (2.15) and (2.16) respectively.

$$\Delta P_P = N_P^{NW} \frac{1}{V_P^{0.36}} \tag{2.15}$$

$$N_P^{NW} = \frac{188.318}{\pi^{1.8}} \rho^{0.176} \mu^{0.2} L \tag{2.16}$$

In Constraint (2.15) it can be seen that pressure drop is now an inverse function to volumetric flowrate which seems counterintuitive. The relation by Peters and Timmerhaus (1991) however ensures that every time a new velocity is chosen the optimal pipe diameter is used, giving the inverse relation. Now the pressure drop through the pipes is only a function of the pipe length, the fluid properties and the volumetric flowrate.

2.3.3 Pressure drop with respect to mass flowrate

The heat exchanger and piping pressure drops shown in Constraints (2.9), (2.10), (2.11), (2.12), (2.15) and (2.16) can be changed so that they can accommodate the fluid mass flowrate as used by the flowrate reduction models developed by Coetzee and Majozi (2008) and implemented in the next section. Constraints (2.17), (2.18), (2.19) and (2.20) show these adjustments for the tube side heat exchanger correlations.

$$\Delta P_t = N_{t1}^* \dot{m}_t^{1.8} + N_{t2}^* \dot{m}_t^2 \tag{2.17}$$

$$N_{t1}^{*}=\frac{1.115567}{\pi^{2.8}}\frac{\rho^{-1}\mu^{0.8}n_{tp}^{2.8}A}{N_{t}^{2.8}d_{o}d_{i}^{4.8}} \tag{2.18}$$

$$N_{t2}^{*}=\frac{20}{\pi^{2}}\frac{n_{tp}^{3}\rho^{-1}}{N_{t}^{2}d_{i}^{4}} \tag{2.19}$$

$$\Delta P_{t,c}=0.5\left(N_{t1}^{*}\dot{m}_{t}^{1.8}+N_{t2}^{*}\dot{m}_{t}^{2}\right) \tag{2.20}$$

Constraints (2.21) and (2.22) show the adjustment for the piping correlations.

$$\Delta P_{P}=N_{P}^{NW*}\frac{1}{\dot{m}_{P}^{0.36}} \tag{2.21}$$

$$N_{P}^{EX*}=\frac{188.318}{\pi^{1.8}}\rho^{0.536}\mu^{0.2}L \tag{2.22}$$

These correlations will be used for pressure drop through equipment throughout this dissertation

2.4 References

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3. Literature Survey

This literature survey gives a background to the work done in the field of heat exchanger network optimisation and design that the work in this dissertation draws upon. There are four sections, the first of which gives a brief background to the field of heat integration and the role of the heat exchanger network. Next several graphical targeting techniques are discussed and their relevance put into perspective. Thereafter follows the network design procedure for process to process heat integration.

Utility system optimisation is the major focus of this literature survey as it constitutes the area of interest for this dissertation. Various utilities such as wastewater, cooling water and steam systems are investigated. The final section concerns pressure drop through heat exchanger networks which is the final issue addressed in the dissertation. Following this is a brief conclusion based on the findings of this literature survey.

3.1 Process Integration

Heat integration is an integral part of process design. Figure 3.1 shows a representation of the various stages of process design and where heat integration occurs within the design framework. At the heart of a process is a reactor that converts raw materials to products. These products must then be separated from any remaining raw materials and any by-products that are formed. Consequently, separation constitutes the second level of process design. The separation and sometimes the reaction stages require the addition and removal of energy. This can often be done within the process by linking streams that require heating to streams that require cooling. This interaction, or heat integration, has been investigated in great detail through Pinch

Analysis. The left over process streams are then heated or cooled using external utilities (Seider *et al.* 2004).

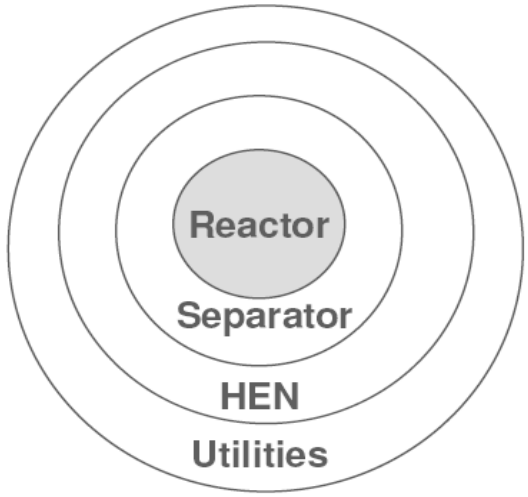

Figure 3.1: Onion diagram depicting the various aspects of plant operation.

The majority of process heating or cooling occurs in heat exchangers (the exceptions may use some form of heat exchange with the environment). Since many streams can be used for heat integration, a systematic means of designing a HEN is necessary. The design and optimisation of process to process heat exchanger networks is a very widely researched topic and some of the major advances will be discussed in the following section. One of the objectives in designing these networks is to reduce the amount of utilities required for the process. The minimum amount of this utility in terms of thermal duty has been found by many techniques, but the actual utility systems have not been explored in as much detail as the heat integration networks. The work done to improve the actual utility systems will therefore be discussed last as this leads on to the work presented in this dissertation.

3.2 Heat Exchanger Network Synthesis

The primary objective of heat integration in plant design is to minimise the external utilities. When the optimisation of HENs started in the 1960's however, utilities were fairly abundant and so networks were designed so as to minimise capital cost, most generally heat exchange area. The earliest forms of optimisation occurred as mathematical models of various types. At this early stage it was found that the HEN problem was extremely complex and as such required simplifications to make the models useable. The earliest models were of the form of assignment problems and tree searches, which led to the branch and bound search methods (Gundersen and Naess, 1987).

The Organisation of Arab Petroleum Exporting Countries (OAPEC) oil embargoes of the 1970's shocked the western world into an energy crisis. Utilities, specifically heating utilities, drastically increased in price and this led to a huge push in the research of utility minimisation. A highly successful optimisation technique was developed that incorporated utility targeting using graphs. This is known today as Pinch Analysis (Gundersen and Naess, 1987). Mathematical programming remained a widely used technique and will be discussed in the next section.

3.2.1 Graphical targeting and pinch

One of the earliest uses of graphical targeting was presented by Hohmann (1971). Although his work was a trade off between utility and area minimisation, this graphical method led to the now famous T-Q diagram approach. Since the work of Hohmann (1971) was graphical in nature it was largely rejected by the HEN synthesis community at first. Hohmann (1971) used a feasible network solution space to show how the minimum approach temperature (ΔT_{min}) affected the HEN design. From this diagram the energy area trade off could clearly be seen. He also proposed the N–1 rule which gives a close target for the minimum number of heat exchange units.

Huang and Elshout (1976) were the first to combine the individual streams that could be represented on a T-Q diagram into composite curves. This allowed all the hot process streams to be viewed as a single curve on the T-Q diagram. The same applied to the cold process streams. Very briefly, the composite curves are constructed by adding all the duties of hot or cold streams in a particular temperature interval. Thus a staggered curve can be made for both the hot and cold process streams.

Umeda, Handa and Shiroko (1979) use a computer aided method based on thermodynamic energy availability to perform heat integration on a process. The heat availability diagram discussed by Umeda (1978) is used as a basis for the available energy concept. Essentially any stream at a particular temperature and pressure can be transferred to another, lower temperature and pressure by releasing energy. Thus by knowing the minimum temperature and pressure of a stream the maximum amount of available energy for that stream can be calculated. The process is divided into heat source and heat sink streams and from these heat source and sink lines are constructed. Since these lines are calculated from the available energy derived from a change in temperature and pressure, the lines themselves are not straight. These are essentially composite curves. The authors then plot the heat source and heat sink curves on a T-Q diagram and attempt to minimise the heat loss, which is represented by the space between the lines on the T-Q diagram, using mathematical programming. Figure 3.2 shows these early composite curves. Q_{CG}, Q_{RC} and Q_{HG} represent the thermal energy in the cooling, heat recovery and heating regions of the T-Q diagram respectively. A_{LHT} represents the energy loss for the regions. By shifting the curves together this is minimised. The point of minimum loss is called the pinch point. They then attempt to eliminate the pinch point by altering the process streams in an attempt to further increase heat recovery, or in their case minimise heat loss. Two computer programmes were created by the authors, the first to recover heat in the method described above, and the second to design a network in which the heat exchange area is minimised. Little detail as to the working of the second model is given.

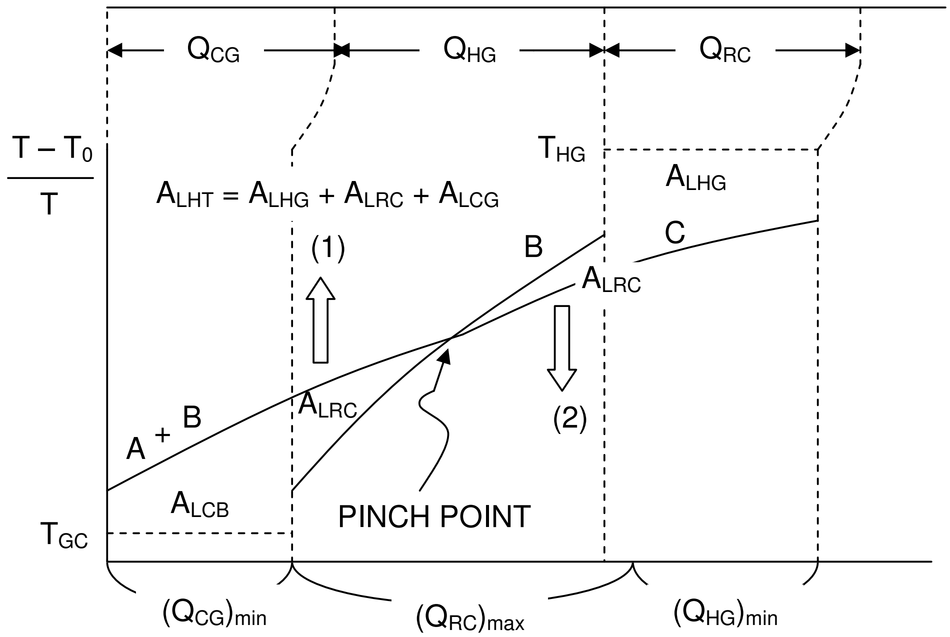

Figure 3.2: Heat availability diagram (Umeda, Handa and Shiroko, 1979).

This early work shows many aspects used in both the mathematical programming approach to the HEN design problem as well as the purely thermodynamic methods. The T-Q diagram and heat source and heat sink lines are essentially composite curves and the Pinch Point concept is the cornerstone of the successful work by Linnhoff and his co-workers. The stream energy balance concept used to derive the first mathematical model is used in many other later models and the concept of minimising area as an objective is also widely adopted.

Linnhoff, Mason and Wardle (1979) investigate several HEN design techniques developed in literature that were tested on fairly simple test problems. They found that although these techniques worked for the problems they were not readily applied in industry. They showed some of the reasons for this by examining several subject areas.

They described how the heat recovery problem is extremely dependent on the minimum approach temperature. They alluded to the concept of the pinch point and how the maximum amount of energy recovery can be achieved for a specific approach temperature by finding the pinch point. Using a T-Q diagram they showed how, if the minimum utilities are not met, any deviation affects both the heating and cooling side. They explained the concept of minimum units and how they can be predicted. They further showed how the heat load is affected by the topology of the network for the case of the minimum number of heat exchange units. They concluded by discussing how in extreme cases changing aspects of the HEN parameters may lead to better results. This implies changing parameters in the actual process streams.

This work was essentially aimed at promoting HEN design techniques for industrial scale problems. Several of the aspects alluded to were used by Linnhoff and Hindmarsh (1982) in their comprehensive HEN design approach however other aspects such as the process stream parameter adjustment were ignored. Some areas of importance with respect to utility minimisation found in this paper are the pinch point as well as the concept that the network topology does play a role in heat recovery in certain cases, such as for minimum units.

Linnhoff and Hindmarsh (1982) noted that the network temperature pinch created a bottleneck to feasible heat recovery in HEN design. Up until their work, HEN design was based around techniques to stimulate designers to create better networks. This was done by observing various network performance targets, such as minimum utilities, minimum area or even minimum heat exchange units or shells. Any of these objectives could be used as the basis to guide the designer but no reliable HEN design method existed.

Minimum utilities were explored by Hohmann (1971) and also by Linnhoff and Flower (1978). They varied the minimum approach temperature for the HEN to find the minimum utilities. From heat exchanger design it was known that heat exchange area was linked to heat recovery for a single unit, however a link between heat recovery in the HEN and heat exchange area was also noted for various designs. Grimes (1980) also found that the objective of minimum utilities along with minimum heat exchanger units was mutually incompatible for systems featuring a network pinch.

Bearing this in mind, Linnhoff and Hindmarsh (1982) then tried to combine many of the above areas into a more holistic HEN design technique. They firstly designed a network for minimum utilities and then began reducing the number of units while the minimum utilities were relaxed. They realised that the minimum cost for the network would occur when the correct combination of minimum utilities and units was found. By varying the minimum approach temperature both the minimum utilities as well as the minimum number of units were changed. By constructing a graph of ΔT_{min} vs. cost, the best approach temperature could be found. This method provided designers with a reasonable trade off situation with which to choose the ΔT_{min}.

The network was designed using the Problem Table Algorithm developed by Linnhoff and Flower (1978). This required constant heat capacity values for the streams as well as a specified ΔT_{min}. The algorithm was then used to find the pinch point of the network as well as the minimum utilities. With the minimum utilities and pinch point the network could then be designed. The pinch technique also gave designers an indication of when streams needed to be split. This removed a large area of design uncertainty. The authors also discussed the significance of threshold problems, where no pinch exists. These problems are characterised by a single hot or cold stream with a very large duty, leading to the need for only a single utility, either hot or cold. By adjusting the ΔT_{min} the utility duty can be altered up until the point where both heating and cooling are required. This point is known as the ΔT_{thresh} representing the threshold approach temperature. This point is important since larger approach temperatures are favourable for smaller heat transfer areas (Sinnot, 2005).

The result of the Problem Table Algorithm can be used to construct the Grand Composite Curve (GCC) as was done by Linnhoff and Flower (1978). This is also a representation of the system however the hot and cold streams are combined as in the Problem Table Algorithm. Figure 3.3 shows an example of a GCC.

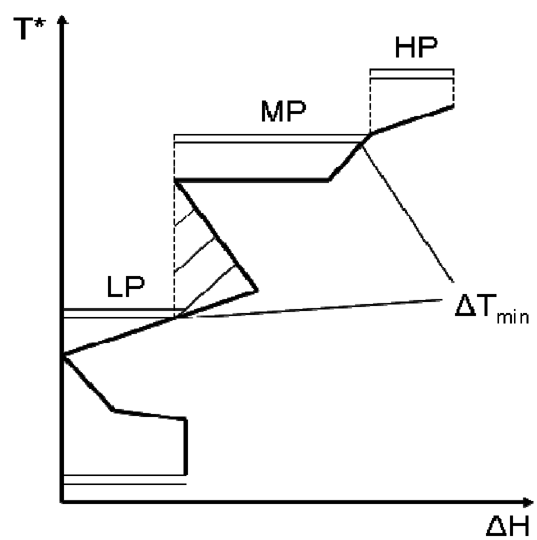

Figure 3.3: GCC showing multiple utility levels.

If a comparison is made between the GCC and regular composite curves, one can see how the process to process heat exchange all happens at the middle temperatures in the pinch analysis of the composite curves, but are spread out in the GCC. As such, the utilities in the regular composite curve method are at the extremes of the temperature range while the utilities for the GCC can be spread out, as can be seen in Figure 3.3. This is essential since high level energy sources such as high pressure steam are much more expensive to produce than low pressure steam. The GCC shows where lower level heat sources can be used to satisfy the energy demands of the system. The same concept applies for the cooling section.

The GCC gives designers a lot of insight into the particular heating system in question. It is a very successful design tool and research is continuing in this area. For example Salama (2009) presented a new procedure, known as the enthalpy flowrate technique. This leads to the formation of a new curve, the compliment GCC. This curve is used as a tool for the presentation of the temperature differential distribution between the composite curves, the estimation of heat exchanger area as well as facilitating the estimation of heat exchanger area in multiple utility targeting.

A grid diagram representation first used by Linnhoff and Flower (1978) was used to specify which streams exchanged heat with one another. The problem was divided into a hot and cold side, separated by the pinch and the design was done away from the pinch at either end. This method successfully embraced the temperature restriction at the pinch, as units close to the pinch were forced to obey the minimum approach temperature whereas those further away had more flexibility. This led to the concept of a crucial match, meaning a hot and cold stream match that was essential for the minimum heating or cooling duty to be realised. Once the minimum utilities were found the network was then adjusted to try to reduce the number of units. Other aspects such as safety and controllability were also considered here as extra constraints. Generally a suitable objective was to minimise cost, which often led to slightly higher than minimum utilities but greatly reduced units and hence capital cost.

In short, Linnhoff and Hindmarsh (1982) had combined many aspects of HEN design that had already been tested and structured the various methods in a useful and holistic manner which would guide the designer to an extremely cost effective and workable design. Their use of the Pinch technique showed their preference to the thermodynamic method, however, collaboration with mathematical models may have been able to improve the final parts of the design stage, such as minimising cost in a more structured method. The advantage of the Linnhoff and Hindmarsh (1982) method is that the designer remains intricately involved in the design and can make appropriate changes at any point. The Pinch design technique remains a widely used and effective technique to date.

External utility targeting could find the thermodynamic minimum utility for a specific global approach temperature. The actual network construction remained a less precise science. Many heuristic techniques made many of the decisions easier, but the simple size and complexity of the problem seemed to lend itself to mathematical programming. Linnhoff *et al.* (1982) did create a very usable and all encompassing design strategy, however many aspects could simply not be solved by graphical methods. The following section examines some network design techniques and even utility targeting using mathematical programming.

3.2.2 Network design and mathematical programming

Papoulias and Grossmann (1983a, 1983b and 1983c) set about creating a large scale chemical plant design and optimisation procedure using mathematical programming in the form of three separate but interlinking MILP models. At the time of their work there was little faith in optimisation as a design tool as it was not widely understood. However the alternative design methods that consisted of thermodynamic and heuristic techniques were not very systematic and it was difficult to create holistic design methods for even simple plants due to the large number of components necessary for their operation.

The advantage of using MILP models for plant design is that they can simultaneously optimise structures and continuous variables. Thus with a superstructure that encompasses all of the design options an optimal solution can be found. The superstructure is created using prior engineering knowledge and judgement of the system. These superstructures do create large optimisation problems however the solution space can be reduced with heuristic techniques and thermodynamic methods.

In the first of a set of three papers Papoulias and Grossmann (1983a) designed an optimal utility system. In the second and third papers they discussed heat integration networks and total chemical plant design respectively however these will be examined later. Most representative models contain a wide variety of constraint types such as linear, nonlinear and even discrete to name a few. Each of these creates different challenges in solving the models however the most problematic are nonlinear nonconvex constraints. These prevent globally optimal solutions from being guaranteed and therefore the authors sought a means to remove or linearise this aspect of their model. They realised that if the state variables of a system, such as the operating temperatures and pressures, have fixed values then linear constraints that describe the operation of units, such as mass and energy balances, can be derived. Therefore they define many different operating conditions and use binary variables to ensure that only one of these conditions are analysed at a time. The objective function minimises cost for the system and this is often a concave function production rate. By creating a linear under-estimator for the cost function the entire model can be linearised. If a single under-estimator is deemed too inaccurate more of these can be added to create a piecewise linear objective function.

There have been several attempts to optimise utility systems prior to the work by Papoulias and Grossmann (1983a). Nishio and Johnson (1979) used a model for the optimisation of steam and power plants. They did however have to use heuristic rules to make several plant configuration decisions which are believed to have overlooked many feasible alternatives. Nishio *et al.* (1980) then used thermodynamic techniques in an attempt to minimise energy losses in plant units. Grossmann and Santibanez (1980) created a MILP to synthesise steam generation systems. This model was able to specify various operating conditions and several aspects such as turbine assignments were excluded.

Papoulias and Grossmann (1983a) set about creating an all encompassing MILP which could overcome the problems up to that point. They created a superstructure which included all the process units typically found in utility systems, such as boilers, turbines, motors, steam headers, condensers, etc. They included three pressure levels in their analysis. Figure 3.4 shows the superstructure used to derive the MILP. This consisted of continuous variables, binary variables to specify the existence of a unit as well as binary variables to specify the operating condition. The creation of the various conditions and the specification of which units can and cannot operate with others requires intimate process knowledge and insight, as well as a large amount of user interaction with the model. The authors then tested the MILP on an example problem and did find significant savings.

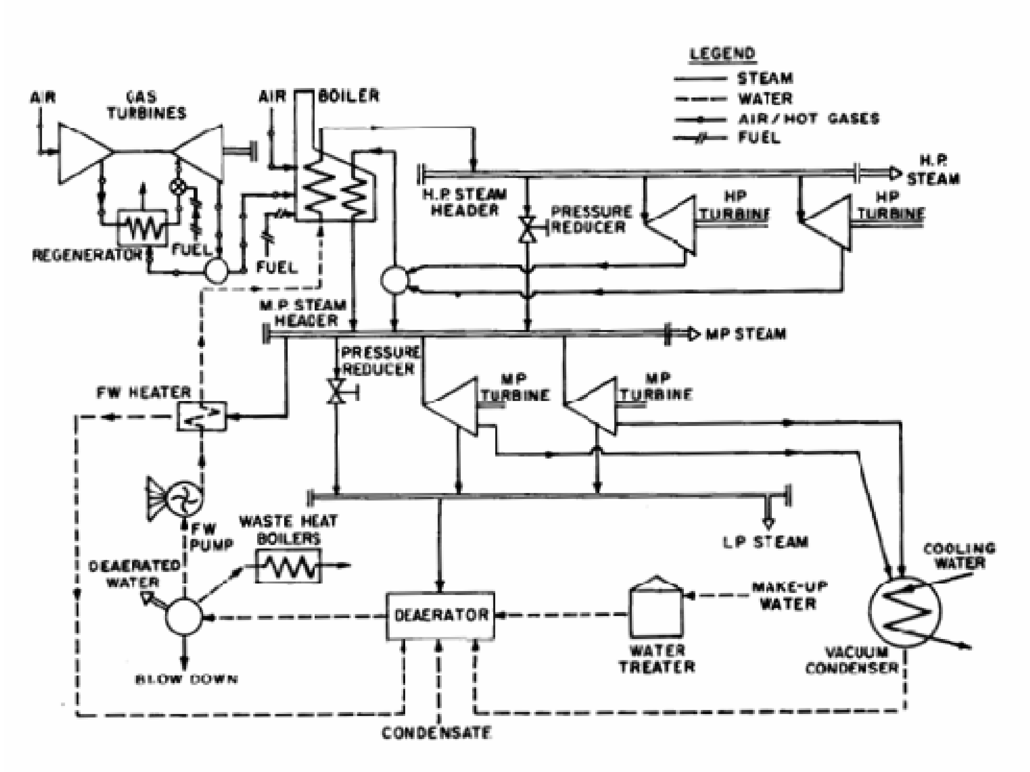

Figure 3.4: Utility system superstructure (Papoulias and Grossmann 1983a).

This work shows how flexible mathematical programming is in process design. The ability of an MILP to perform simultaneous variable and structural optimisation is of great significance. The superstructure includes all the possible solutions and the model optimises both the configuration and the continuous variables. The authors created sets of operating conditions that could be discretely chosen so as to create a linear model for which numerous solution techniques were available. Global optimality is also guaranteed. They did not renounce other methods such as thermodynamic techniques or heuristic rules as these are essential for creating the superstructure as well as reducing the solution space of the model.

The overall problem definition of work that encompasses the minimisation of utilities by maximising process to process heat exchange is, with given limiting temperatures and flowrates, to minimise the total annualised cost of the HEN and its associated utilities. Three objectives were commonly defined by many authors in this particular field, such as Linnhoff and Flower (1978) and Cerda and Westerberg (1981). The first objective is to minimise external utilities or their costs for a fixed approach temperature. This should be done prior to the actual network design so as to give the designer an insight as to the ideal amount of heat integration. Secondly the number of heat transfer units should be minimised. Since many networks could achieve the minimum amount of utilities, another objective should choose between them and the heat transfer units, or shells, often represents a large part of the cost. Thirdly, once the pinch point for the particular system was found it could be eliminated by changing the process. In this way the amount of possible heat integration could be increased.

Papoulias and Grossmann (1983b) then introduced the second part of their plant design MILP where they attempted to minimise the external utilities required by a plant. More specifically, since utilities such as steam occur at various levels and have different costs, the heat integration was done so as to minimise the external utility cost. Other integration techniques often fail to integrate aspects such as stream splitting and forbidden matches into their designs. They approached the HEN synthesis problem as a transhipment framework, as often dealt with in the field of operations research. In this case heat is considered the resource that is to be distributed throughout the network by means of temperature intervals. Several methods in literature exist to divide the system of hot and cold streams into temperature intervals, such as the technique of Linnhoff and Flower (1978). Any of these methods can be used for the transhipment model although the Papoulias and

Grossmann (1983b) recommend the methods of Grimes (1980) and Cerda (1980) as they lead to fewer intervals and thus lower solution times. The authors develop an LP model that deals with the minimisation of utilities for systems with no restricted matches. This model is then expanded to include restricted matches by allocating weighting cost factors to the restricted stream matches, although this results in a larger problem than the original LP model. An MILP model is then developed to minimise the number of process units for the network. Once again weighting factors are used to implement restricted matches. The final aspect of eliminating pinch points is dealt with in the final paper in the series.

Many successful heat integration techniques originated at about the same time. The method of Papoulias and Grossmann (1983b) is arguably the most comprehensive mathematical programming method as it deals with aspects such restricted matches and minimising heat transfer units. Most importantly the MILP nature of their work allows it to collaborate with another model relating to the actual plant structure. This interaction creates an opportunity to actually alter the process based on the pinch points and improve heat integration.

In the third instalment of their chemical plant design model, Papoulias and Grossmann (1983c) derived an MILP model to help design a chemical plant. Some attempts at creating similar models were found by Papoulias and Grossmann (1983c) in literature and these essentially follow two approaches. The first method uses no initial structure but instead uses artificial intelligence techniques, such as the general problem solver principle of Siirola *et al.* (1971). These were based heavily on heuristic rules and as such may overlook or exclude very feasible designs. The second uses a superstructure to derive the model and an optimisation programme to find the particular objective. These optimisation approaches do however give very large problems due to the complexity of plant structures and as such are often difficult to solve.

Papoulias and Grossmann (1983c) had already constructed an MILP to deal with heat integration. As such they could construct their overall plant optimisation superstructure primarily out of the reactors and separation equipment. This resulted in a smaller model. The heat integration MILP would then receive variable inputs from the chemical plant MILP. As before many operating conditions for the plants are established and optimised using the MILP. Heuristic rules also help to reduce the size of the model by applying appropriate constraints. The MILP thus optimises continuous variables, structural binary variables and operating condition binary variables. This approach is identical to the utility system optimisation of Papoulias and Grossmann (1983a).

With the three MILP models formed the authors allow them to interact. The inputs to the heat integration MILP come from the chemical plant MILP. In this way the process can actually be changed if it will reduce the overall cost. The authors would then implement the utility system MILP to reduce the utility cost. All three models could be used simultaneously, however similar results were found by performing the utility optimisation after the structure was formed since the heat integration step attempts to reduce utility costs.

The combination of pinch technology as well as mathematical modelling has made the area of HEN design very effective. The actual utility systems are however complex and it was found that improvements could be made in these areas using techniques developed from the HEN design area.

3.3 Utilities

Pinch technology can be extended from heat pinch and used in what has become known as mass pinch. This has been successfully applied to many industrial cases and is a standard teaching agent in process design (Zhelev, 2006). This is made possible by the analogy that exists between the driving forces in heat and mass transfer. In heat transfer the driving force is a difference in temperature and in mass transfer it is a difference in concentration. This consequently allows principles involved in either optimisation to be used in the other.

3.3.1 Water utility systems

Minimisation of wastewater is arguably the largest aspect of mass integration. The concentration driving force was investigated by Wang and Smith (1994). Savelski and Bagajewicz (2000) proved mathematically that a certain condition for optimality exists when outlet streams are at their maximum concentration in wastewater minimisation. This is intuitively accurate since a higher outlet concentration would imply a lower water usage, providing the process stream remained the same.

Kuo and Smith (1998) examined the interactions between the design of water using operations and effluent treatment in an attempt to create a more holistic approach to designing water using systems in plants. They explored the option of reusing water streams that contained only small amounts of effluent, and in such a way reduce the amount of water used by the system. They used a similar superstructure for their optimisation as that developed by Takama *et al.* (1980).

The authors utilised the mass pinch concept, where streams and their concentrations are represented on composite curves, to find the minimum water flowrate for the system. This technique was used as a starting point for the design procedure, as once the target flowrate was known a network had to be created. Besides minimising flowrate several other objectives had to be considered, such as practicality, controllability, safety and cost. The pinch procedure resulted in an effluent stream with higher concentration, as this is how water was reduced.

As aforementioned a parallel can be drawn between mass and heat systems. The effluent removed in these water treatment systems is very similar to heat being added or removed by heating or cooling systems. Both systems require a driving force, be it concentration or temperature difference to cause mass or heat to be transported. The pinch procedure for both is also very similar. As such the concept was taken and used on cooling systems by Kim and Smith (2001).

3.3.2 Cooling water utility systems

Cooling water is a very effective medium to remove heat from process streams. Cooling water systems essentially consist of cooling towers that remove the heat gained by the cooling water from the process as well as the process HEN that needs to be cooled. The reduction of cooling water can reduce the capital and utility cost of a plant and as such has been investigated.

Kim and Smith (2001) extended the water reuse concept used by Kuo and Smith (1998) to cooling system design. Traditional cooling system HENs consisted of heat exchangers connected in series and as such each heat exchanger received cooling water directly from a cooling tower. By increasing the concentration of heat in the return stream to the cooling tower, as Kuo and Smith (1998) had done with the mass concentration, they were able to reduce the amount of cooling water needed for the system. This was achieved by rearranging the HEN into a series/parallel configuration and reusing the cooling water between units.

Kim and Smith (2001) set about developing a systematic means of reducing the cooling water flowrate for cooling systems. They used the heat pinch concept, much the same as Kuo and Smith (1998) had used the mass pinch concept, to find the minimum cooling water flowrate for a set of heat exchangers. The heat exchangers were represented on a T-Q diagram as composite curves. The cooling water supply line is then represented by a straight line on the diagram. By bringing the composite curve and the supply line together a pinch point is found and the flowrate associated with the supply line at that point represents the minimum flowrate for that particular HEN. Figure 3.5 shows the composite curves and the supply curve for an example problem shown by Kim and Smith (2001) where the pinch point can clearly be seen.

Furthermore, Kim and Smith (2001) went on to consider the effects of reducing the amount of cooling water in cooling systems. They did this by viewing all aspects of the cooling system in a holistic manner. The two main areas that a higher return temperature were expected to affect were fouling in the heat exchangers and pipes and the effect on the cooling tower. Increased fouling could be dealt with by adding additional chemicals or increasing the rate of cleaning. A cooling tower model was developed by Bernier (1994) and used to study the effects of simultaneously increasing the return temperature to the cooling tower as well as decreasing the flowrate. It was found that the cooling tower performance was in fact increased by these factors and as such it was deemed safe to reduce the cooling water flowrate.

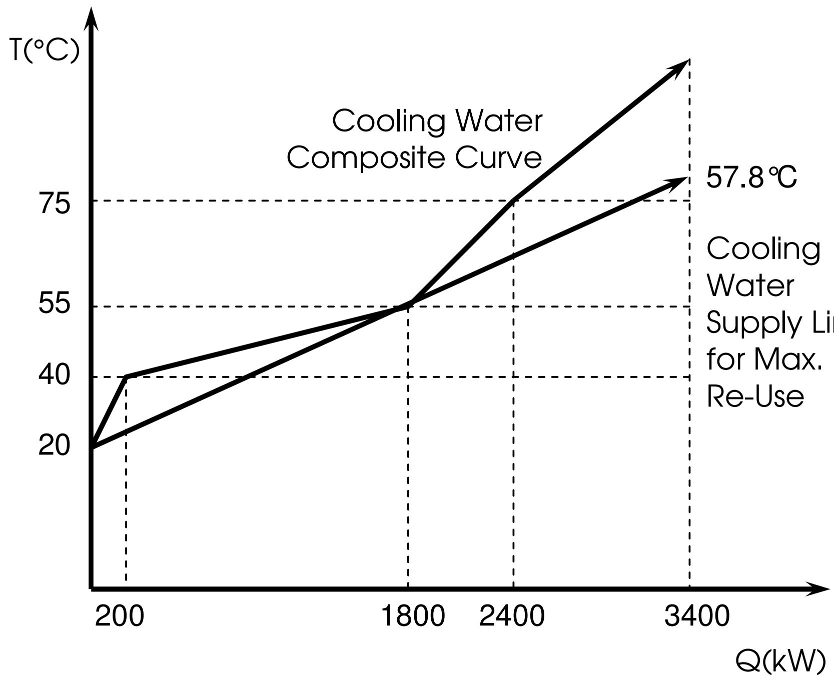

Figure 3.5: System curve and cooling water supply curve.

With the minimum flowrate known, the network can then be designed or rearranged to achieve it. The mains procedure used to design networks by Wang and Smith (1994) is used to design the network. This gives the designer a hands on approach and a deeper understanding of the network. The designer can then also make adjustments based on forbidden matches or control and safety aspects to name but a few. This approach is insightful but as with most heuristic design techniques it can often take a large amount of time.

Majozi and Moodley (2007) expanded the holistic approach to cooling water system design of Kim and Smith (2001) to include multiple cooling towers, as often found in process plants. This extra dimension adds complexity and creates the need to deviate from the procedure followed by Kim and Smith (2001). The graphical targeting technique is more difficult to implement as a result of the extra cooling tower. Thus a mathematical model is used for both the targeting and network design simultaneously. The mathematical modelling also gives the designer the ability to incorporate additional constraints into the design problem.

A rigorous superstructure was developed by the authors to formulate the necessary constraints to represent the system. The superstructure is not only a model design aid as the constraints can clearly be derived from it, but also an aid for presenting and explaining, and even expanding the model. With the superstructure, four cases relating to the return conditions and the network layout were explored:

In case 1 there was no limitation for both return temperature and topological matches. This is seen to be slightly unrealistic as high return temperatures increase fouling, and topological problems are likely to exist in large plants. In case 2 there were no temperature limitations but several dedicated source and sink cooling towers, creating a realistic topological situation. In case 3 there were return temperature limitations but no restriction on source and sink cooling towers. Finally, case 4 created the most realistic situation with a limited return temperature as well as several dedicated sources and sink cooling tower matches.

The resulting models contained many nonlinear constraints brought about by bilinear terms. The authors used a condition of optimality first described and proved by Savelski and Bagajewicz (2000) to fix the heat exchanger outlet temperatures to their limits. This linearised the mass and energy balance constraints. Several other constraints based around the cooling towers remained nonlinear and so a solution technique taken from Quesada and Grossmann (1999) was used to solve the models. This technique involved the creation convex envelopes developed by McCormick (1976) where a linearised solution can be found. This solution is then used as a starting point for the nonlinear model. Global optimality is only guaranteed if the relaxed solution matches the exact, nonlinear solution.

The work of Majozi and Moodley (2007) creates a very realistic design procedure where mathematical modelling is used to both target for a minimum flowrate as well as design a network. They conclude that for proper integration to occur, geometric and process feasibility must exist.

3.3.3 Steam utility systems

Steam is a very popular heating medium in chemical plants. Steam systems essentially consist of a steam boiler that generates steam at various pressure levels and thermodynamic conditions such as superheated or saturated. Steam is used as an energy source in operations that require heat. A reduction in steam flowrate can debottleneck the boiler in retrofit cases and save on capital costs in grassroot design as a smaller boiler can be installed. The cost of makeup water is also reduced with lower steam flowrates.

Shang and Kokossis (2004) used mathematical programming to attempt to account for the interaction of the site and the process in grassroot design. Their main aim was to optimise steam levels in the process while satisfying the entire hot utility demand. To do this they set about a fairly holistic approach as they considered the performance of both the boiler, with their boiler hardware model (BHM) and turbines, with their turbine hardware model (THM). They discovered how the utility levels influence the cogeneration potential, that is electricity from turbines and heat from steam, and optimise these levels in the context of cost.

The BHM is developed from a thermodynamic point of view by considering all aspects of the boiler. A correlation for heat loss and boiler capacity developed by Pattison and Sharma (1980) in a study for British Gas led the authors to a simple correlation for boiler efficiency and capacity and boiler return stream temperature. The THM is taken from work by Mavromatic and Kokossis (1998). This model was found to successfully predict efficiency trends based on load and operating conditions.

The authors successfully integrated the HEN with steam level optimisation in this rather holistic approach. They created an MILP model which guarantees optimality. This work is quite orientated to grassroot design, as it attempts to lead a design towards the most effective steam pressure levels. Other areas of the steam system are considered but not altered in any way to create a better overall system.

Coetzee and Majozi (2008) attempted to reduce the steam flowrate to process plant heating systems. They drew on the work by Kim and Smith (2001) and Majozi and Moodley (2007) to create a system for reducing the steam flowrate to a HEN using both a graphical targeting technique and mathematical programming.

The authors used the same premise as Kim and Smith (2001) to rearrange a parallel HEN into a series/parallel design to reuse some of the hot steam condensate. Traditional steam systems utilise heat exchangers connected directly to the boiler. Steam is then condensed in the heat exchangers, transferring latent heat to the process stream. The saturated steam condensate then proceeds to the rest of the steam system where cooling takes place to prevent cavitation. The steam is then pumped to high pressure and ultimately returned to the boiler. Coetzee and Majozi (2008) realised that there was still a significant amount of energy in the saturated condensate. This condensate could be recycled and reused in the same way as Kim and Smith (2001) and Majozi and Moodley (2007) had done with cooling systems. This procedure was expected to reduce the required stream flowrate.

The authors first attempted to find the minimum steam flowrate in a graphical manner using a T-Q diagram. A similar method to that of Kim and Smith (2001) was used however the additional aspect of heat being provided in the form of latent as well as sensible heat had to be dealt with. This was dealt with by means of a flat portion to the system curve representing no change in temperature but a change in energy, as typically seen when steam condenses. The remainder of the system curve consisted of a slanted line where the gradient represented the flowrate. The flat and slanted lines were combined into the system curve which would be matched against the process composite curve. Figure 3.6 shows the composite curves of a process as well as the two-part system curve. As with cooling systems a pinch point was found and this represented the minimum steam flowrate.

The network design procedure was approached in a different manner to that of Kim and Smith (2001). Coetzee and Majozi (2008) attempted to create a more formal method for network design. They turned to mathematical programming and derived a model for the system based on a superstructure similar to that of Majozi and Moodley (2007). With the minimum flowrate known from the graphical targeting, the resulting model only contained nonlinear terms in the energy balance constraints. These can be linearised using the same conditions for optimality by Savelski and Bagajewicz (2000) as used in the work by

Majozi and Moodley (2007). The resulting LP model was subsequently solved in a relatively short amount of time.

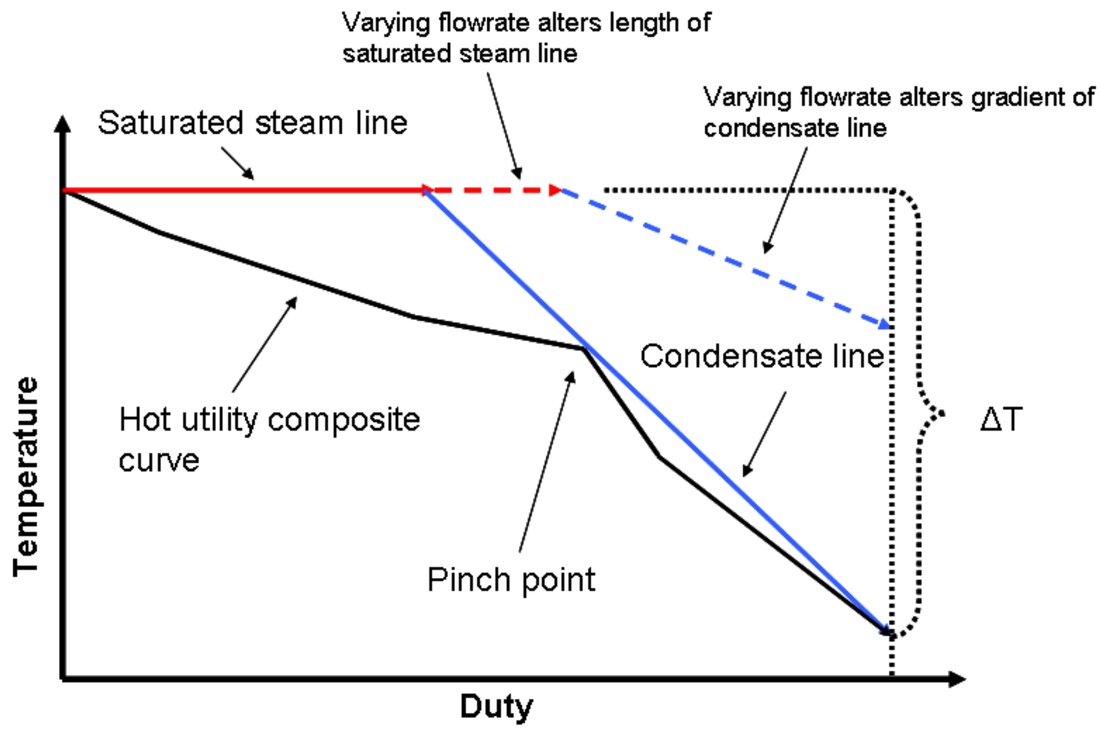

Figure 3.6: Composite and system curve showing latent and sensible heat.

The authors then attempted to create a mathematical model of the entire system that could both design the network as well as target for a minimum flowrate. This model contained binary variables to distinguish between a heat exchanger receiving steam or condensate, as a feed of both was physically unrealistic. The binary variables rendered the model an MINLP, however using the same conditions of optimality as before the model was linearised to an MILP. The presence of the binary variables did increase the solution time, however the need for the graphical targeting technique was removed. The simplicity of the mathematical approach does make it easy to include constraints, such as removing local recycle of condensate, as this would be thermodynamically ineffective for heating.

In conclusion, the authors developed two techniques to target for a minimum flowrate and design the HEN for heating systems. The two techniques both draw from previous work in cooling systems, but do cater for the additional aspect of sensible and latent heat with ease. The work does however only focus on the HEN and not the other aspects of the heating system such as turbines and the steam boiler. Further aspects not only unique to heating systems, such as pressure drop have also not been considered. This work therefore provides an excellent basis for further research into the effects of reducing the steam flowrate in heating systems.

3.4 Pressure Drop

Heat integration using the principles of reuse and recycle is very effective at reducing the utility requirement of a system. One of the major consequences of this reuse and recycle is the consequent increase in the pressure drop of the system. The increased pressure drop could lead to the need for additional fluid movers in retrofit design and additional capital expenditure in grassroot design. Thus pressure drop is often included as an optimisation objective in many HEN design techniques.

Nie and Zhu (1999) identified the need to consider pressure drop in a HEN retrofit design. The changes to the HEN could be as a result of changes to the process or even plant expansion. They found that pressure drop is complex as it is not only dependent on the process streams themselves, but also on the network topology.

They decomposed the retrofit problem into two parts, the first being to find the optimal exchangers that require additional heat exchange area, the second to find the shell arrangement that is most beneficial for pressure drop. They initially tried to include the possibility for additional area and all possible shell arrangements in one superstructure. This resulted in an extremely complex and unmanageable optimisation problem; hence the problem was split into two parts. The resulting mathematical model utilises pressure drop correlations based on velocity and heat exchange area. These correlations were derived by Nie (1998).

This work highlights the importance of considering pressure drop within a manageable framework due to its complex nature. Relationships for pressure drop as a function of fluid velocity and heat exchange area are also given, which make it possible to relate pressure drop to other easily accessible stream information. This work is however limited by the retrofit nature of the problem.

A grassroot HEN design with pressure drop considerations was considered by Nie and Zhu (2002), which is discussed next. They considered another aspect of heat exchanger network design that adds to capital cost. Pressure drop occurs in all fluid movement since fluids are most commonly transported from a point of high pressure to one of lower pressure. Sometimes this pressure drop becomes too high and reduces the effectiveness of the HEN. Pumps and compressors are used to overcome pressure drop but these add to the capital cost of the design. As such the authors seek to minimise pressure drop for the network.

Pressure drop is largely interconnected with other design variables. As a result the authors determine the optimal ΔT_{min} for the network by a three way trade off with heat transfer area, utilities and pressure drop.

Several earlier attempts at optimising pressure drop have been done by the likes of Jegede (1990) and Jegede and Polley (1992). These authors attempted to optimise the heat transfer coefficient, or h-values, and then calculate pressure drop based on these h-values as they are closely related in heat exchanger design (Sinnot, 2005). A flaw in this approach can be picked up instantly as the optimal pressure drop is in fact not determined, but it is simply the by-product of another optimisation. Also this work did not consider the intrinsic relationship between the pressure drop and the network structure or even the number of heat transfer units, or shells.

The authors then developed a new pressure drop correlation that was less dependent on the network geometry so as to avoid some complexity. They used a similar technique to Ahmed and Smith (1989), where the number and shells and heat exchange area were optimised for individual streams. Then the authors calculated pressure drop related costs for each stream. Nie and Zhu (2002) then used this approach along with a block decomposition technique from Zhu *et al.* (1995) with a fixed ΔT_{LM}. This resulted in linear area and pressure drop constraints which allow the utilities, area and pressure drop to be optimised in the same operation.

This work attempts to decouple the complex pressure drop relationship from the individual heat exchangers and the network structure. As a result a very complicated superstructure is developed to represent the HEN. The overall model is a MINLP and a global optimum is subsequently not guaranteed. Useful correlations for pressure drop and heat transfer area are developed and collected by the authors, however true pressure drop optimisation should include the network structure.

Having worked on a heuristic approach to cooling system design that considered both the cooling tower and the heat exchanger network, Kim and Smith (2003) then proceeded to examine the effects of water reuse on the pressure drop for the system. Their previous work resulted in series/parallel heat exchanger networks with water reuse between individual heat exchangers. Consequently more pressure drop would occur in the network. Additional pressure drop creates a problem in retrofit designs as additional fluid movers may be required to compensate. This increases the capital cost of the retrofit operation. Thus the authors attempted to a find a means of minimising pressure drop in a systematic manner.

In their previous work the authors minimised the cooling water flowrate using conceptual methods. The complex relationships that exist in pressure drop, such as the dependence on both flowrate and network structure make it infeasible to use graphs to minimise pressure drop. Thus the authors turned to mathematical modelling. Pressure drop correlations were developed in the work by Nie and Zhu (1999). They used velocity as the bridge between heat exchanger design variables such as heat transfer coefficient, heat transfer area and pressure drop. They present rather simple equations that are used by Kim and Smith (2003) to create their pressure drop model. The very nature of retrofit operations implies that additional piping will be required; as such the authors use conventional piping system design correlations for pressure drop. Normally many aspects to piping systems create large optimisation problems. The authors therefore remove several degrees of freedom by using optimal piping design heuristics of Peters and Timmerhaus (1991) which consider aspects such as flowrate to diameter ratios. With the above correlations the authors were able to present rather simple but fairly accurate representations for shell and tube side pressure drop as well as pressure drop in pipes as a function of flowrate.

The systematic approach then proposed for considering pressure drop in cooling system design would be to examine the thermal performance of the cooling tower and design the network so as to minimise the necessary flowrate. Then the pressure drop for networks using that specified flowrate would be minimised to arrive at a network design. If the minimal pressure drop was above certain limits the flowrate could be adjusted so as to satisfy the pressure drop limitations of the system fluid movers.

The intensive nature of pressure drop in HENs was dealt with by Kim and Smith (2001) by viewing the problem in a node format. Pressure drop over pipes and heat exchangers join nodes of different pressure. The nodes then represent mixers and splitters appropriately linked to heat exchangers. The source node would be the splitter connected to the high pressure stream from the cooling tower. The sink mixer would then represent the collection of all the streams before they proceed to the cooling tower. The authors then used the concept of longest or critical path problems, discussed in Gass (1985), to structure the problem such that the largest pressure drop over the network is considered. This overall pressure drop from the source splitter to sink mixer would then be minimised by rearranging the network. This would ensure that the minimal correct pressure drop over the network would be found. A superstructure was used to derive the mathematical model to represent this system. Since certain nodes may or may not be active for various networks binary variables were introduced to cater for this. As with most mathematical models developed from a generalised superstructure, heuristic rules can be used to reduce the size of the model. In this case impractical node pairings such as local reuse were removed as this would violate the primary principle of fluid flowing from a point of high pressure to a point of low pressure.

The pressure drop correlations and energy balances for the heat exchangers are nonlinear. This, combined with the binary variables for the existence of nodes, makes the model of the MINLP type. These are usually difficult to solve but a number of techniques are used by the authors to find a solution. The nonlinear terms from the energy balance are of the form of a flowrate multiplied by a temperature. The authors realise that by using the limiting outlet temperatures of the streams they can linearise these bilinear streams. They then examine the pressure drop correlations for the flowrate ranges of the HEN. They found that these can be approximated by a straight line which greatly simplifies the solution. The piping pressure drops are more nonlinear however, but these are approximated by piecewise linear functions. With these techniques the model is simplified to a MILP type problem for which there are several solution algorithms. The authors also consider the bilinear technique used by Quesada and Grossmann (1995). This relaxation linearisation gives the MINLP a starting point. The MINLP and linearised solutions can then be compared and if they coincide it can be concluded that the solution is globally optimum.

The authors explore a major concern in HEN rearrangement by using a novel node representation and simple yet accurate approximations for pressure drop. The resulting MINLP is then linearised using various relaxation or approximation techniques so as to give a feasible solution. These steps are intended to guide designers to retrofit designs that could eliminate the need for additional cooling capacity or fluid moving capacity.

3.5 Conclusions

Process to process heat integration has been extensively investigated and optimised. Techniques such as Pinch Analysis and mathematical programming have yielded extremely large savings in capital and operating expenses. Utility systems have as yet not received the same degree of complex analysis. Kim and Smith (2001) showed the value of holistic utility optimisation in their work on cooling systems. They successfully reduced the necessary cooling water flowrate as well as improved the cooling tower operation. Coetzee and Majozi (2008) reduced the steam flowrate for a heating system HEN. They did not consider the effect on other units in the heating system such as the steam boiler.

The reduction of utilities is accomplished by rearranging a HEN from a parallel connection to a series/parallel connection (Kim and Smith, 2001; Coetzee and Majozi, 2008). Kim and Smith (2003) found that this more complex network connection had the effect of increasing the pressure drop throughout the network. They then addressed this problem for cooling systems.

From the literature it can be seen that a holistic steam system optimisation methodology is required to increase the effectiveness of the work by Coetzee and Majozi (2008) by considering the system boiler efficiency as well as the network pressure drop.

3.6 References

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4. Model Development

The mathematical models that are developed in this chapter are shown separately but used in conjunction with one another. Essentially there are three areas which are modelled, i.e. the HEN, the rest of the steam system with special focus on the boiler and finally the pressure drop through the HEN. Since the core of the problem is concerned with the reduction of the steam flowrate to the HEN, this area will be investigated first. The steam system and boiler efficiency models will be used with the HEN models and are therefore developed second. The pressure drop minimisation will be presented last.

4.1 HEN Synthesis Model

The hot utility system on a plant provides heat to process streams that cannot be heated by ordinary process to process heat integration. As such the primary objective for the heating system and consequently the HEN constraints is to ensure that the process streams receive the necessary amount of heat. Also the heat provided to the process streams must be at a sufficient temperature to satisfy the ΔT_{min} of the heat exchangers.

The first objective in the problem statement is to reduce the amount of steam required for heating in a plant. This is done by reducing the amount of steam used to heat process streams in a HEN. Reducing the amount of steam to a HEN at a single pressure level has been dealt with by Coetzee and Majozi (2008). The mathematical formulation of this work will be included as it forms a part of models presented in this chapter. Catering for multiple steam levels requires a different superstructure and a slightly altered mathematical framework and as such this model will be developed after the recap of the model by Coetzee and Majozi (2008).

The HEN constraints can be used to find the minimum steam flowrate for a particular HEN. In the completed models that include the boiler efficiency and pressure drop constraints, the minimum flowrate is found and used as a lower bound for the model. The HEN constraints are then also included in the entire model to ensure that the necessary amount of heat is always available for the process, even if the predetermined minimum steam flowrate cannot be met. This gives the model the flexibility to rearrange the HEN in a manner that ensures that the process streams are always supplied with the correct amount of heat.

4.1.1 Single steam pressure level (Coetzee and Majozi, 2008)

Figure 4.1 shows the superstructure used to derive the constraints in this MILP. Referring to the figure, *FS* is the total amount of saturated steam from the boiler. Each heat exchanger, represented by *i* or *j* in the figure, can receive saturated steam or recycled/reused condensate, represented in the figure by *SS* and *FRR* respectively. The outlet from each heat exchanger can be saturated or subcooled condensate, depending on the nature of the inlet stream. The same outlet can return to the boiler in the form of *FR* or be recycled/reused to other heat exchangers. The total return to the boiler is then represented by the *FB* variable. The indices in the constraints and in the figure show which heat exchanger the variables are associated with. If two indices appear, the first represents the source heat exchanger and the second the sink. In reality, each heat exchanger in the superstructure represents a process stream requiring heating, so these terms are used interchangeably. This fact becomes important when multiple heat exchangers are introduced to heat a single process stream.

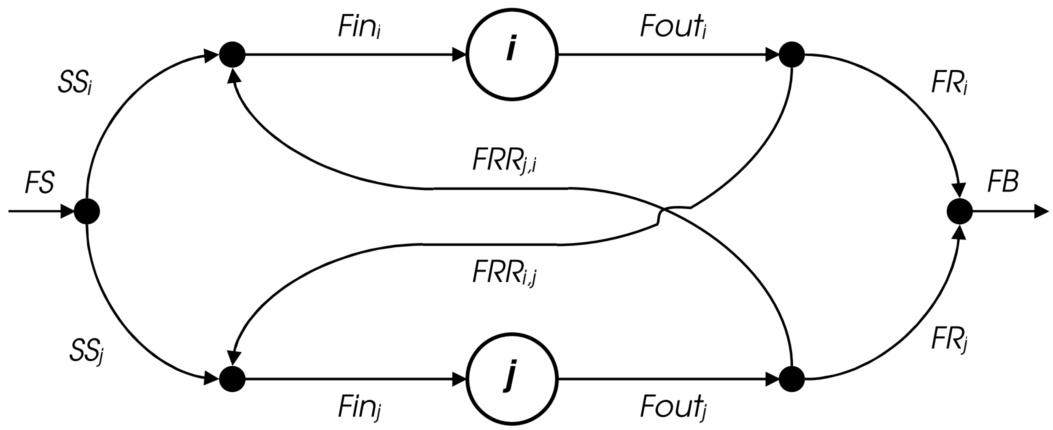

Figure 4.1: Superstructure representing the HEN at single pressure level.

The first section of the MILP encompasses the mass balance constraints. Constraint (4.1) shows how FS is comprised of the sum of SS_i for all the heat exchangers i. Constraint (4.2) is the inlet mass balance for each heat exchanger i, while Constraint (4.3) shows the outlet mass balance. Constraint (4.4) is then the mirror of Constraint (4.1), showing the return stream to the boiler.

$$FS = \sum_{i \in I} SS_i \tag{4.1}$$

$$Fin_i = SS_i + \sum_{j \in I} FRR_{j,i} \qquad \forall i \in I \tag{4.2}$$

$$Fout_i = FR_i + \sum_{j \in I} FRR_{i,j} \qquad \forall i \in I \tag{4.3}$$

$$FB = \sum_{i \in I} FR_i \tag{4.4}$$

The two remaining mass balances simply state the conservation of mass for a single heat exchanger i, in the case of Constraint (4.5) and for the entire HEN in Constraint (4.6).

$$Fin_i = Fout_i \qquad \forall i \in I \tag{4.5}$$

$$FS = FB \tag{4.6}$$

The inlet stream for a heat exchanger cannot consist of both steam and condensate, since this is realistically impractical. As such the inlet to a heat exchanger is controlled using binary variables. The variable controlling condensate is represented by x_i and the variable controlling steam is represented by y_i. Therefore if a heat exchanger receives saturated steam the binary variable y_i associated with that heat exchanger will take on a value of 1, whereas x_i for that heat exchanger will take on the value of 0.

Implementing these binary variables in the normal mass balance constraints would lead to a situation where the inlet flowrates SS_i and $FRR_{i,j}$ would be multiplied by the binary variables y_i and x_i respectively. This nonlinearity can be linearised by the Glover transformation (Glover, 1975). An explanation of this technique is given in Appendix B. There is however a simpler means of employing the binary variables which require the upper limits of steam and condensate to the heat exchanger. Constraints (4.7) and (4.8) show these limits.

$$SS_i^U = \frac{Q_i}{\lambda} \qquad \forall i \in I \tag{4.7}$$

$$FRR_i^U = \frac{Q_i}{c_p\left(Tin_i^L - Tout_i^L\right)} \qquad \forall i \in I \tag{4.8}$$

In Constraint (4.7), Q_i is the duty for heat exchanger i while λ is the latent energy of the saturated steam. In Constraint (4.8), Tin^L_i and $Tout^L_i$ are the limiting temperature values for the heat exchanger while c_P is once again the specific heat capacity. These limits are constant for each heat exchanger and as such the inequalities shown in Constraints (4.9) and (4.10) are linear.

$$SS_i \leq SS_i^U y_i \qquad \forall i \in I \tag{4.9}$$

$$\sum_{j \in i} FRR_{j,i} \leq FRR_i^U x_i \qquad \forall i \in I \tag{4.10}$$

Constraint (4.11) then ensures that each process stream, represented by the heat exchangers, is only heated by one heat exchanger. This constraint effectively ensures that steam and condensate will not enter the same heat exchanger.

$$y_i + x_i = 1 \qquad \forall i \in I \tag{4.11}$$

A situation may arise where the restriction of only one heat exchanger supplying heat to a process stream leads to a sub-optimal minimum flowrate. Thus an alternative to Constraint (4.11) is given. By allowing a certain number of process streams to be heated by two heat exchangers (implying one heat exchanger supplied with saturated steam, the other with condensate) an improved minimum flowrate may be found. Therefore the variable *n* is included in the upper limit of the sum for the binary variables, where *n* is the number of process streams that may be heated by both steam and condensate. The value for *n* is not known beforehand, but can take a maximum value of the number of heat exchangers and a minimum of zero. Thus it can be found by iteration or included in the model as a variable. Consequently Constraints (4.12) and (4.13) can be used instead of Constraint (4.11).

$$\sum_{i \in I} y_i + \sum_{i \in I} x_i \geq |i| \quad (4.12)$$

$$\sum_{i \in I} y_i + \sum_{i \in I} x_i \leq |i| + n \quad (4.13)$$

The energy balance section of the MILP will be considered next. The energy gained by saturated steam is shown in Constraint (4.14), while energy gained by condensate is shown in Constraint (4.15). The duty for each heat exchanger/process stream must be satisfied and Constraint (4.16) ensures this. The restrictions on the mass flowrates of steam and condensate will propagate through the energy balance constraints and as such there is no need for restrictions in the energy balances. Q_i^S and Q_i^L represent the actual amount of energy gained from steam and condensate in the following constraints respectively.

$$Q_i^S = SS_i \lambda \qquad \forall i \in I \quad (4.14)$$

$$Q_i^L = \sum_{j \in i} \left(c_p SL_{j,i} T_{sat} \right) + \sum_{j \in I} \left(c_p L_{j,i} Tout_j \right) - \left(c_p Fout_i Tout_i \right) \qquad \forall i \in I \quad (4.15)$$

$$Q_i = Q_i^S + Q_i^L \qquad \forall i \in I \quad (4.16)$$

Several other constraints are required for the completeness of the MILP. As previously mentioned the recycle/reuse variable $FRR_{j,i}$ can be saturated or subcooled condensate, represented by $SL_{j,i}$ and $L_{j,i}$ respectively. Constraint (4.17) is needed for this distinction.

$$FRR_{j,i} = SL_{j,i} + L_{j,i} \qquad \forall i, j \in I \quad (4.17)$$

The return streams to the boiler from each heat exchanger *i* can be saturated or sub cooled. Constraint (4.18) allows for this distinction, where FRS_i represents saturated condensate return flow to the boiler and FRL_i subcooled condensate return flow to the boiler.

$$FR_i = FRS_i + FRL_i \qquad \forall i \in I \tag{4.18}$$

The amount of saturated condensate recycled/reused or returned to the boiler is limited by the amount of saturated steam supplied to the heat exchanger which is illustrated in Constraint (4.19). Constraint (4.20) then shows the equivalent for subcooled condensate and subcooled return to the boiler. In this constraint *j* and *j'* represent any other heat exchangers.

$$SS_i = \sum_{j \in I} SL_{i,j} + FRS_i \qquad \forall i \in I \tag{4.19}$$

$$\sum_{j' \in I} SL_{j',i} + \sum_{j' \in I} L_{j',i} = \sum_{j \in I} SL_{i,j} + FRL_i \qquad \forall i \in I \tag{4.20}$$

Local recycle of subcooled condensate is practically not common, but mathematically possible with the current constraints. Constraint (4.21) is included in the formulation to prevent this.

$$L_{i,j} = 0 \qquad \forall i, j \in I, \qquad i = j \tag{4.21}$$

The final section of the MILP is the objective function. The overall steam flowrate to the HEN is to be minimised, which is shown in Constraint (4.22). In this constraint *FS* and *FB* can be used interchangeably as they are equal.

$$MinZ = FS \tag{4.22}$$

Constraint (4.15) is however nonlinear. Savelski and Bagajewicz (2000) proved that setting wastewater outlet concentrations to their limit would lead to minimum wastewater flowrates. Since an analogy can be made between concentration and temperature as driving forces in mass and heat transfer respectively Constraint (4.15) can be linearised by setting the outlet temperature of each heat exchanger to its limiting value, which is constant. Thus Constraints (4.1) to (4.22) constitute the HEN MILP for single steam pressure levels.

4.1.2 Multiple steam pressure levels

The multiple steam levels addressed in this section typically originate in systems where turbines are employed. The exhaust streams from the turbines that operate at various pressures differ. These streams are then often used as a heating utility and as a result these pressure levels must be catered for in the HEN constraints. Figure 4.2 shows a superstructure that can be used to derive the mass and energy balances for multiple pressure levels. The constraints in this section are very similar to the single pressure level constraints, however an extra index is added to the variables depicting saturated steam or condensate and a number of additional restrictions are needed.

In Figure 4.2, high pressure steam is produced inside the boiler. This then proceeds to the steam system as *FS*. The various steam pressure nodes are denoted as *l*, *m* and *n*, with *l* being at the highest pressure levels. The other lower pressure levels come as a result of the turbines which produce shaft work. Steam at various pressure levels can provide heat to a process stream, along with recycled/reused condensate. These are shown in the figure as *SS* and *FRR* respectively. The outlet from the process heat exchangers can proceed to the boiler or be recycled/reused. This is done using the *FR* and *FRR* variables respectively. Then the total return stream to the boiler, *FB*, completes the cycle. Here any makeup water has been omitted for the sake of simplicity.

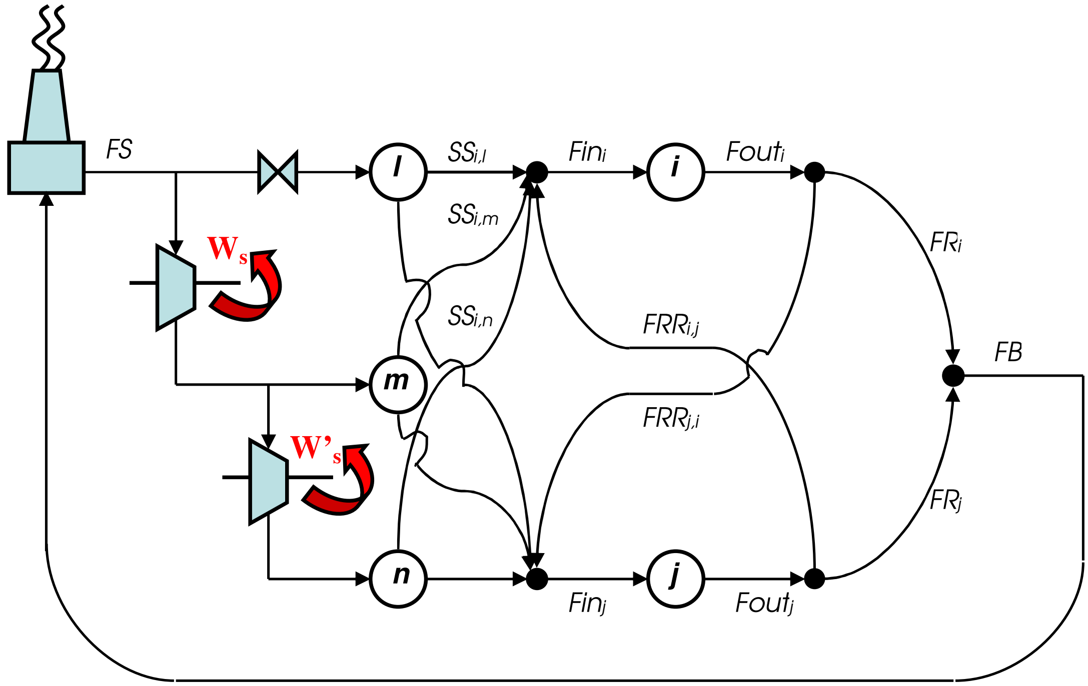

Figure 4.2: Superstructure incorporating multiple pressure levels.

Similar to the single pressure level MILP, the mass balances are derived first. As mentioned previously, the saturated steam variable includes an additional index, *l*, so as to cater for the multiple pressure levels. Thus the total steam flow from the boiler is split into $SS_{i,l}$ for each heat exchanger *i*, shown in Constraint (4.23). The inlet to each heat exchanger can be made up of saturated steam and recycled/reused condensate as shown in Constraint (4.24). Since the condensate could be saturated, the associated variable must include the pressure level index, making it $FRR_{i,l}$. The outlet from each heat exchanger can then return to the boiler or be recycled/reused as shown in Constraint (4.25). The boiler return stream can also be saturated and as such this variable also receives the pressure level index, forming $FR_{i,l}$ The boiler return streams then join to form *FB* in Constraint (4.26).

$$FS = \sum_{i\in I, l\in L} SS_{i,l} \tag{4.23}$$

$$Fin_i = \sum_{l \in L} SS_{i,l} + \sum_{j \in I, l \in L} FRR_{j,i,l} \qquad \forall i \in I \tag{4.24}$$

$$Fout_i = \sum_{l \in L} FR_{i,l} + \sum_{j \in I, l \in L} FRR_{i,j,l} \qquad \forall i \in I \tag{4.25}$$

$$FB = \sum_{i \in I, l \in L} FR_{i,l} \tag{4.26}$$

As before, the two conservation of mass constraints complete the mass balance. Constraint (4.27) is for individual heat exchangers and Constraint (4.28) is for the entire HEN.

$$Fin_i = Fout_i \qquad \forall i \in I \tag{4.27}$$

$$FS = FB \tag{4.28}$$

As in the case for a single steam pressure level, a heat exchanger cannot receive saturated steam as well as any form of condensate as an inlet. A heat exchanger can also not receive saturated steam at different pressure levels. To cater for these restrictions binary variables are employed once again. In this case the binary variable x_i is still used to represent condensate supply to a heat exchanger, while the binary variable $y_{i,l}$ will represent saturated steam at pressure level l. The extra index for $y_{i,l}$ will be used to prevent steam at various pressure levels entering the same heat exchanger. As before, if a heat exchanger receives saturated steam at pressure level l as an inlet, the binary variable $y_{i,l}$ will take the value of 1, whereas the binary variable x_i will take the value 0 for that heat exchanger.

A process stream associated with heat exchanger *i* can be heated by steam at more than one pressure level if an additional heat exchanger is added for that stream. This will incur additional capital cost for the HEN and as such it can be restricted. If the sum of the $y_{i,l}$ is limited to a value of 1 or 0 for a heat exchanger *i* then it can be inferred that heat exchanger *i* can only be heated by steam at one pressure level or by no steam at all. This is demonstrated in Constraint (4.29). If $y_{i,l}$ can take a value of greater than 1 then more steam pressure levels can be used to heat the process stream associated with *i*. Constraint (4.30) shows this, with the variable *m* being used to show when additional steam pressure levels are allowed. Either one of these constraints can be implemented in the model, depending on the circumstances. Constraint (4.30) is the more flexible and may lead to additional steam savings, but the capital cost of additional heat exchangers may not be worthwhile.

$$\sum_{l \in L} y_{i,l} \leq 1 \qquad \forall i \in I \qquad (4.29)$$

$$\sum_{l \in L} y_{i,l} \leq 1 + m \qquad \forall i \in I \qquad (4.30)$$

Constraints (4.29) and (4.30) are used in conjunction with the binary variable restrictions given for the single steam pressure level model. They will be restated however so that $y_{i,l}$ can be summed over all the pressure levels *l* for a single heat exchanger *i*. Once again the upper bounds for the steam and condensate flowrates are calculated and used in the restrictions so that the Glover, (1975) transformation need not be implemented. The two upper bound constraints are shown below. Constraint (4.31) now includes the different λ values associated with steam at various pressure levels, shown as λ_l. Constraint (4.32) is essentially identical to Constraint (4.8) in the single pressure level model.

$$SS_{i,l}^{U} = \frac{Q_i}{\lambda_l} \qquad \forall i \in I \tag{4.31}$$

$$FRR_i^U = \frac{Q_i}{c_p\left(Tin_i^L - Tout_i^L\right)} \qquad \forall i \in I \tag{4.32}$$

These constraints are used along with the binary variables to restrict the flow of steam or condensate to heat exchanger *i*. Constraint (4.33) sets an upper bound on steam usage while Constraint (4.34) stipulates the maximum condensate usage.

$$SS_{i,l} \leq SS_{i,l}^{U} y_{i,l} \qquad \forall i \in I,\ \forall l \in L \tag{4.33}$$

$$\sum_{j \in i} FRR_{j,i} \leq FRR_i^U x_i \qquad \forall i \in I \tag{4.34}$$

The following constraints then control whether saturated steam, condensate or both steam and condensate are used to heat the process stream associated with heat exchanger *i,* much the same as for the single steam pressure level case. In the case where only steam or only condensate is to be used, Constraint (4.35) is implemented. This will have to be used with Constraint (4.29) to prevent any additional heat exchangers. Constraints (4.36) and (4.37) are then used if some process streams can be heated with both steam and condensate. If no process streams may be heated by multiple steam levels then Constraints (4.36) and (4.37) should be used with Constraint (4.29). Then there can be *n* streams heated by steam and condensate. If multiple steam levels are allowed for the same process stream then Constraints (4.36) and (4.37) should be used in conjunction with Constraint (4.30) and there can be at most n + m streams heated by steam and condensate. This does then allow steam at multiple pressures to heat a stream.

$$\sum_{l \in L} y_{i,l} + x_i = 1 \qquad \forall i \in I \tag{4.35}$$

$$\sum_{i \in I, l \in L} y_{i,l} + \sum_{i \in I} x_i \geq |i| \tag{4.36}$$

$$\sum_{i \in I, l \in L} y_{i,l} + \sum_{i \in I} x_i \leq |i| + n + m \tag{4.37}$$

As before the energy balances are considered next. Constraint (4.38) shows the energy gained from saturated steam, $Q_{i,l}^S$. This differs slightly, as $Q_{i,l}^S$ includes the steam pressure level. Constraint (4.39) deals with the energy gained from condensate Q_i^L. By summing $Q_{i,l}^S$ and Q_i^L the duty for each process stream associated with heat exchanger i must be satisfied. Constraint (4.40) ensures this condition.

$$Q_{i,l}^S = SS_{i,l}\lambda_l \qquad \forall i \in I, \ \forall l \in L \tag{4.38}$$

$$Q_i^L = \sum_{j \in I, l \in L} \left(c_p SL_{j,i,l} T_{sat,l}\right) + \sum_{j \in I} \left(c_p L_{j,i} Tout_j\right) - \left(c_p Fout_i Tout_i\right) \qquad \forall i \in I \tag{4.39}$$

$$Q_i = \sum_{l \in L} Q_{i,l}^S + Q_i^L \qquad \forall i \in I \tag{4.40}$$

The remaining constraints complete the multiple steam level model. Constraint (4.41) expands $FRR_{j,i,l}$ into saturated and subcooled condensate while Constraint (4.42) expands $FR_{i,l}$ into saturated and subcooled condensate as well.

$$\sum_{l\in L} FRR_{j,i,l} = \sum_{l\in L} SL_{j,i,l} + L_{j,i} \qquad \forall i, j \in I \qquad (4.41)$$

$$\sum_{l\in L} FR_{i,l} = \sum_{l\in L} FRS_{i,l} + FRL_{i} \qquad \forall i \in I \qquad (4.42)$$

Constraint (4.43) and (4.44) are steam and condensate balances for heat exchanger *i* respectively.

$$SS_{i,l} = \sum_{j\in I} SL_{i,j,l} + FRS_{i,l} \qquad \forall i \in I,\ \ \forall l \in L \qquad (4.43)$$

$$\sum_{j'\in I, l\in L} SL_{j',i,l} + \sum_{j'\in I} L_{j',i} = \sum_{j\in I, l\in L} SL_{i,j,l} + FRL_{i} \qquad \forall i \in I \qquad (4.44)$$

As before, a restriction for local recycle must be implemented for practical purposes as shown in Constraint (4.45).

$$L_{i,j} = 0 \qquad \forall i, j \in I, \qquad i = j \qquad (4.45)$$

The hot stream in a heat exchanger has to be at a higher temperature than the cold stream for heat transfer to take place. While this may be an obvious statement in reality, mathematically it has to be catered for specifically in either the single or the multiple pressure level models. It is, however implicitly dealt with by Constraint (4.15) in the single pressure level model and Constraint (4.39) in the multiple pressure level model. The difference of the inlet and outlet temperatures occurs in both of the constraints. If the inlet temperature were to be lower than the outlet temperature Q_i^L would be negative for heat exchanger *i*. This would lead to infeasibility in the optimisation and as such this situation is avoided. However, when multiple steam levels are considered the constraint dealing with latent energy does not implicitly cater for this occurrence. A situation could therefore arise where saturated steam at a lower pressure level (and consequently lower temperature) provides energy to a process stream requiring high temperature heating. Thus Constraint (4.46) is required to prevent this from occurring and therefore replaces Constraint (4.38).

$$SS_{i,l} = 0 \qquad \forall i \in I, \ \forall l \in L, \qquad T_{sat,l} \leq Tin_i^L \tag{4.46}$$

By restricting the amount of saturated steam to heat exchanger *i* the restriction will propagate through the model to restrict the other associated variables such as $FRS_{i,l}$ and $SL_{j,i,l}$.

The flowrate of steam to the turbines is fixed, since it is assumed all of the shaft work delivered by the turbines is utilised in the process and this cannot be compromised. As such the total flow of saturated steam to the process at all the pressure levels associated with the turbines is fixed. These streams are represented by stream 5 and stream 6 if Figure 2.2 is used as a reference. Therefore the only steam stream that can be reduced is the stream going directly to the process which is shown as stream 2 in Figure 2.2. If this stream pressure level is denoted as *PP* (representing the process pressure, resulting from passing through the let down valve) then Constraint (4.47) is used to fix the other saturated steam flowrates.

$$\sum_{i \in I} SS_{i,l} = FF_l \quad \forall l \in L, \qquad l \neq PP \tag{4.47}$$

In Constraint (4.47) FF_l is the fixed flowrate at pressure level *l* resulting from association to the turbine train. *PP* is the process pressure, and thus the only sum of saturated steam not fixed is the steam at process pressure.

The objective function for this model is to minimise the saturated steam at process pressure. Constraint (4.48) shows this objective.

$$MinZ = \sum_{i \in I} SS_{i,l} \qquad l = PP \tag{4.48}$$

Constraints (4.23) to (4.48) represent the HEN model for multiple steam pressure levels. Once again the condensate energy balance, Constraint (4.39), is the only nonlinear constraint and as in the single pressure level case this constraint is linearised using the premise proposed by Savelski and Bagajewicz (2000).

4.2 Considering Boiler Efficiency with an External Preheater

The following sections will consider boiler efficiency in the steam system. In the region of interest in Figure 2.4 it can be seen that the boiler efficiency function is dependent on temperature. Bearing this in mind, the original boiler efficiency may be maintained by slightly increasing the return temperature, even for a large reduction in steam flowrate as shown in Figure 2.4.

4.2.1 Single pressure level HEN

A method for the reduction of steam flowrate to a HEN was developed by Coetzee and Majozi (2008). Section 3.2 showed how reducing the steam flowrate to a HEN had the effect of reducing the boiler efficiency as a result of the lower flowrate, but more significantly the lower return temperature to the boiler. The HEN constraints for single steam pressure level systems shown in Section 4.1 can effectively find the minimum steam flowrate for a particular HEN. Using this flowrate, as well as several other easily calculated variables, the new boiler efficiency for the steam system can be calculated. Figure 4.3 is a reproduction of Figure 2.1 which has been included here for convenience as the additional variables are explained by the figure. Since the steam flowrate has been reduced, the condensate tank and condenser have been removed from the figure as it is unlikely that they will be needed as the return stream is not as near the saturation point.

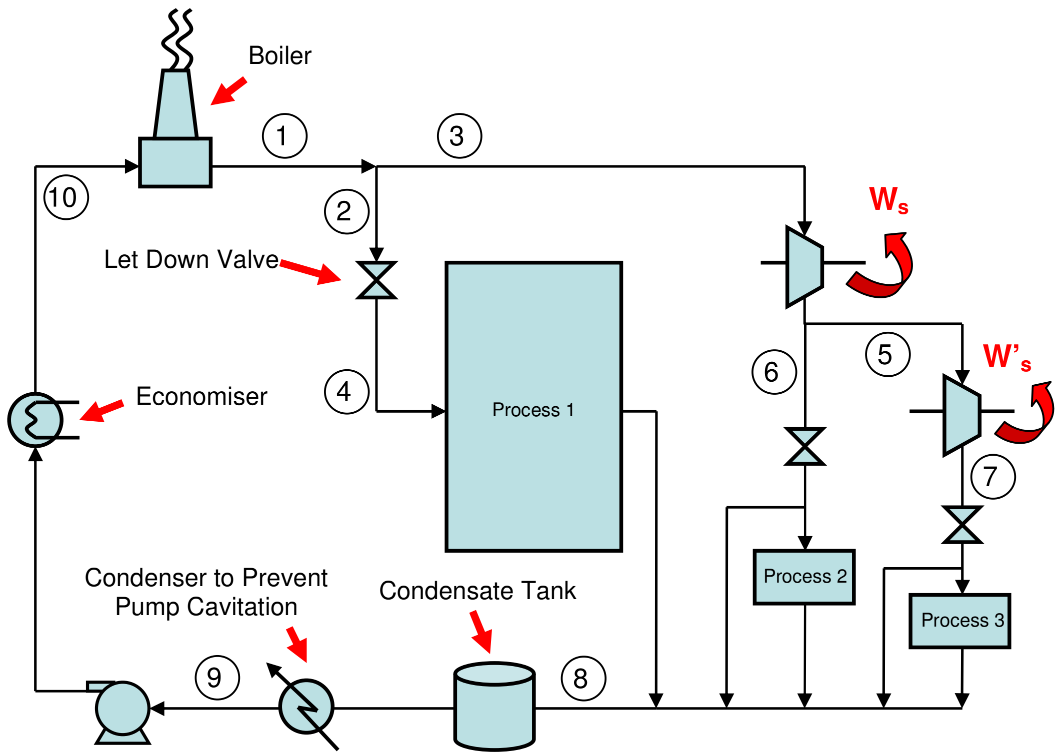

Figure 4.3: Steam system with labelled streams.

The constraint used to calculate the boiler efficiency requires the return flowrate and temperature to the boiler. The flowrate to the boiler is seen as stream 9 in Figure 4.3. Stream 9 is a combination of the outlet of the process that has undergone heat integration, stream 4, as well as the combined outlet of the two turbine background process, streams 6 and 7 which combine into stream 8. Thus the mass flowrate and outlet temperature of streams 4 and 8 must be known so as to calculate the return temperature and flowrate to the boiler. The mass flowrate of stream 4 is simply the steam flowrate through the HEN, calculated as *FS* in the HEN model. The temperature of stream 4 is determined by the return streams of all the heat exchangers in the HEN. Constraint (4.49) shows this calculation.

$$T_{proc} = \frac{\sum_{i \in I} FRS_i T_{sat} + \sum_{i \in I} FRL_i Tout_i^L}{FS} \tag{4.49}$$

As part of the linearisation of Constraint (4.15) the outlet of each heat exchanger is set to its lower limit, thus the subcooled condensate return temperature for each heat exchanger *i* is known. T_{proc} represents the return temperature of the process while *FS* is the mass flowrate. Stream 4 then combines with stream 8 to form stream 9. If it is assumed that the mass flowrate to the turbines is constant and that the outlet temperature of the background processes is also known then the temperature and flowrate of stream 8 will be known and can be assumed constant. These will be referred to as T_{turb} and F_{turb} in the following constraints respectively. The flowrate of stream 9 is thus simply the sum of *FS* and F_{turb}, whereas the temperature is calculated by Constraint (4.50).

$$T_{pump} = \frac{(T_{proc} FS) + (T_{turb} F_{turb})}{(FS + F_{turb})} \tag{4.50}$$

Since stream 9 proceeds to the pump, this temperature is referred to as T_{pump}. Stream 9 then passes through the preheater, where any additional heat is represented by $Q_{preheat}$. Stream 10 is thus the return stream to the boiler at flowrate *FS* + F_{turb} and temperature T_{boil}, calculated in Constraint (4.51).

$$T_{boil} = T_{pump} + \frac{Q_{preheat}}{c_P (FS + F_{turb})} \tag{4.51}$$

Then, substituting these variables into the constraint for boiler efficiency produces Constraint (4.52) for single steam pressure levels.

$$\eta_b = \frac{q\left(\left(FS + F_{turb}\right)/F^U\right)}{\left(c_p\left(T_{sat} - T_{boil}\right) + q\right)\left[\left(1+b\right)\left(\left(FS + F_{turb}\right)/F^U\right) + a\right]} \tag{4.52}$$

By considering the sensitivity analysis of this constraint that was done in Section 2.2, it can be seen that for a mass flowrate reduction of up to 50%, a significant reduction can be realised with a small increase in return temperature whilst still maintaining the boiler efficiency. The preheater in the system typically utilises the heat from stack gases in the boiler and is often referred to as an economiser. This preheater may not have the capacity to increase the boiler return temperature to a point where the boiler efficiency can be maintained. Therefore an additional heat exchanger is proposed that will make use of the sensible heat from the superheated high pressure steam that is usually lost in the let down valve. Figure 4.4 shows the proposed change to the steam system. In this figure the preheater using hot stack gases (hereafter referred to as the economiser) is omitted and assumed to be part of the boiler. Thus stream 9 passes through the new preheater, exiting as stream 10, the new boiler return stream.

Thus using this steam system set up the only variable to change is the temperature of stream 10, T_{boil}. Constraint (4.53) shows the new calculation for T_{boil}, where the preheater duty is calculated by the second term on the right hand side.

$$T_{boil} = T_{pump} + \frac{FS\left(h_{\sup} - h_{sat}\right)\theta}{c_P\left(FS + F_{turb}\right)} \tag{4.53}$$

In Constraint (4.53), h_{sup} is the enthalpy of the superheated high pressure steam leaving the boiler and h_{sat} is the enthalpy of the saturated steam entering the process. The variable θ represents the fraction of the sensible heat that can be used without the risk of condensation in stream 2.

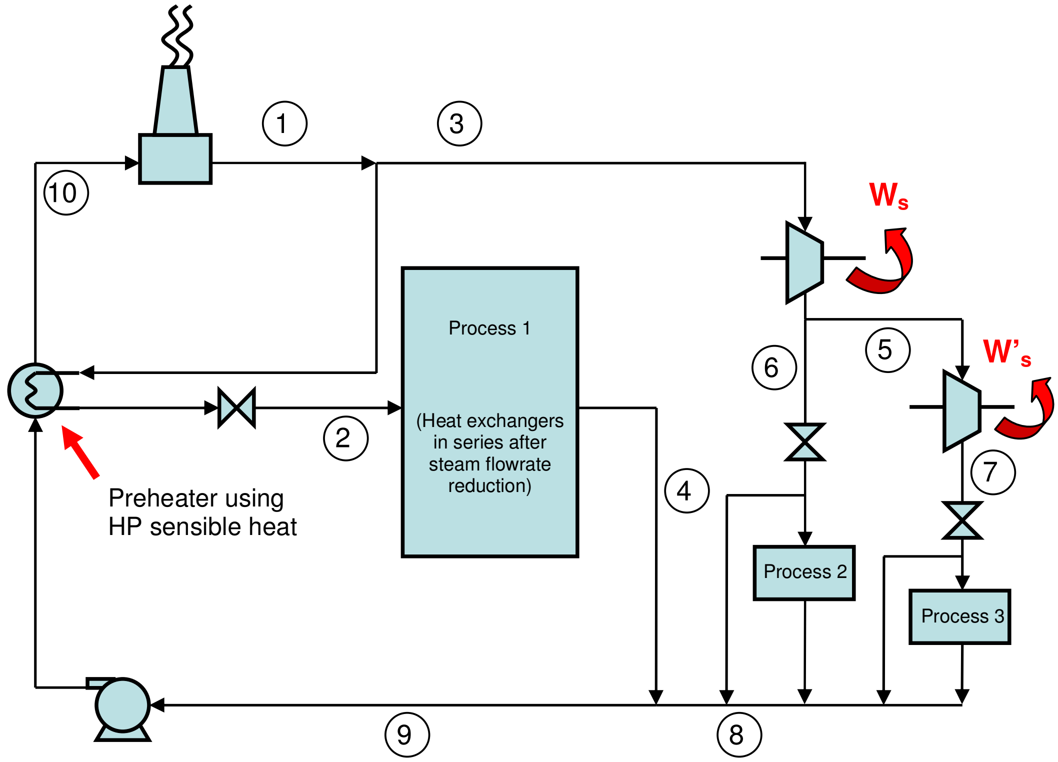

Figure 4.4: Steam system with proposed preheater.

These constraints are then combined with the single steam pressure level HEN constraints to form the mathematical model that reduces steam flowrate to a HEN whilst maintaining the original boiler efficiency. Two separate situations can be considered for the steam system, each using the same basic constraints, but focussing on two different objectives.

Case 1: Maintaining boiler efficiency with a possible compromise in steam flowrate

A situation may arise where the degree of superheating of the steam leaving the boiler is not able to provide sufficient energy to the boiler return stream to maintain the boiler efficiency. In this case the primary objective of the mathematical formulation may be to maintain the boiler efficiency and then reduce the steam flowrate as far as possible.

The solution strategy is as follows. Firstly the minimum steam flowrate is found using the MILP of Coetzee and Majozi (2008). Then a slack variable is added to the minimum flowrate and included in all the constraints pertaining to boiler efficiency. The boiler efficiency is then set to the original steam system value and used as a constraint in the formulation. Consequently the slack variable is minimised, as this represents the additional steam above the minimum flowrate required to maintain the boiler efficiency. Since the minimum flowrate will possibly be compromised a new HEN must also be found by the model, thus the HEN constraints must be included in the final formulation. Constraints (4.54) to (4.57) show the inclusion of the slack variable in the Constraints (4.49) to (4.53), with Constraint (4.53) replacing (4.51) as before. Constraint (4.58) is then the objective function.

$$T_{proc} = \frac{\sum_{i \in I} FRS_i Tsat + \sum_{i \in I} FRS_i Tout_i}{FS + slack^+} \tag{4.54}$$

$$T_{pump} = \frac{\left(T_{proc}\left(FS + slack^+\right)\right) + \left(T_{turb} F_{turb}\right)}{\left(FS + slack^+ + F_{turb}\right)} \tag{4.55}$$

$$T_{boil} = T_{pump} + \frac{\left(FS + slack^+\right)\left(h_{\sup} - h_{sat}\right)\theta}{c_p\left(FS + slack^+ + F_{turb}\right)} \tag{4.56}$$

$$\eta_b = \frac{q\left(\left(FS + slack^+ + F_{turb}\right)/F^U\right)}{\left(c_p\left(T_{sat} - T_{boil}\right) + q\right)\left[(1+b)\left(\left(FS + slack^+ + F_{turb}\right)/F^U\right) + a\right]} \tag{4.57}$$

$$MinZ = slack^+ \tag{4.58}$$

This formulation includes several nonlinear terms. One means to deal with this situation is to implement Quesada and Grossmann, (1994) type relaxation linearisations. An explanation of this linearisation appears in Appendix B. The solution to the relaxed problem is then found and used as a starting point for the exact, nonlinear model. In the case where the relaxed solution and the exact solution coincide then a globally optimum minimum flowrate is found.

Case 2: Minimum flowrate is maintained with a slight compromise in boiler efficiency

A situation may arise where the minimum flowrate is more desirable than maintained boiler efficiency. The only constraint to be changed in this situation is that of the boiler efficiency calculation and the objective function. These are shown in Constraints (4.59) and (4.60).

$$\eta_b - slack^- = \frac{q\left(\left(FS + F_{turb}\right)/F^U\right)}{\left(c_p\left(T_{sat} - T_{boil}\right) + q\right)\left[\left(1+b\right)\left(\left(FS + F_{turb}\right)/F^U\right) + a\right]} \qquad (4.59)$$

$$MinZ = slack^- \qquad (4.60)$$

An extension to Case 2 is a situation where there is indeed sufficient sensible energy to preheat the boiler feed and to maintain the efficiency. In this case it may be possible to utilise the sensible energy further, and increase the boiler efficiency. Since Constraint (2.8) and all of the constraints that are derived from it are not strict definitions of boiler efficiency any increase may not be physically realised to the extent calculated by the constraints, but some increased efficiency may be possible. Thus by adding a slack variable to the efficiency constraint and maximising this slack variable in the objective Constraints (4.61) and (4.62) are formed.

$$\eta_b + slack^+ = \frac{q\left(\left(TS + M_{turb}\right)/M^{\max}\right)}{\left(c_p\left(T_{sat} - T_{boil}\right) + q\right)\left[\left(1+b\right)\left(\left(TS + M_{turb}\right)/M^{\max}\right) + a\right]} \tag{4.61}$$

$$MaxZ = slack^+ \tag{4.62}$$

The objectives of Case 1 and Case 2 are quite different, and therefore the choice of which of the two to employ depends on the steam system under investigation. For instance a steam system operating in an arid region may be more inclined to Case 2 where efficiency is compromised for steam and consequently water savings. Other areas where fuel is expensive may be more inclined to use Case 1 so as not to compromise the boiler efficiency.

4.2.2 Multiple pressure level HEN

The multiple steam pressure level HEN constraints derived in Section 4.1 can effectively find a minimum steam flowrate for a particular HEN. This has the effect of reducing the boiler efficiency as a result of the reduced steam flowrate as well as the reduced boiler return temperature. As in the single steam pressure level case, the boiler efficiency can be calculated with several variables obtained from the steam system. Figure 4.5 is a reproduction of Figure 3.3 and has been included here for convenience. As a result of the lowered return temperature, the condensate tank and condenser have been removed since there is little risk of cavitation from the subcooled outlet of the HEN.

To calculate the boiler efficiency the return flowrate and temperature are required by Constraint (2.8). That translates to stream 9 in Figure 4.5. The flowrate of stream 9 is simply the reduced flowrate from the HEN, *FS* as calculated by the HEN section of the model. The temperature can be calculated once the temperature of stream 7 and the duty of the preheater are known. Stream 7 is the outlet from the process and as such the temperature T_{proc} is calculated by Constraint (4.63). The temperature of stream 9 is then calculated using Constraint (4.64). Using these variables the boiler efficiency is calculated by Constraint (4.65).

$$T_{proc} = \frac{\sum_{i \in I, l \in L} FRS_{i,l} Tsat_l + \sum_{i \in I} FRL_i Tout_i^L}{FS} \tag{4.63}$$

$$T_{boil} = T_{proc} + \frac{Q_{preheat}}{FS \times c_p} \tag{4.64}$$

$$\eta_b = \frac{q\left(FS/F^U\right)}{\left(c_p\left(T_{sat} - T_{boil}\right) + q\right)\left[\left(1+b\right)\left(FS/F^U\right) + a\right]} \tag{4.65}$$

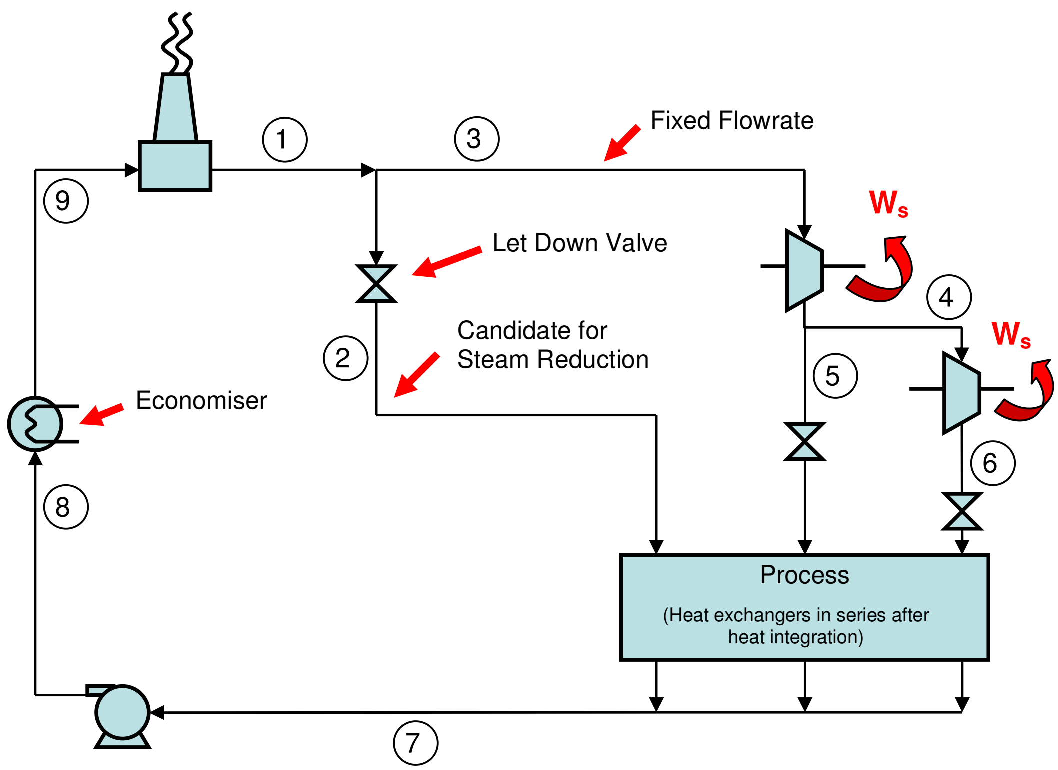

Figure 4.5: Steam system for multiple pressure level HEN.

The boiler efficiency can be maintained if the return stream to the boiler, stream 9, is heated to a high enough temperature. As in the single pressure level case the existing preheater may not have the capacity to heat the return stream to the necessary extent. Thus two additional preheaters are added that utilise sensible energy from the superheated high pressure steam directly from the boiler and from the medium pressure exhaust from the high pressure turbine. Figure 4.6 shows these proposed improvements to the steam system. The existing preheater, or economiser, is still included in the figure after the second additional preheater.

Streams 2 and 5 are now redirected to the additional preheaters. The flowrates of streams 2 and 5 are designated PP and MP, signifying that stream 2 is at the process pressure and stream 5 is at medium pressure. The stream from the boiler has flowrate *FS* from the HEN model. The flowrate of stream 6 is thus taken as LP, as it is at low pressure. The temperature of stream 9 is calculated by Constraint (4.66).

$$T_{boil} = T_{proc} + \frac{PP\left(h_{\sup}^{HP} - h_{sat}^{PP}\right)\theta + MP\left(h_{\sup}^{MP} - h_{sat}^{MP}\right)\phi}{FS \times c_P} \tag{4.66}$$

In Constraint (4.66), $h_{\sup}^{HP}$ is the superheated enthalpy of high pressure steam and h_{sat}^{PP} is the saturated enthalpy of steam at the process pressure. Similarly $h_{\sup}^{MP}$ and h_{sat}^{MP} are the superheated and saturated enthalpies of medium pressure steam. PP and MP are the flowrates of streams 2 and 5 as mentioned before. Finally *θ* and *Φ* are the fractions of sensible heat that can be extracted from streams 2 and 5 without causing any condensation in each.

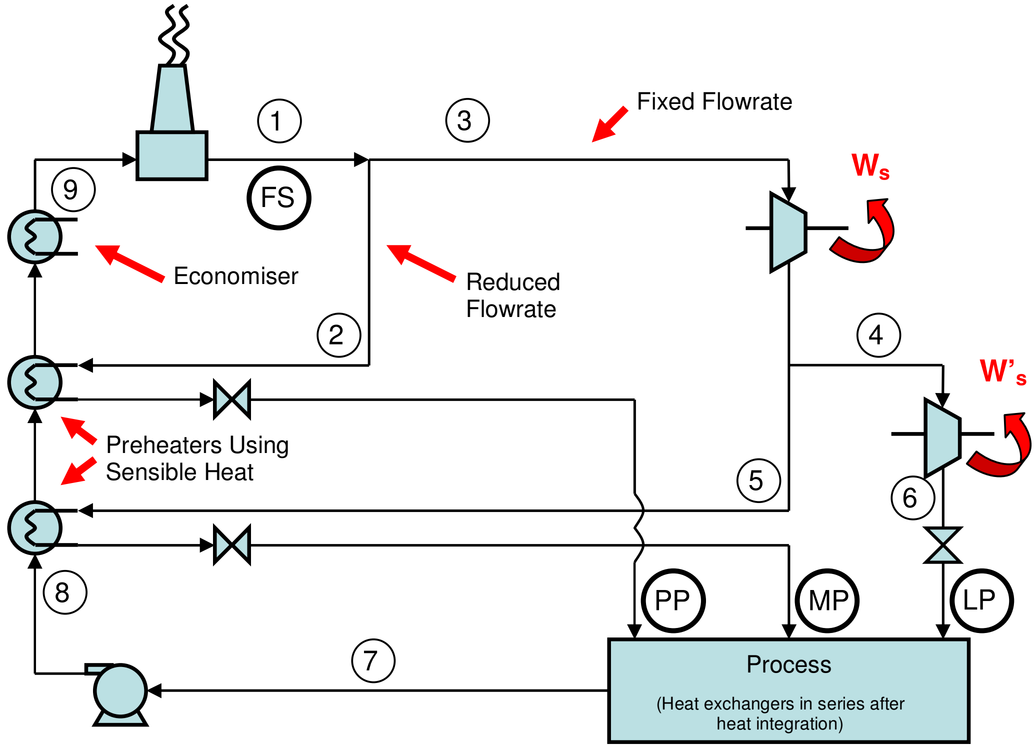

Figure 4.6: Steam system with additional preheaters utilising sensible heat.

Constraints (4.63) to (4.66) then constitute the rest of the multiple steam pressure level model. Constraint (4.66) replaces Constraint (4.64) to calculate the temperature of stream 9, so this portion of the model consists of only 3 constraints. As before, 2 situations can be considered where these constraints can be used to form 2 different objectives which are discussed below.

Case 1: Maintaining the boiler efficiency with a possible compromise in the minimum flowrate

As in the single pressure level case, when the boiler efficiency is made the primary objective it is possible that the minimum flowrate for the HEN may not be reached. In this scenario the minimum flowrate is determined by the HEN model and used as a lower bound in the formulation. A slack variable is then added to the minimum flowrate, while the boiler efficiency is set at the level prior to steam reduction, thereby becoming a parameter. The slack variable is then minimised. If the minimum flowrate is indeed compromised a new HEN will have to be found with this flowrate. Thus the HEN model forms part of the entire formulation. Constraints (4.67) to (4.69) replace Constraints (4.63) to (4.66), with Constraint (4.68) replacing (4.64) as before. Constraint (4.70) then becomes the objective function for the formulation.

$$T_{proc} = \frac{\sum_{i \in I, m \in M} FRS_{i,l} T_{sat,l} + \sum_{i \in I} FRS_i Tout_i}{FS + slack^+} \tag{4.67}$$

$$T_{boil} = T_{proc} + \frac{\left(FS + slack^+\right)\left(h_{\sup}^{HP} - h_{sat}^{PP}\right)\theta + MP\left(h_{\sup}^{MP} - h_{sat}^{MP}\right)\phi}{c_p\left(FS + slack^+\right)} \tag{4.68}$$

$$\eta_b = \frac{q\left(\left(FS + slack^+\right)/F^U\right)}{\left(c_p\left(T_{sat} - T_{boil}\right) + q\right)\left[(1+b)\left(\left(FS + slack^+\right)/F^U\right) + a\right]} \tag{4.69}$$

$$MinZ = slack^+ \tag{4.70}$$

This formulation contains several nonlinear terms. The Quesada and Grossmann, (1994) relaxation linearisation solution strategy is employed to solve a relaxed model and use this solution as the starting point for the exact model. An explanation of this linearisation appears in Appendix B.

Case 2: Maintain the minimum steam flowrate with a possible compromise in boiler efficiency

In this scenario a slack variable is added to the efficiency constraint while the minimum flowrate is set as a parameter. Since the minimum flowrate is achieved, there is no need to design a different network. As such the HEN model is only used to determine the minimum flowrate for the HEN under investigation. Constraint (4.71) is the new calculation for efficiency while Constraint (4.72) is the objective function.

$$\eta_b - slack^- = \frac{q\left(FS/F^U\right)}{\left(c_p\left(T_{sat} - T_{boil}\right) + q\right)\left[\left(1+b\right)\left(FS/F^U\right) + a\right]} \tag{4.71}$$

$$MinZ = slack^- \tag{4.72}$$

As before, this formulation can be extended. If the boiler efficiency can be maintained by not using all of the available sensible heat then it may be possible to increase the boiler efficiency. Once again it must be clearly stated that since Constraint (2.8) is not a true representation of boiler efficiency any gains using this formulation may not be achieved in reality, however some improvement is possible. Constraint (4.73) shows the new representation of the efficiency constraint while Constraint (4.74) is the objective function.

$$\eta_b + slack^+ = \frac{q\left(FS/F^U\right)}{\left(c_p\left(T_{sat} - T_{boil}\right) + q\right)\left[\left(1+b\right)\left(FS/F^U\right) + a\right]} \tag{4.73}$$

$$MaxZ = slack^+ \tag{4.74}$$

Since these 2 cases are quite different, the environment in which the steam system operates will likely play a role in which of the two formulations will be most beneficial to implement.

4.3 Considering Boiler Efficiency with a Dedicated Preheater

Another means of heating the boiler return stream to a point such that the boiler efficiency can be maintained is to add an additional heat exchanger to the HEN whose sole purpose is to preheat the return stream.

4.3.1 Single steam pressure level

Figure 4.7 shows the steam setup with this addition. As with the other improvements to the steam system the condensate tank and condenser have been omitted from the figure. Stream 9 is treated as the return stream to the boiler since the economiser is assumed to form part of the boiler.

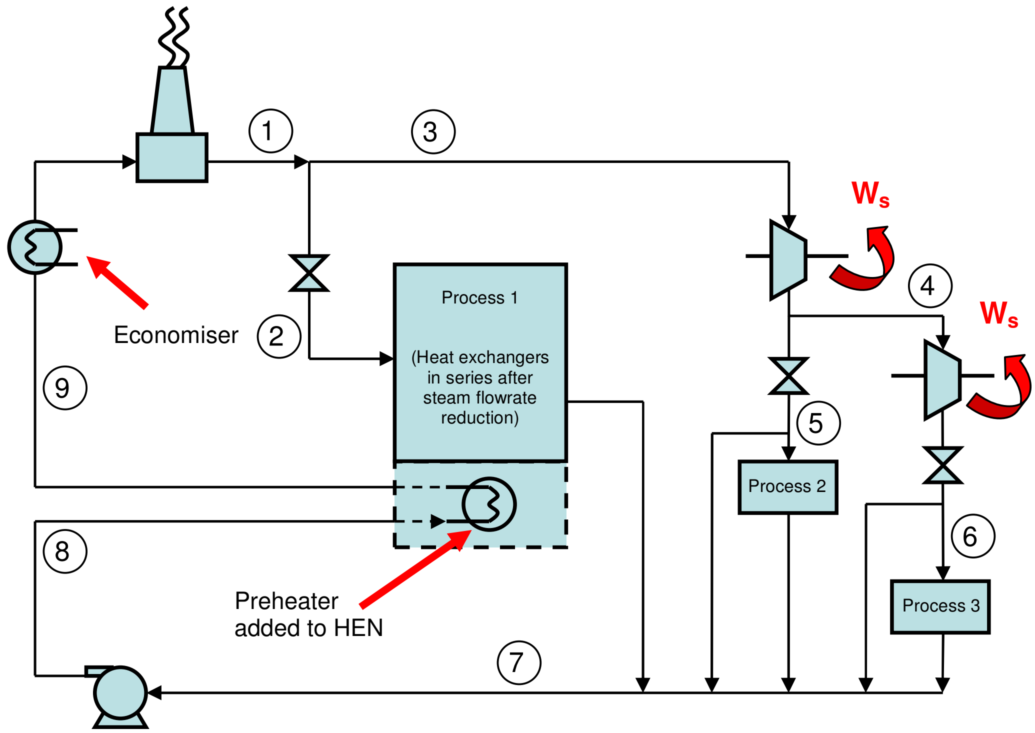

Figure 4.7: Additional preheater added to HEN.

Modelling this situation is slightly more complex than in any of the other scenarios already discussed. The data concerning the HEN, namely the limiting temperature information and duty, were all fixed. The addition of a variable heat exchanger complicates the HEN model slightly. Thus the constraints shown first will be those that are added to the HEN model, and then the constraints concerning the boiler efficiency will be included.

The HEN under investigation is of the single steam pressure level and as such the HEN model presented in Sections 4.1.1 and 4.2.1 will be augmented. Constraints (4.1) to (4.22) will be used as defined already. If it is taken that there are $|i|$ heat exchangers in the HEN the set of all heat exchangers must be expanded by 1. The additional heat exchanger will be designated as *i** to identify it in the formulation. In the model, the *i** heat exchanger behaves as all the others. It is able to receive steam and condensate and transfers condensate to other heat exchangers. The limiting temperatures and the duty of the *i** heat exchanger will, however, be variables in the formulation as these can change depending on the arrangement of the HEN.

The inlet stream to the *i** heat exchanger is stream 8 in Figure 4.7. By referring to Section 4.2, the stream entering the preheater had a designated temperature of T_{pump}. The outlet of the *i** heat exchanger is stream 9 and once again with reference to Section 4.2, this stream temperature is T_{boil}. The limiting temperatures for the *i** heat exchanger are thus T_{pump} and T_{boil}, with the addition of the global ΔT_{min}. Constraints (4.75) and (4.76) show these limits. Constraint (4.77) shows the duty of the * heat exchanger.

$$Tin_{i*}^{L} = T_{pump} + \Delta T_{\min} \qquad \forall i^* \in I \tag{4.75}$$

$$Tout_{i*}^{L} = T_{boil} + \Delta T_{\min} \qquad \forall i^* \in I \tag{4.76}$$

$$\begin{aligned} Q_{i*} = SS_{i*}\lambda + \sum_{j \in I}\left(c_p SL_{j,i*} T_{sat}\right) + \sum_{j \in I}\left(c_p L_{j,i*} Tout_j\right) \\ -\left(c_p Fout_{i*}\left(T_{boil} + \Delta T_{\min}\right)\right) \qquad \forall i^* \in I \end{aligned} \tag{4.77}$$

Since the boiler efficiency is fixed in this formulation, only the constraints in Case 1 of Section 4.2 will be used. In Section 4.2 the preheater uses sensible heat to increase T_{pump} to T_{boil}. In this case the duty of the i^* preheater is used. As such Constraint (4.78) is used to calculate T_{boil}.

$$T_{boil} = T_{pump} + \frac{Q_*}{c_P\left(FS + slack^+ + F_{turb}\right)} \tag{4.78}$$

The remaining part of the formulation is the same as that in Section 4.2.1. Thus Constraints (4.54) to (4.58) complete the model, with Constraint (4.78) replacing Constraint (4.56).

This formulation contains many nonlinear terms. Some are products of continuous variables which are treated with the technique of Quesada and Grossmann (1994). The others are products of continuous variables and binary variables and as such Glover (1975) transformations are made. An explanation for both of these techniques appears in Appendix B.

Since the additional heat exchanger uses steam from the boiler to preheat the feed this improvement is unlikely to be as appealing as simply using sensible heat. The model derived in this section is simply for a steam system that has not included any improvements mentioned thus far. There is no reason that a steam system could not include a preheater utilising sensible heat as well as a preheater inside the HEN as described in this section. Figure 4.8 shows the steam system incorporating this dual preheater concept.

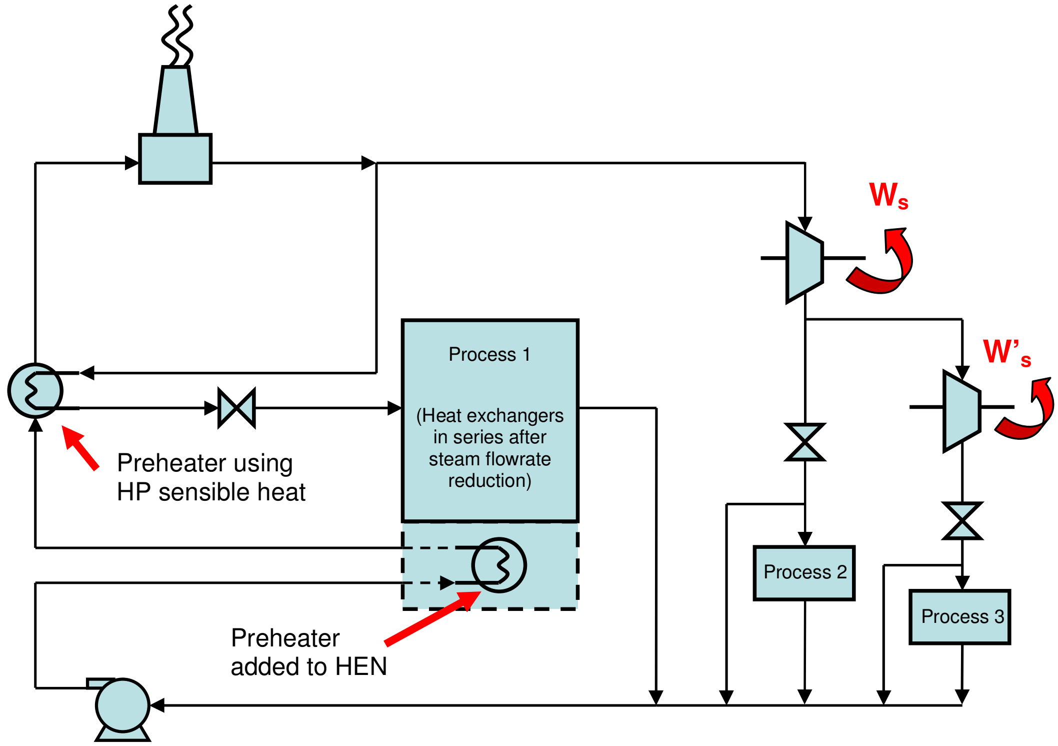

Figure 4.8: Preheater added to HEN with preheater using sensible heat.

4.3.2 Multiple steam pressure levels

The following section is simply an extension of the previous section with the inclusion of multiple pressure levels. Figure 4.9 shows the proposed steam system. Once again the condensate tank and condenser have been omitted, and the return stream to the boiler is stream 9.

As before the set of heat exchangers *i* has to be extended by 1. i^* will be used to denote the extra heat exchanger in the formulation. The inlet to the i^* heat exchanger, stream 8, is at temperature T_{proc}. The outlet, stream 9, is at temperature T_{boil}. The utility limiting temperatures for the i^* heat exchanger are thus T_{proc} and T_{boil} along with the global ΔT_{min}. Constraints (4.79) and (4.80) represent the limiting temperatures for the i^* heat exchanger and Constraints (4.81) to (4.82) are the steam duty, condensate duty and total duty for this heat exchanger respectively.

$$Tin_{i*}^{L} = T_{pump} + \Delta T_{\min} \qquad \forall i^* \in I \tag{4.79}$$

$$Tout_{i*}^{L} = T_{boil} + \Delta T_{\min} \qquad \forall i^* \in I \tag{4.80}$$

$$Q_{i*,l}^{S} = SS_{i*,l}\lambda_l \qquad \forall i^* \in I \tag{4.81}$$

$$Q_{i*}^{L} = \sum_{j \in I, l \in L}\left(c_P SL_{j,i*,l} T_{sat,l}\right) + \sum_{j \in I}\left(c_P L_{j,i*} Tout_j\right) - \left(c_P Fout_{i*}\left(T_{boil} + \Delta T_{\min}\right)\right) \qquad \forall i^* \in I \tag{4.82}$$

$$Q_{i*} = \sum_{l \in L} Q_{i*,l}^{S} + Q_{i*}^{L} \qquad \forall i^* \in I \tag{4.83}$$

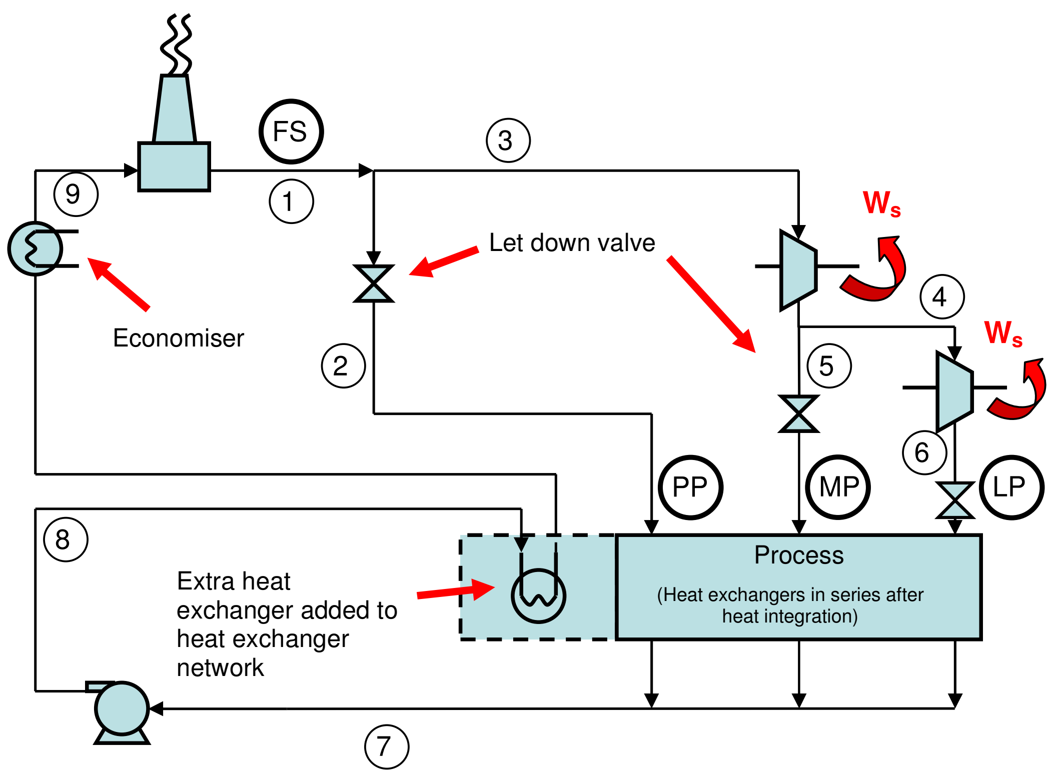

Figure 4.9: Additional HEN heat exchanger in multiple pressure level system.

The boiler efficiency is fixed for this formulation, thus the appropriate elements in Section 4.2.2 will be used. The calculation of T_{boil} in

Constraint (4.68) is changed since sensible heat is no longer used to preheat the boiler feed. Constraint (4.84) now shows the calculation of T_{boil}.

$$T_{boil} = T_{proc} + \frac{Q_{i^*}}{c_P\left(FS + slack^+\right)} \tag{4.84}$$

The remaining elements of the formulation are from Section 4.2.2. These are Constraints (4.65) to (4.68) with Constraint (4.84) replacing Constraint (4.66). The nonlinear terms are dealt with using the linearisation techniques of Quesada and Grossmann, (1994) and Glover, (1975). These techniques are explained in Appendix B.

The sensible heat preheater from Section 4.2 and the dedicated preheater from Section 4.3 could be incorporated into the same steam system which is demonstrated in Figure 4.10. This system could be incorporated if additional energy was required to maintain, or even to potentially increase the boiler efficiency.

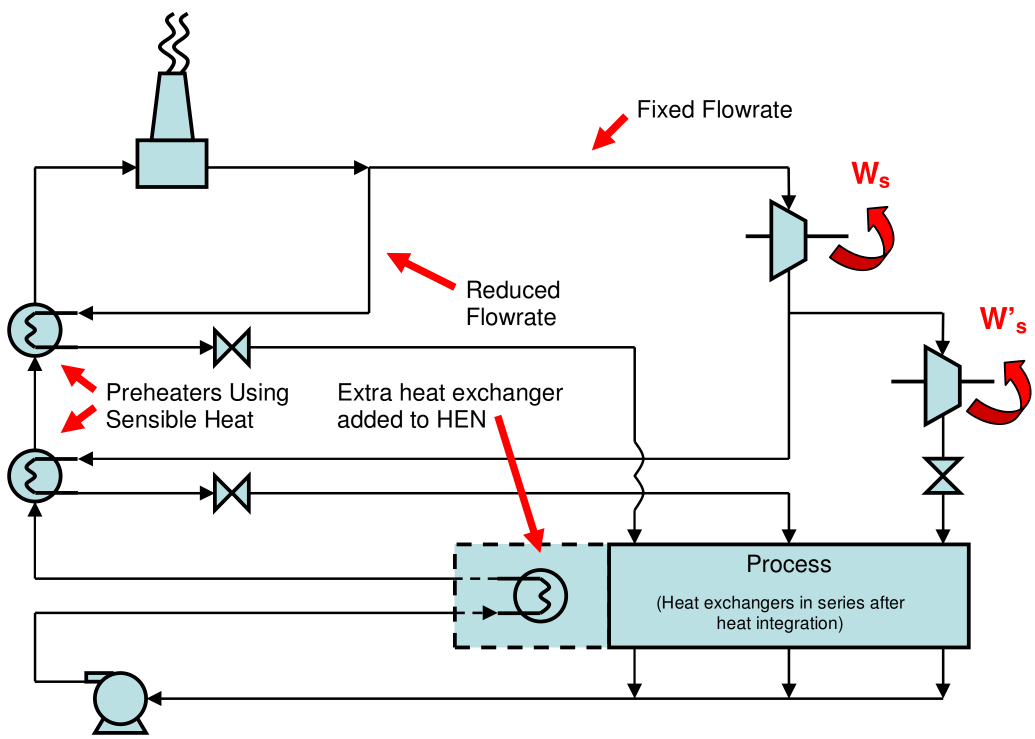

Figure 4.10: Combination of different preheater concepts.

4.4 Pressure Drop Considerations

Another major concern for recycle and reuse in HENs is the aspect of increased pressure drop. With recycling, hot condensate, utility streams have to pass through multiple heat exchangers which greatly increases the pressure drop of the system. Kim and Smith (2003) identified the need to minimise pressure drop in such systems from their work in cooling systems. Following a similar approach pressure drop is minimised for the HEN in steam systems presented in this dissertation. Coetzee and Majozi (2008) found that multiple networks could be found for the same minimum flowrate due to the linear nature of the network design model. Thus it would seem prudent to choose amongst these networks for the one that best suits another objective such as pressure drop.

4.4.1 Pressure as an intensive property

It has been widely established that pressure drop in HENs is not only dependent on stream variables such as flowrate, but also on the network layout. To account for the network layout Kim and Smith (2003) utilise the concept of longest or Critical Path Algorithms (CPA) which are prevalent in mathematical programming. The total pressure drop of the network is essentially represented by the largest pressure drop of a connection of streams. This critical path should then be minimised to minimise the total pressure drop of the system. Kim and Smith (2003) used a node superstructure as a framework to establish the critical path model. The nodes represent mixers that combine streams before heat exchangers as well as splitters that redirect streams after heat exchangers. Each of these nodes has a pressure associated with it. Pressure is then lost between nodes, for example in the heat exchanger between mixers and splitters or in the pipes between the mixers and splitters of different heat exchangers. The mixers are linked to a source node and the splitters to a sink node. The source node represents the maximum pressure of the system, usually the outlet pressure of the utility source, i.e. the cooling tower or steam boiler. The sink node represents the minimum pressure of the system, i.e. the stream returning to the utility source. The objective of the model is then to find the maximum pressure drop through the network and then minimise this pressure drop using mathematical programming.

Kim and Smith (2003) developed a superstructure with which to model the pressure drop for the network. Figure 4.11 shows this superstructure. From the figure the source and sink nodes for each heat exchanger can be seen. The superstructure greatly resembles that used for mass and energy balances in the work on heating systems by Coetzee and Majozi (2008).

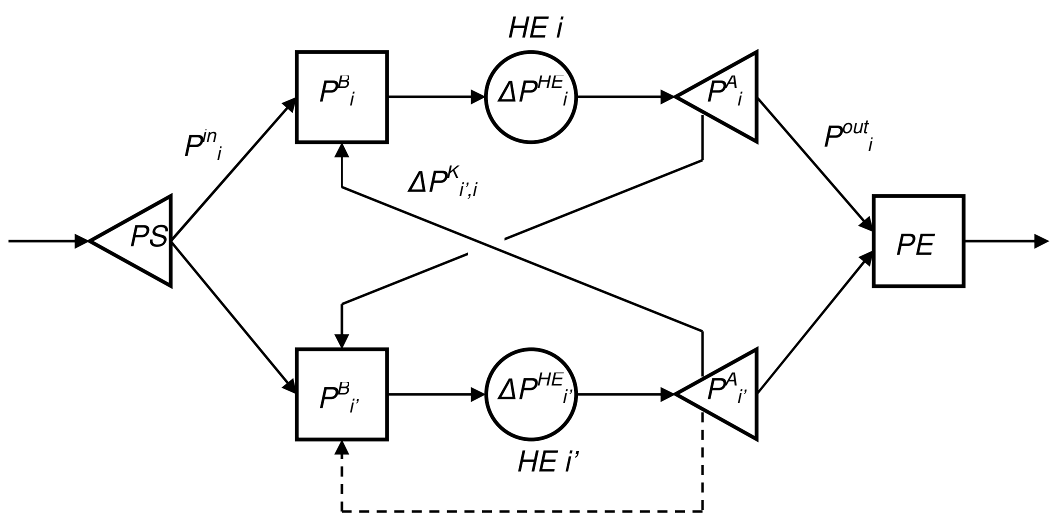

Figure 4.11: Node superstructure used by Kim and Smith (2003).

This superstructure accurately caters for cooling systems, where it is assumed the cooling water stays in the liquid phase. A problem arises in adapting this superstructure to steam systems where steam condenses and condensate is recycled and reused. The pressure drop for condensers is different to that of ordinary heat exchangers where the streams remain in the same phase. Constraint (2.12) is therefore used for the pressure drop for the initial part of the network. Since the remaining pressure drop in the pipes and heat exchangers is dependent on the volumetric flow through them, it is essential that the network design model already established defines streams in great detail. Thus pressure drop for individual units can easily be calculated. The node superstructure does however have to be changed. A new superstructure that accommodates the condensers is shown in Figure 4.12.

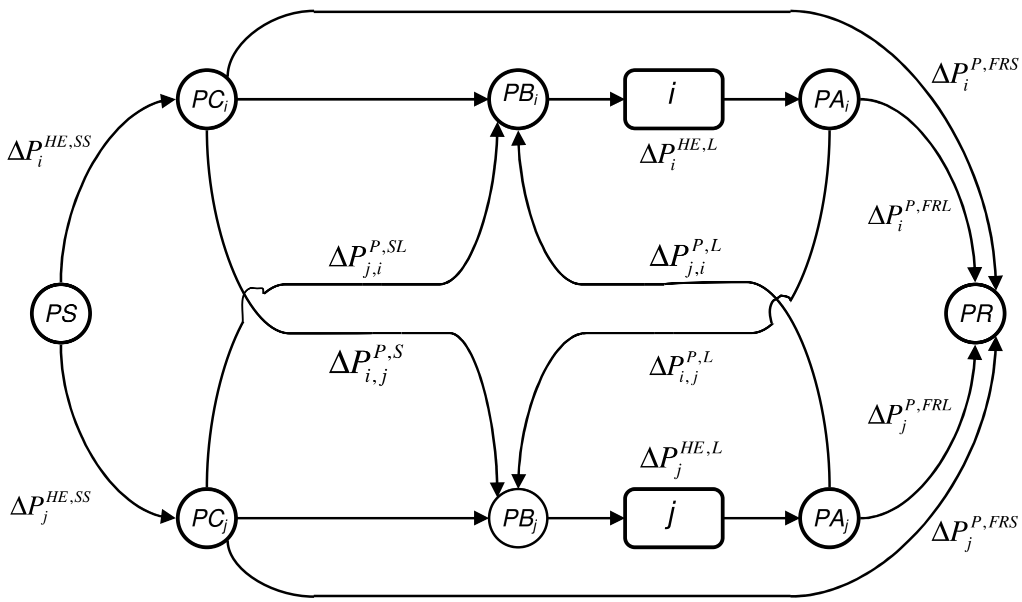

Figure 4.12: New node superstructure that accommodates phase change.

In the figure it can be seen that the condensers are connected to the main source node at source pressure P_S. Since it is assumed that steam does not lose pressure in pipes, $\Delta P^{HE,SS}$ is only as a result of the pressure drop in the condensers. The distributing or splitting node after the condensers then has pressure P_C. This node is then connected to each condensate heat exchangers source node or mixer P_B, as well as the final sink node for return to the boiler P_R. Since saturated condensate exits the condensers, the pressure drop in the pipes between the P_C and P_B is a function of the saturated condensate flowrate and the pressure drop is designated $\Delta P^{P,SL}$. It must also be noted that condensate can be reused by the same process stream, and as such the $\Delta P_{i,i}^{P,SL}$ and $\Delta P_{j,j}^{P,SL}$ terms also exist. The direct return to the boiler is accomplished by the saturated return flowrate *FRS*, thus this pressure drop is designated $\Delta P^{P,FRS}$.

The mixer node for each heat exchanger can also receive condensate from the splitter nodes of other heat exchangers. This occurs in the form of subcooled liquid, designated *L*, with associated pressure drop $\Delta P^{P,L}$. The two condensate streams then combine and pass through the heat exchanger. The pressure drop for each heat exchanger is therefore a function of the sum of saturated and subcooled condensate entering it. This is shown as $\Delta P^{HE,L}$.

Finally each splitter node has the pressure P_A. The return stream to the boiler, *FRL*, then proceeds to the return mixer with pressure P_R. The pressure drop in the pipes of this return stream is thus a function of this flowrate and is subsequently labelled $\Delta P^{P,FRL}$.

The method used by Kim and Smith (2003) based on the CPA to determine the maximum pressure drop for the system is in the form of a difference in pressure between nodes and the pressure drop between the nodes. This is represented in a manner similar to Constraint (4.85) below.

$$P_A - P_B \geq \Delta P_{A,B} \tag{4.85}$$

In Constraint (4.85) it is understood that fluid flows from node *A* to node *B*. The pressure difference between the nodes is essentially a result of the piping pressure loss. Since the mixing nodes before a heat exchanger may receive fluid from multiple sources, this constraint will occur several times with different source splitter nodes and consequently different pressure drop values. The inequality in Constraint (4.85) ensures that the *PB* assumes the lowest pressure value that satisfies all of the constraints. In this way the node pressure is always at this low value when the model is solved.

This constraint, though simple, is very elegant and effective at finding the critical path, or in this case critical pressure drop for a given network. However, since it will occur in the network design stage a means of eliminating those nodes that do not exist must be made. Kim and Smith (2003) turned to binary variables to achieve this. They established a connection existence binary variable for each connection in the network. This binary variable takes the value of 1 if the connection exists and 0 if it does not. An additional term is then added to Constraint (4.85) to render it redundant for the cases where no connection exists as shown in Constraint (4.86). This is done by adding a large pressure term such that it is satisfied for these cases. The large pressure is represented by the term *BP* in Constraint (4.86).

$$P_A - P_B + BP\left(1 - y_{A,B}\right) \geq \Delta P_{A,B} \tag{4.86}$$

The binary variables are created using the actual flow variables that are part of the network design model and that are used to calculate the various pressure drops in the model. The only nodes affected by this phenomenon are those that could possibly receive multiple inputs. These are the P_B and P_R nodes, the mixers before the condensate heat exchangers and the boiler return respectively. Only four flow variables are associated with these and they are *SL*, *L*, *FRL* and *FRS*. Each of the binary variables requires two constraints as well as known upper and lower bounds for the flowrates. Constraints (4.87) and (4.88) are used to demonstrate this for the variable *SL*.

$$SL - SL^{U}\left(y^{SL}\right) \leq 0 \quad (4.87)$$

$$SL - SL^{L}\left(y^{SL}\right) \geq 0 \quad (4.88)$$

The binary variable will assume the correct value if the appropriate connection is active. In Constraints (4.87) and (4.88) SL^U and SL^L are the upper and lower limits respectively. Constraints (4.89) to (4.94) represent the longest path constraints that exist according to the superstructure in Figure 4.12.

$$PS - PC_i = \Delta P_i^{HE,SS} \qquad \forall i \in I \quad (4.89)$$

$$PC_j - PB_i - BP\left(1 - y_{j,i}^{SL}\right) = \Delta P_{j,i}^{P,SL} \qquad \forall i, j \in I \quad (4.90)$$

$$PA_j - PB_i - BP\left(1 - y_{j,i}^{L}\right) = \Delta P_{j,i}^{P,L} \qquad \forall i, j \in I \quad (4.91)$$

$$PA_i - PR - BP\left(1 - y_i^{FRL}\right) = \Delta P_i^{P,FRL} \qquad \forall i \in I \quad (4.92)$$

$$PC_i - PR - BP\left(1 - y_i^{FRS}\right) = \Delta P_i^{P,FRS} \qquad \forall i \in I \quad (4.93)$$

$$P_{B,i} - P_{A,i} = \Delta P_i^{HE,L} \qquad \forall i \in I \quad (4.94)$$

The constraints shown above, as well as those showing the various pressure drops are combined to form the pressure drop model. All that remains is to state the objective function. Since the CPA finds the maximum pressure drop through the network, the objective function then simply minimises this pressure drop. Constraint (4.95) is thus the objective function.

$$MinZ = P_S - P_R \quad (4.95)$$

A number of heuristic methods can be used to remove some of the constraints. Topological restrictions are common, but a constraint from the original network design model is also relevant for the pressure drop model. Local recycle cannot occur for thermodynamic reasons, but also due to the pressure gradient since for a given heat exchanger $P_{A,i}$ is less than or equal to $P_{B,i}$ according to Constraint (4.94).

4.4.2 Pressure drop constraints in terms of mass flowrates

The HEN model represents steam and condensate flowrates in terms of mass. To integrate the pressure drop constraints it will be appropriate to also represent them in terms of mass flowrate. In this way the HEN model can be used to rearrange the network so as to find the minimum network pressure drop using the pressure drop constraints. As such Constraints (2.17) to (2.22) will be linked to the appropriate mass flowrate variables from this model in Section 4.1. Constraint (4.96) shows the pressure drop through a condenser associated with stream *i*.

$$\Delta P_i^{HE,SS} = 0.5\left(N_{t1}^* SS_i^{1.8} + N_{t2}^* SS_i^2\right) \qquad \forall i \in I \qquad (4.96)$$

Constraint (4.97) shows the pressure drop through a heat exchanger utilising condensate. This constraint must utilise all of the condensate flowing through a particular heat exchanger. The constants N_{t1}^* and N_{t2}^* were described in Section 2.3.

$$\Delta P_i^{HE,L} = N_{t1}^*\left(\sum_{j\in I} SL_{j,i} + \sum_{j\in I} L_{j,i}\right)^{1.8} + N_{t2}^*\left(\sum_{j\in I} SL_{j,i} + \sum_{j\in I} L_{j,i}\right)^2 \quad \forall i \in I \qquad (4.97)$$

The piping pressure drops are for the various piping connections. These include the steam flow from the boiler to condensers, the saturated condensate from the condensers to heat exchangers where no phase change occurs, the saturated condensate from condensers to the boiler, the recycle streams between heat exchangers and the subcooled condensate return flow to the boiler. The piping pressure drop relations for these flowrates will be shown below. The pressure drop for the steam flowing from the boiler to the condensers is considered negligible due to the low density of steam and the density dependence of Constraint (2.17). The pressure drop from saturated condensate flow from heat exchanger *j* to *i* is shown in Constraint (4.98). Similarly the pressure drop for subcooled condensate flow from heat exchanger *j* to *I* is shown in Constraint (4.99).

$$\Delta P_{j,i}^{P,SL} = N_P^{NW*} \frac{1}{\left(SL_{j,i}\right)^{0.36}} \qquad \forall i \in I \tag{4.98}$$

$$\Delta P_{j,i}^{P,L} = N_P^{NW*} \frac{1}{\left(L_{j,i}\right)^{0.36}} \qquad \forall i \in I \tag{4.99}$$

Constraint (4.100) shows the pressure drop through the pipe connecting the condenser outlet to the boiler return stream, while Constraint (4.101) shows the pressure drop through the pipe connecting the condensate heat exchanger outlet and the boiler return stream.

$$\Delta P_i^{P,FRS} = N_P^{NW*} \frac{1}{\left(FRS_i\right)^{0.36}} \qquad \forall i \in I \tag{4.100}$$

$$\Delta P_i^{P,FRL} = N_P^{NW*} \frac{1}{\left(FRL_i\right)^{0.36}} \qquad \forall i \in I \tag{4.101}$$

Constraints (4.96) to (4.101) show the various pressure drop correlations for the heat exchanger network. These will be incorporated into the overall network pressure drop scheme discussed below.

4.4.3 Minimum network pressure drop

If a situation arose where heat integration caused an increased pressure drop that was above the set maximum for the network it is likely that additional fluid movers would be required for the system (Kim and Smith, 2003). The pressure drop model can be restructured in such a way that the minimum pressure drop for the system can be found. This network must satisfy the energy demands of the process streams however the minimum steam flowrate will be relaxed in order to reduce the pressure drop.

In this section the minimum mass flowrate for the system was found as before. This minimum is then used as a basis and a slack variable is added to it in the overall inlet and outlet mass balances from Section 4.1. Since single pressure levels are used to demonstrate this, Constraint (4.1) becomes Constraint (4.102) and Constraint (4.4) becomes Constraint (4.103). Given that the overall mass balance is unaffected, these are the only constraints that are changed. The number of heat exchanger splits, *n*, can also be made a variable as this appears in the HEN model. In this way the model will minimise pressure drop by compromising the minimum flowrate by an amount equal to the slack variable and adjust the number of heat exchanger splits by altering *n*. As with all the HEN based models the heating demands of the network will be satisfied. The objective function remains the same and is shown in Constraint (4.95)

$$FS = \sum_{i \in I} SS_i + slack \tag{4.102}$$

$$FR = \sum_{i \in I} FR_i + slack \tag{4.103}$$

This method could give a designer insight into whether an additional fluid mover will likely be necessary or not.

4.4.4 Solution strategy

The network design portion of the model has already been linearised into an MILP using the maximum outlet conditions specified by Savelski and Bagajewicz (2000). The pressure drop correlations are highly nonlinear however, which creates an MINLP that is difficult to solve. Kim and Smith (2003) examined the individual constraints and determined that the pressure drop for heat exchangers was fairly linear in the design flowrate region. The piping pressure drop was more nonlinear, however they proposed a piecewise linear approximation.

The same approach is used for the steam system. The results from the case study shown previously are used to create the appropriate limits and variables to plot pressure drop against flowrate for the condensers, condensate heat exchangers and pipes. These are shown in the case study.

4.5 Combined Boiler Efficiency and Pressure Drop

The formulations shown in Sections 4.2.1 and 4.3.1 use sensible heat from superheated steam to maintain the boiler efficiency. The formulation in Section 4.4 shows how the minimum pressure drop for a network can be found for a given mass flowrate. By combining these models the minimum pressure drop can be determined for a system where the boiler efficiency has been maintained using one of the methods in Section 4.2.1 or 4.3.1. Various options will be discussed and the necessary constraints mentioned as the constraints have already been shown in the relevant sections.

4.5.1 Solution strategy

The mathematical intricacies of the CPA require that it is solved independently from the other models. Thus the boiler efficiency models will be used to find a minimum flowrate. This flowrate will then be used in the pressure drop model as a parameter. The actual boiler efficiency constraints will also have to be included in the pressure drop section to ensure that the efficiency is maintained, even if the network is rearranged in finding the minimum pressure drop. The number of heat exchanger splits can be included as a variable or set manually. Since fewer heat exchanger splits are favourable in terms of cost, the effect of varying this will also be examined. The pressure drop models are nonlinear as such these will be solved as a MINLP, whereas the flowrate minimisation models remain the combined MILP and MINLP.

4.5.2 Sensible heat preheater

Section 4.2.1 explores the option of using sensible heat from the superheated high pressure steam to preheat the boiler feed such that the boiler efficiency is maintained. Consequently these constraints will be used to find the minimum steam flowrate which is used as a parameter in determining the minimum pressure drop for the associated network. Constraints (4.1) to (4.22) will once again be used to find the minimum steam flowrate for the network. As before, the minimum steam flowrate will be relaxed using a slack variable. Since the boiler efficiency is the focus, Case 1, where the boiler efficiency is maintained with a possible compromise in the minimum flowrate, will be explored. Thus Constraints (4.54) to (4.58) will be used to maintain the efficiency while minimising the relaxed minimum steam flowrate. The total flowrate will then be used as a constant for the pressure drop section which will consist of Constraints (4.1) to (4.22) for the network design, Constraints (4.54) to (4.58) for the efficiency and Constraints (4.89) to (4.101) for the pressure drop.

4.5.3 Dedicated preheater

Section 4.3.1 explores the possibility of maintaining the boiler efficiency by adding a heat exchanger dedicated to reheating the boiler return stream. Using the solution strategy from this section the network design is once again done with Constraints (4.1) to (4.22). The boiler efficiency Constraints are (4.54) to (4.58) and the additional heat exchanger Constraints are (4.75) to (4.78), where Constraint (4.78) replaces Constraint (4.56). This model finds the minimum steam flowrate such that the extra preheater can maintain the boiler efficiency. This flowrate is then set for the pressure drop section which consists of Constraints (4.1) to (4.22) for the network design, Constraints (4.54) to (4.58) for the efficiency, Constraints (4.75) to (4.78) for the extra preheater where Constraint (4.78) replaces Constraint (4.56) and Constraints (4.89) to (4.101) for the pressure drop.

4.6 References

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5. Case Study

Two steam system setups exist, the single steam pressure level system and the multiple steam pressure level system. Within these two problems several means of maintaining boiler efficiency are presented. The first is using dedicated preheaters that use sensible heat to preheat the boiler return stream. The second is a dedicated preheater that uses process steam to preheat the boiler return stream. As such the same case study will be used for these cases so that the various methods can be compared.

The models that are applied to the case study will be used in the following order. The single pressure level models will appear first, followed by the multiple steam pressure level models. The pressure drop section will be presented last. For simplicity Table 5.1 has been included to show which sections the boiler efficiency models are described in, as well as the name they are referred too in this section. For example the HEN model for the single steam pressure level system is presented in Section 4.1.1 and called Model A in this section. All of the boiler efficiency models utilise the HEN model to design the networks.

Table 5.1: Sections where formulations are found.

	HEN model	Sensible heat preheater	Dedicated preheater
Single pressure level	4.1.1 (A)	4.2.1 (B)	4.3.1 (C)
Multiple pressure levels	4.1.2 (D)	4.2.2 (E)	4.3.2 (F)

All of the mathematical programming was completed in GAMS. The solvers utilised were Cplex for the linear models and Dicopt (which utilises Cplex as the MIP solver and Conopt as the NLP solver) for the nonlinear models.

The case study used for the investigation will be that of Coetzee and Majozi (2008). Table 5.2 shows the limiting temperature data and duty for the utility streams in the HEN. The relevant steam information can be seen in Table 2.1. The additional information about the turbine exhaust will be given in Table 5.3. Given all of the above information it is then possible to solve the formulations discussed above.

Table 5.2: Hot utility stream data.

Stream	T^L Target (°C)	T^L Supply (°C)	Duty (kW)	Flowrate (kg/s)
1	35	55	135	0.07
2	35	55	320	0.17
3	219	225	3 620	1.97
4	89	195	12 980	7.08
5	217	217	1 980	1.08
6	54	80	635	0.34
7	54	80	330	0.18
Total			20 000	10.98

Table 5.3: Relevant data for turbine portion of steam system.

Turbine condensate mass flowrate	7.27 kg/s
Turbine condensate return temperature	113 °C

The original steam system is a HEN from a petrochemicals company. This HEN is designed in parallel as shown in Figure 5.1, along with the relevant steam flowrates. This information will be used as the basis for comparison in all of the formulations. All flowrates in the figures are in kg/s.

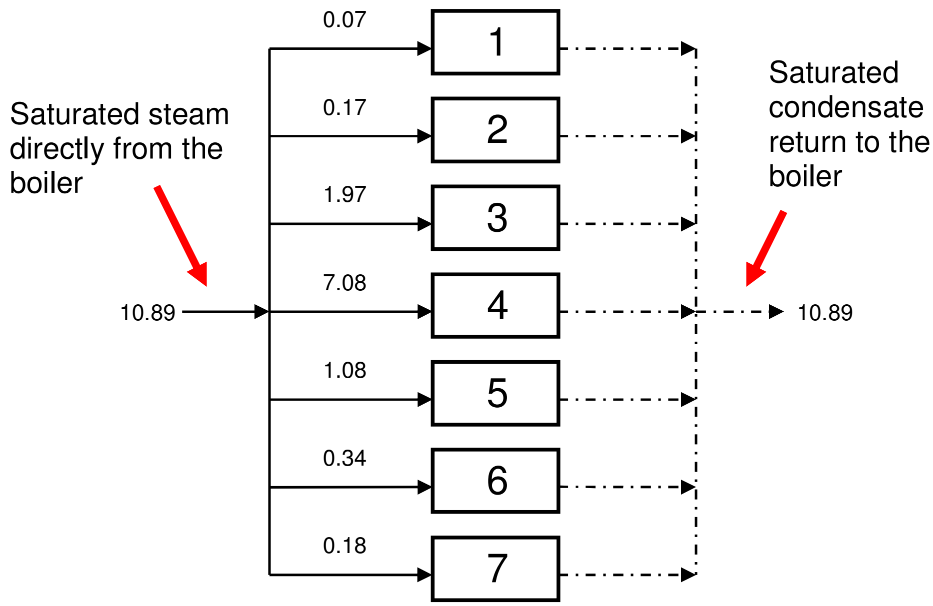


Figure 5.1: Parallel configuration of the original steam system.

5.1 Single Pressure Level Model

Firstly the Model A will be applied to the case study to find the minimum flowrate to the HEN. The effects on the outlet temperature of the HEN and consequently the boiler efficiency will then be examined. Then the sensible heat preheater formulation will be implemented from Section 4.2.1 and the results compared to the original. Then the extra HEN preheater formulation will be implemented from Section 4.3.1 and the results compared to the original as well as those of the sensible heat preheater.

5.1.1 Steam reduction for the HEN

Implementing the single pressure level HEN model on the case study revealed that the minimum steam flowrate for the HEN is 7.68 kg/s. This is a 29.6% saving in the steam flowrate as compared to the parallel design. The HEN arrangement is shown in Figure 5.2. Clearly from this figure it can be seen that the process stream associated with heat exchanger 4 requires heating from both saturated steam and saturated condensate. Thus the HEN requires 1 extra heat exchanger. A split heat exchanger is indicated by the main heat exchanger number being accompanied by a subscript. Heat exchanger 4 in Figure 5.2 is made up of 4_1 and 4_2, representing two separate physical heat exchangers.

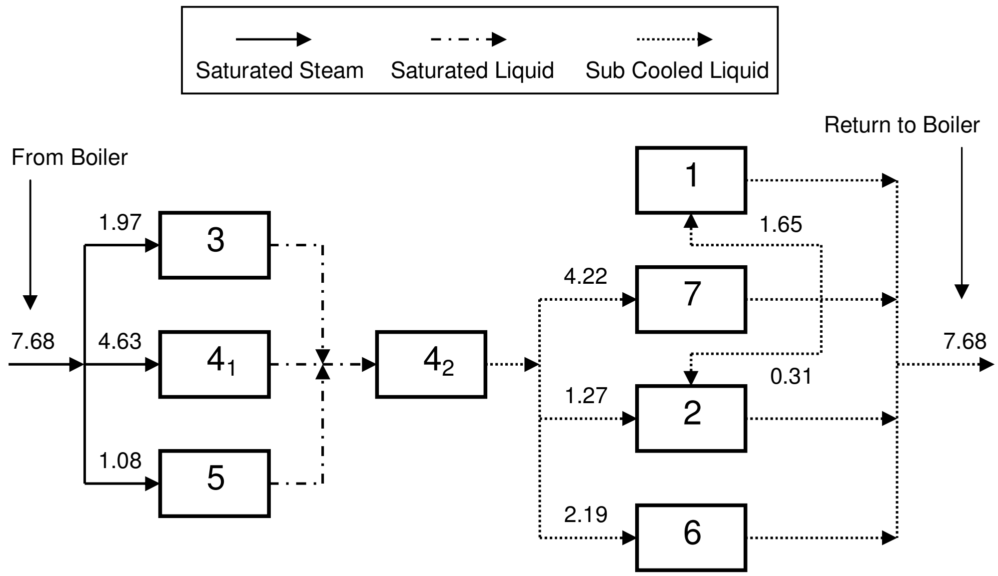


Figure 5.2: HEN after steam reduction.

The original steam system outlet was at the saturation point. This stream then combines with the turbine background process condensate. Since the temperature of this combined stream is still quite high the system had to incorporate a condensate tank and/or condenser to prevent cavitation. As seen in Table 2.1 a standard pumping temperature is approximately 116°C. The saturated temperature of the high pressure (HP) steam was taken from Table 2.1 as 254°C. The combined return flowrate is the sum of the stream from the turbines, which totals 7.27 kg/s, and that of the parallel HEN, which is equivalent to 10.98 kg/s. Therefore the overall flowrate returned to the boiler is 18.17 kg/s. No information on the boiler capacity was given therefore the load was assumed to be 90%, implying the maximum load to be 20.19 kg/s. The regression parameters, *a* and *b*, used by Shang and Kokossis (2004) were 0.0126 and 0.2156 respectively. Since the original report by Pattison and Sharma (1980) was not found (in almost a year of searching) these values reported by Shang and Kokossis (2004) were used throughout the case study. The change in enthalpy from saturated condensate to superheated steam was calculated using steam tables as 2110 kJ/kg. This value will therefore be used as *q*. The specific heat capacity was assumed constant and taken as 4.3 kJ/kg.K. Using these parameters the original efficiency of the boiler was calculated as 63.5% using Constraint (2.8).

After steam reduction the flowrate from the HEN was reduced to 7.68 kg/s. Thus the combined return stream flowrate was reduced to 14.96 kg/s. As a result of the utilisation of heat from the steam condensate the outlet temperature of the HEN was reduced from saturation temperature to 46°C. Combining this with the turbine stream made a return temperature of 78.6°C. These results conform to those of Coetzee and Majozi (2008). As a result of this reduction in temperature the need for a condensate tank and/or condenser was eliminated. The resulting flowrate and temperature reduction adversely affected the boiler efficiency. The new boiler efficiency was calculated as 59.8%. This is a 5.9% reduction in the boiler efficiency.

5.1.2 Sensible heat preheater to maintain boiler efficiency

According to Constraint 2.8, to maintain the boiler efficiency with the reduced flowrate the return temperature will have to be increased to 117.6°C. The steam system changes from Model B are therefore implemented. The enthalpy change between superheated high pressure steam and saturated steam at the process pressure was calculated using steam tables and found to be 413.4kJ/kg. The fraction of sensible heat that can be used safely without risking condensation is not, to the knowledge of the authors, discussed in literature either experimentally or theoretically. Thus a maximum value of 80% was used in all cases. The θ value in Constraint (4.54) is thus also taken as a variable in the model, with the upper bound of 0.8.

Case 1 in Section 4.2.1 was implemented first and it was found that the improvements were indeed able to maintain the boiler efficiency. 79.2% of the available sensible heat was needed to raise the return temperature to the required 117.6°C. Since the minimum steam flowrate was not compromised the HEN arrangement shown in Figure 5.2 was also not changed.

Case 2 from Section 4.2.1 revealed that the minimum flowrate could be achieved without compromising the boiler efficiency. Since all of the available sensible heat was not used to maintain the boiler efficiency, the possibility exists to increase the efficiency. By altering the available sensible heat, or θ value, different efficiencies can be calculated. Table 5.4 shows various θ values, their corresponding boiler efficiencies and the increase from the original boiler efficiency.

Table 5.4: Results for varying fractions of sensible heat use.

Fraction	Return Temperature (°C)	New Efficiency	Increase (%)
0.792	117.6	0.6349	0
0.8	118.1	0.6350	negligible
0.85	120.6	0.6380	0.5
0.9	123.1	0.641	1.0
0.95	125.5	0.6430	1.3
1	128.0	0.6460	1.7

In Table 5.4 it can be seen that if 100% of the available sensible heat could be used safely without the risk of condensation the boiler efficiency could be increased by up to 1.7%. As mentioned in Section 4.2.1, this is merely the increase according to Constraint (2.8).

The improvements in the steam system were able to achieve the minimum steam flowrate without compromising the boiler efficiency. If the amount of available sensible heat were to be further restricted, the formulations in Section 4.2.1 may show how they compromise either boiler efficiency or the minimum flowrate so as to maintain the other. For this reason the upper bound on the sensible heat, or θ value was set to 0.3 for the case study.

Case 1 revealed that a compromise in the minimum flowrate was needed to maintain the boiler efficiency. The relaxed solution for the problem revealed a flowrate of 8.25 kg/s, while the exact solution was found to be 8.32 kg/s. This is a 0.64 kg/s increase on the minimum, a fraction of 8.3%. The savings on the parallel configuration was still a substantial 23.6%. Figure 5.3 shows the new HEN designed by the model to correspond to the compromised minimum flowrate.

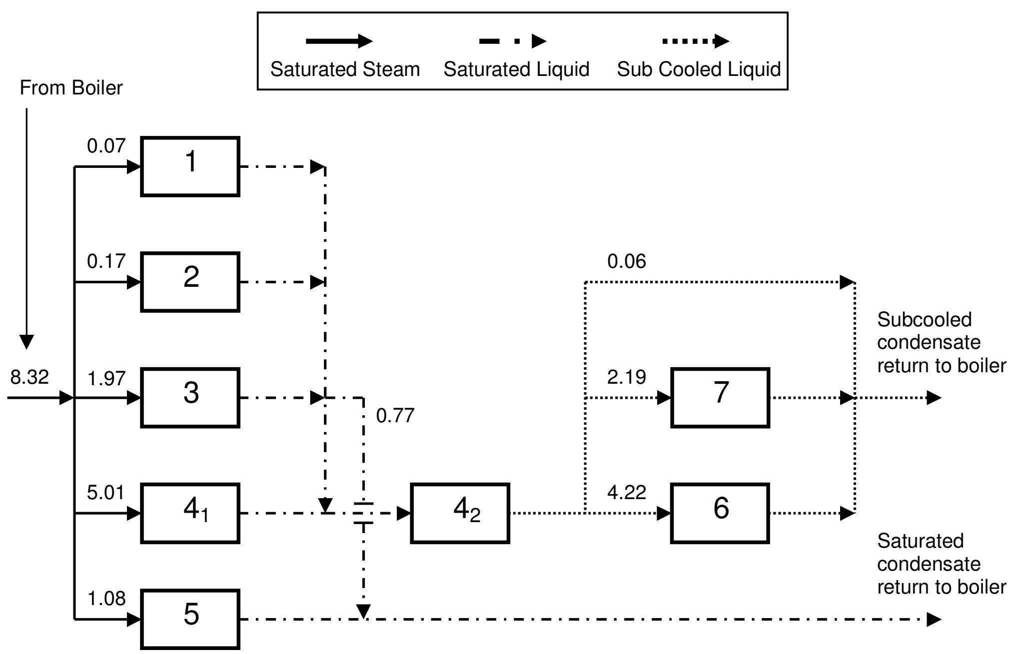


Figure 5.3: New HEN arrangement so as to maintain the boiler efficiency.

It is interesting to note in Figure 5.3 that some of the saturated condensate outlet from heat exchangers 3 and 5 is returned to the boiler directly. This greatly increases the HEN return temperature to 92.3°C. It was also found that since the minimum flowrate was slightly compromised the return temperature necessary to maintain the boiler efficiency was slightly lower at a value of 117.3°C.

As expected, case 2 compromised the boiler efficiency so as to achieve the minimum steam flowrate. The new boiler efficiency was found to be 61.3%, a decrease of 3.5% from the original boiler efficiency. Since the minimum steam flowrate was maintained, the HEN arrangement is the same as that of Figure 5.2.

5.1.3 Dedicated preheater to maintain the boiler efficiency

Section 4.3.1 describes the process of adding an additional heat exchanger to the HEN with the sole purpose of preheating the boiler feed such that the boiler efficiency is maintained. Since this heat exchanger will be using steam from the boiler, it is difficult to calculate what return temperature will maintain the boiler efficiency, as the return flowrate changes. This process could be completed by iteration. However a mathematical optimisation framework is much more elegant.

Using Model C it was found that the boiler efficiency could be maintained using the additional dedicated preheater. The steam flowrate had to be increased to 8.76kg/s. This is 13.9% higher than the minimum flowrate for the network. However it is still a saving of 19.6% on the original steam flowrate for the parallel HEN arrangement. Figure 5.4 shows the new HEN arrangement with the additional heat exchanger being number 8 and distinguished by an asterisk (*).

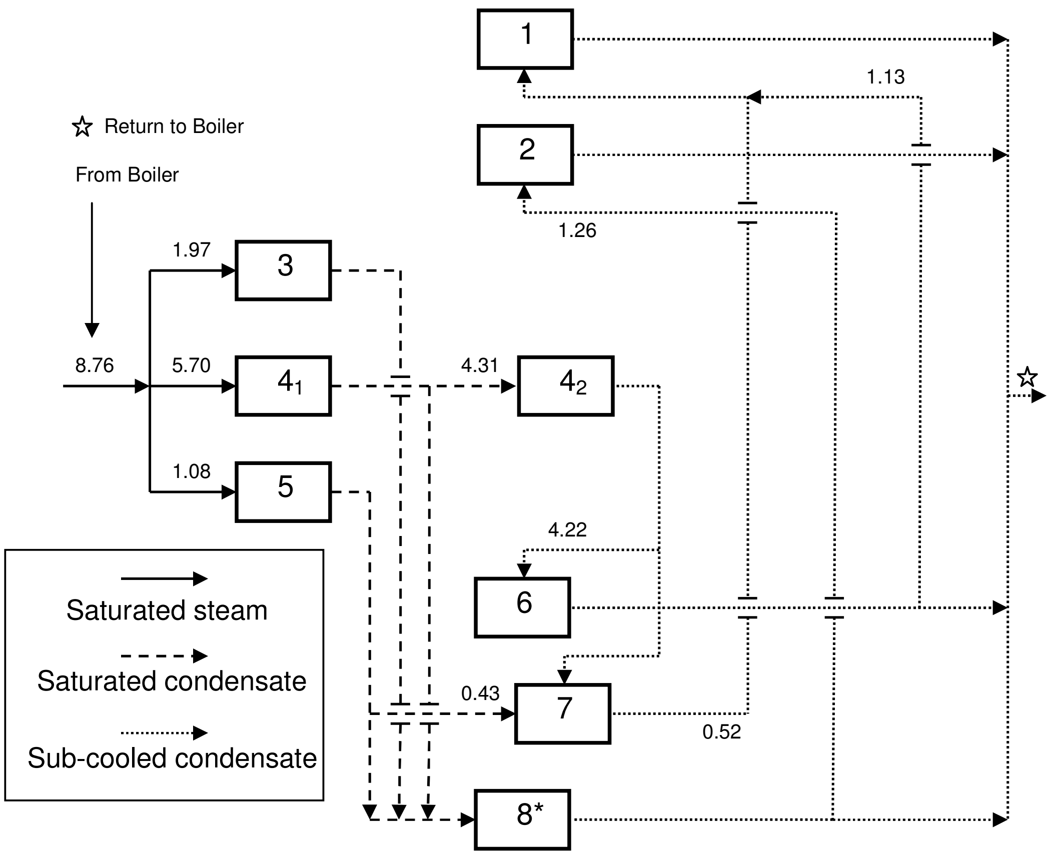


Figure 5.4: HEN with extra heat exchanger.

It was found that the process outlet temperature was 60.4°C. This resulted in a temperature of 84.2°C when combined with the turbine stream. The duty of the additional heat exchanger was calculated as 2256.4kW. The return stream to the boiler thus had a temperature of 117.0°C.

5.2 Multiple Pressure Level Model

The same HEN from the single pressure level case study was used in this investigation. Since the HEN was to receive steam at multiple pressure levels, the steam system layout shown in Figure 2.3 was used, where the turbine exhaust streams are also sent to the HEN. Therefore the return stream to the boiler will consist solely of the HEN outlet stream.

Consequently the original boiler efficiency for the system in this configuration was slightly different. The return temperature was still taken from Table 2.1 as 116°C, as it is the outlet from the condenser/condensate tank. The flowrates from high and medium pressure turbines are fixed since the power ratings on these turbines are fixed. The outlet of the high pressure turbine is shown as the sum of streams 4 and 5 in Figure 2.3. The exhaust from the medium pressure turbine is stream 6. Stream 5 enters the process at medium pressure and stream 6 at low pressure. Table 5.5 shows the flow information and the saturated properties of streams 5 and 6.

Table 5.5: Relevant data for turbine portion of steam system.

Condensate leaving the HP turbine, not proceeding to the LP turbine (stream 5 in Figure 2.3)	2.0 kg/s
Condensate leaving the MP turbine (stream 6 in Figure 2.3)	0.23 kg/s

The total duty for the HEN was divided among the 3 steam pressure levels. Since the flowrate for the medium and low pressure steam was set the flowrate of the process pressure steam, shown by stream 2 in Figure 2.3, could be calculated. The flowrate of process pressure was thus found to be 8.50kg/s. This then leads to a total return flowrate of 10.73kg/s by combining the medium and low pressure steam flowrates. The load of the boiler was taken as 80%. This value was chosen to simply vary the case study from that of the single pressure level case and not for any process constraint. This led to a maximum load of 13.41kg/s. The other variables in the efficiency constraint remained the same as for the single pressure level system. The efficiency calculated by Constraint (2.8) was thus found to be 63.4%.

Model D was applied to this case study so as to find the minimum steam flowrate, and the effects of this flowrate on the boiler efficiency.

Then the improvements from Model E and Model F will be applied to the steam system in an attempt to maintain the boiler efficiency.

5.2.1 Steam reduction for the HEN

Using Model D the minimum flowrate to the HEN at process pressure was found to be 5.68 kg/s. This is a 33.2% reduction from the parallel HEN arrangement. The flowrates from the high and medium pressure turbines were taken as constant, as shown in Table 5.5. Consequently the outlet temperature from the HEN was reduced to 63.0°C. The total return flowrate was thus reduced to 7.91 kg/s. These then reduced the boiler efficiency to 58.2% which is 8.2% lower than the original value. The new HEN arrangement is shown in Figure 5.5 below.

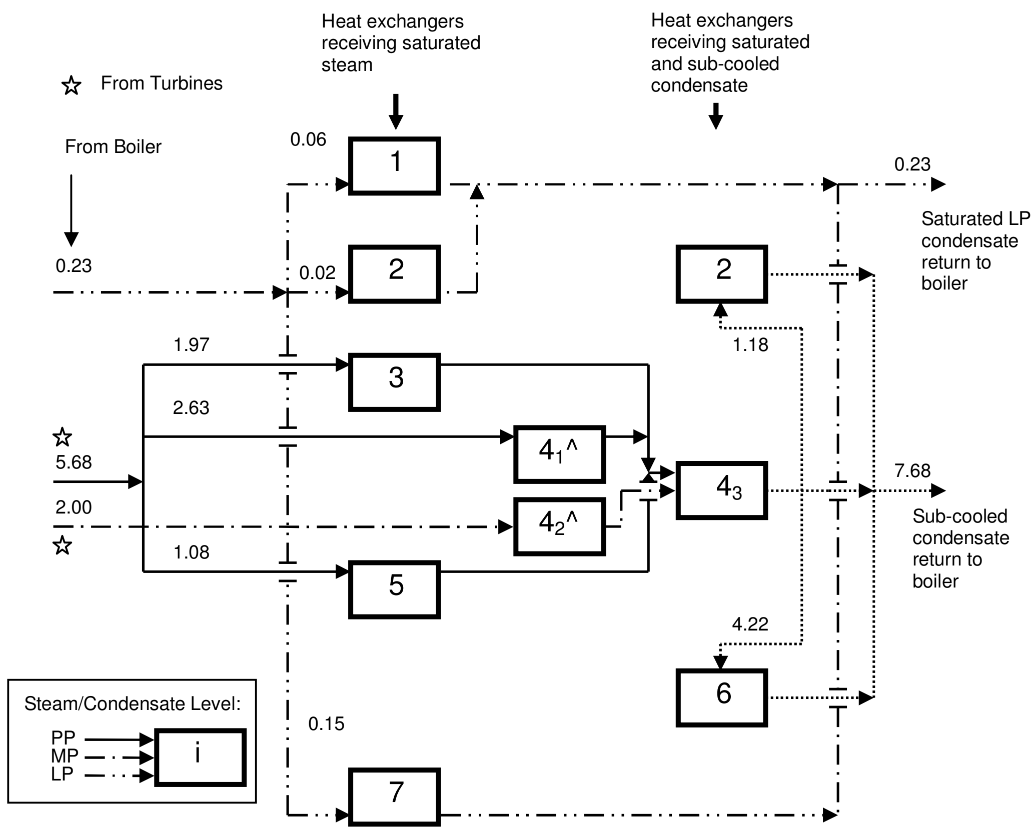


Figure 5.5: HEN arrangement corresponding to minimum steam flowrate.

From the figure it can be seen that two process streams had to be heated by both steam and condensate, and stream 4 was heated by steam at two different pressure levels. This has been distinguished with a (^).

5.2.2 Sensible heat preheaters to maintain boiler efficiency

The improvements in Model E were implemented into the steam system. It was found that the return stream to the boiler had to be heated to 119°C. Once again the amount of sensible heat that could be used in the preheaters was limited to 80%. This applies for θ and Φ in Constraint (4.66).

For case 1 in Section 4.2 it was found that the boiler efficiency could be maintained for the minimum process steam flowrate. This required the preheaters to use 60.1% of the high pressure sensible heat and the maximum 80% of the medium pressure sensible heat. Since the minimum flowrate was not changed the HEN arrangement seen in Figure 5.5 could remain the same.

As before, case 2 revealed that the boiler efficiency did not have to be compromised in order to utilise the minimum process flowrate. Since not all of the sensible energy was needed, the opportunity to increase the boiler efficiency was presented once again. Table 5.6 shows the fractions of sensible heat and the relevant boiler efficiencies that can be achieved.

Table 5.6: Increase in boiler efficiency for different sensible heat fractions.

Θ	Φ	Return Temperature (°C)	New Efficiency	Increase (%)
60.1	0.8	119.0	0.6340	0
0.8	0.8	132.7	0.6480	2.2
0.85	0.85	137.1	0.6530	3.0
0.9	0.9	141.4	0.6580	3.8
0.95	0.95	145.8	0.6620	4.4
1	1	150.1	0.6670	5.2

From the table it can be seen that if all of the sensible heat from the superheated steam could be used, the boiler efficiency could be increased by up to 5.2%. Once again it must be stressed that this improvement is for the efficiency according to Constraint (2.8). Increasing the boiler return temperature should, however, be beneficial to the steam system and as such these changes should be favourable.

Once again the available sensible heat is sufficient to heat the boiler feed so as to maintain the efficiency. By reducing the available sensible heat the formulations from Section 4.2.2 may show how they compromise either boiler efficiency or the minimum flowrate so as to maintain the other. For this purpose the upper limits of both θ and Φ were reduced to 0.3.

Case 1 revealed that the boiler efficiency could indeed be maintained however the minimum steam flowrate was compromised. The process steam flowrate was found to be 6.10kg/s. This is an increase of 7.3% from the minimum process pressure steam flowrate although it is still 28.3% savings on the original parallel HEN arrangement flowrate. The new return stream to the boiler was thus 8.33kg/s and the return temperature had to be increased to 118.4°C. Figure 5.6 shows the new HEN arrangement corresponding to the new flowrate.

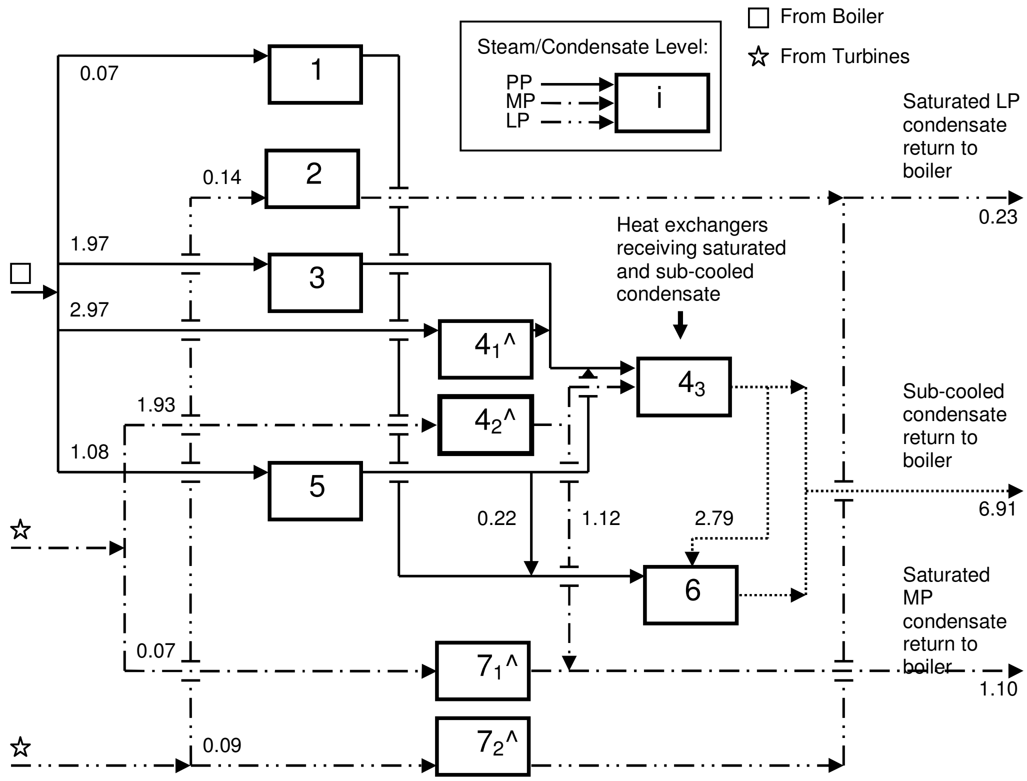


Figure 5.6: New HEN arrangement to maintain boiler efficiency.

In the figure it can be seen that the process streams associated with heat exchangers 4 and 7 required an extra heat exchanger to accommodate the multiple steam levels needed to heat these streams. These heat exchangers have been distinguished by a (^). The process stream associated with heat exchanger 4 also requires saturated condensate and as such is heated by 3 heat exchangers. Overall 10 heat exchangers are required for this HEN arrangement. The return temperature was found to be 118.4°C.

Case 2 showed that the minimum flowrate could indeed be achieved but the boiler efficiency was decreased to 60.5%, a 4.6% reduction. The boiler return temperature dropped to 89.1°C. The HEN arrangement is the same as that of Figure 5.5 as the minimum flowrate is not compromised.

5.2.3 Dedicated preheater to maintain boiler efficiency

The improvement from Model F was implemented into the steam system. The minimum steam flowrate of the process pressure stream was found to be 6.48kg/s. This is 14.1% higher than the minimum steam flowrate, but still a fairly considerable 23.7% reduction on the original parallel HEN process pressure steam flowrate. Figure 5.7 shows the HEN arrangement for this solution.

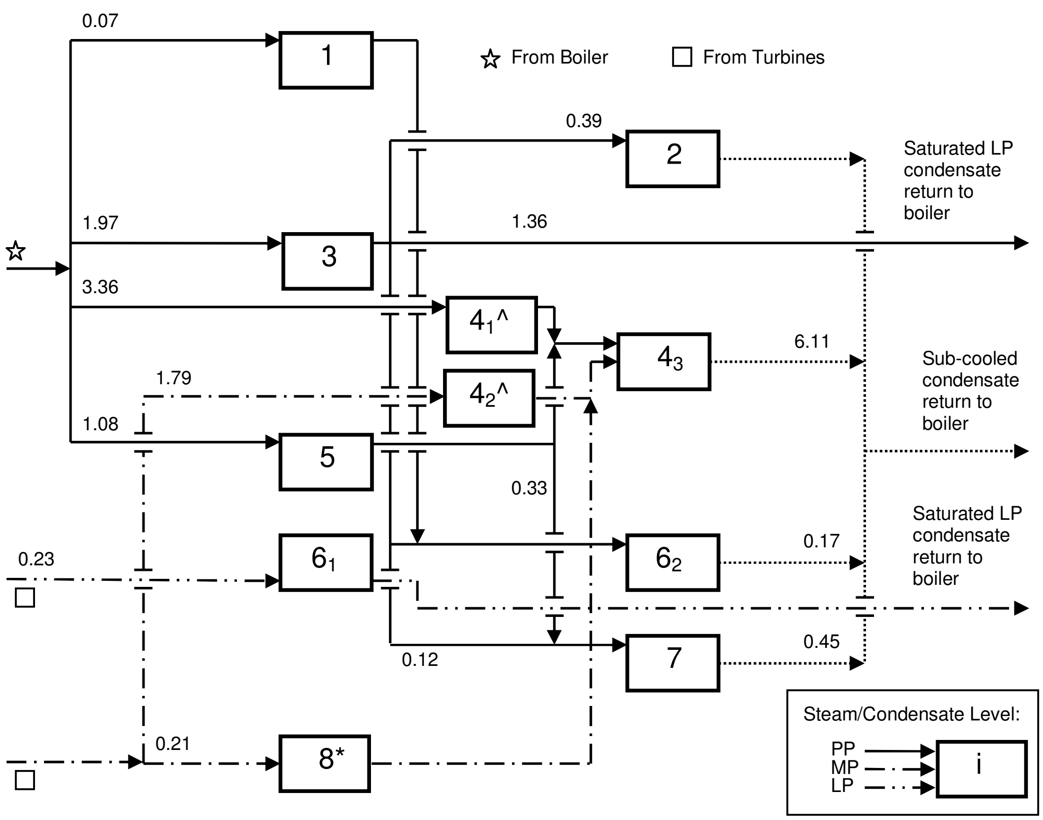


Figure 5.7: HEN arrangement showing use of additional heat exchanger.

The process outlet temperature was calculated as 106.0°C. This is fairly high and there may be a risk of pump cavitation. The duty of the dedicated preheater was calculated as 409.8kW. If this value is increased it may allow the process outlet temperature to be lower. A lower limit of 800 kW was set fro the preheater duty. The same flowrate of 6.48kg/s was calculated, and the duty of the preheater was found to be 1033.0kW. This had the effect of reducing the process outlet temperature to 89.4°C which should greatly reduce the risk of cavitation. This procedure can be explained by a simple energy balance around the process in Figure 4.9 from Section 4.3.2. If the steam flowrate remains the same, any energy added to the preheater can be taken away from the process outlet stream. This process could be continued until an acceptable process outlet temperature is reached. The drawback is however a larger preheater which increases the capital cost of the system.

5.3 Pressure Drop Considerations

The case study considered in the previous sections concerning flowrate minimisation and boiler efficiency will be used here. By assuming an overall heat transfer coefficient and using the duty and the minimum approach temperature the relevant heat exchangers in the case study were designed according to the guidelines discussed in Appendix A. As already mentioned these may not be optimal heat exchangers as the optimal design is beyond the scope of this work. They are however realistic heat exchangers according to the guidelines discussed in Sinnot (2005).

5.3.1 Pressure drop correlations

Tables 5.7 and 5.8 show the relevant design information for the heat exchangers in the case study. Since the design conditions are different for the condensers and the condensate heat exchangers different profile designs are used. Table 5.7 is for the heat exchangers where phase change occurs and Table 5.8 is for the condensate heat exchangers. Using these parameters as well as the mass flowrates for the heat exchangers, ranging from zero to the maximum flowrate, the pressure drop for the heat exchanger is plotted against mass flowrate. Typical plots are shown in Figure 5.8 and 5.9 showing the pressure drop for heat exchanger 1 as a condenser and a condensate heat exchanger respectively.

Table 5.7: Design information for phase change heat exchangers.

Heat exchanger	Duty (kW)	ΔT_{min}	Area (m^2)	n_{tp}	N_t	L (m)	d_i (m)
1	135	189.82	0.71	5	8	1.83	0.0128
2	320	189.82	1.69	7	10	1.83	0.024
3	3620	12.77	283.57	10	660	8.54	0.0128
4	12980	81.87	158.54	16	363	3.66	0.0304
5	1980	10.00	198	10	404	9.76	0.0128
6	635	167.66	3.787	5	42	1.83	0.0128
7	330	167.66	1.968	5	22	1.83	0.0128

Table 5.8: Design information for heat exchangers.

Heat exchanger	Duty (kW)	ΔT_{min}	Area (m^2)	n_{tp}	N_t	L (m)	d_i (m)
1	135	10.00	16.875	5	69	4.88	0.0128
2	320	10.00	40	7	218	3.66	0.0128
3	3620	10.00	452.5	10	923	9.76	0.0128
4	12980	10.00	1622.5	16	1557	20.74	0.0128
5	1980	10.00	247.5	10	808	6.1	0.0128
6	635	10.00	79.375	5	259	6.1	0.0128
7	330	10.00	41.25	5	135	6.1	0.0128

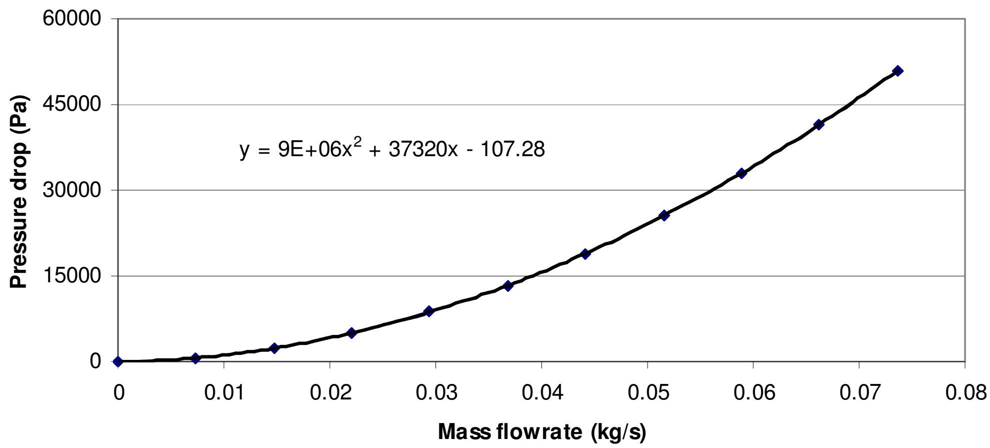


Figure 5.8: Pressure drop for heat exchanger 1 with phase change.

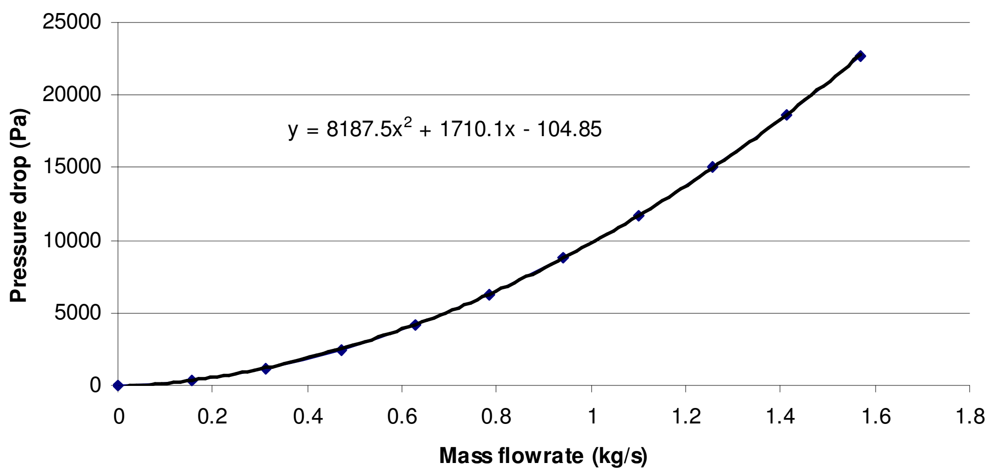


Figure 5.9: Pressure drop for heat exchanger 1 without phase change.

With these plots a trend line can be constructed and an approximation found for the pressure drop as a function of mass flowrate. A second order approximation for each of the plots is found to be the most appropriate. Since this creates a nonlinear pressure drop relation the plots are then approximated by two straight lines. This then produces a piecewise linear approximation for the pressure drop. An example of the piecewise linear approximation is shown in Figure 5.10, using heat exchanger 1 with phase change as the pressure drop plot.

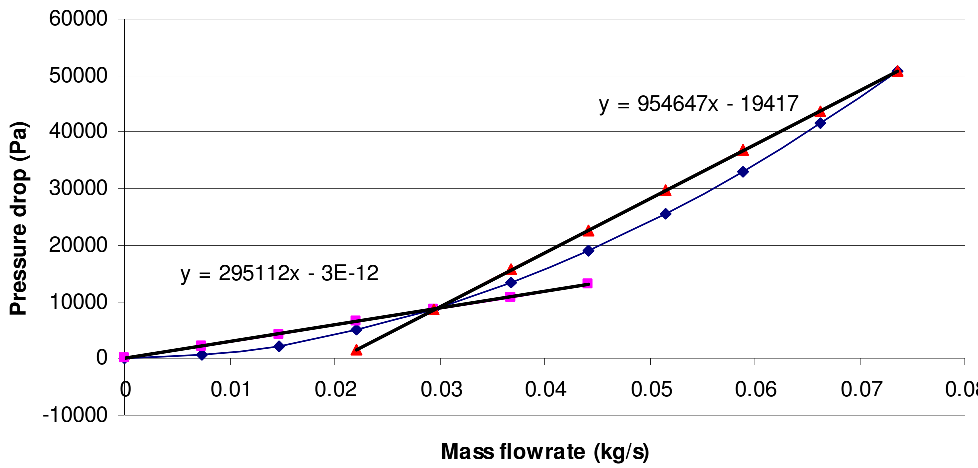


Figure 5.10: Piecewise linear approximation for condenser pressure drop.

This procedure is repeated for each heat exchanger, thus creating a linear approximation as well as a more accurate nonlinear approximation for the pressure drop as a function of mass flowrate which is used in the model.

The four piping systems are designed in a similar manner. The only decision variable in Constraint (2.22) is pipe length. This is a very plant specific variable and as such values are assumed for the various piping systems. Figure 5.11 shows the pressure drop as a function of mass flowrate for the system of pipes connecting the phase change heat exchangers to the heat exchangers receiving saturated condensate. A trend line has been constructed and the equation shown.

The piping system pressure drop correlation is also largely nonlinear and can best be approximated by a cubic function. Two lines are used to approximate the piping system pressure drops in the same piecewise linear fashion as for the heat exchangers. Figure 5.12 shows these approximation lines.

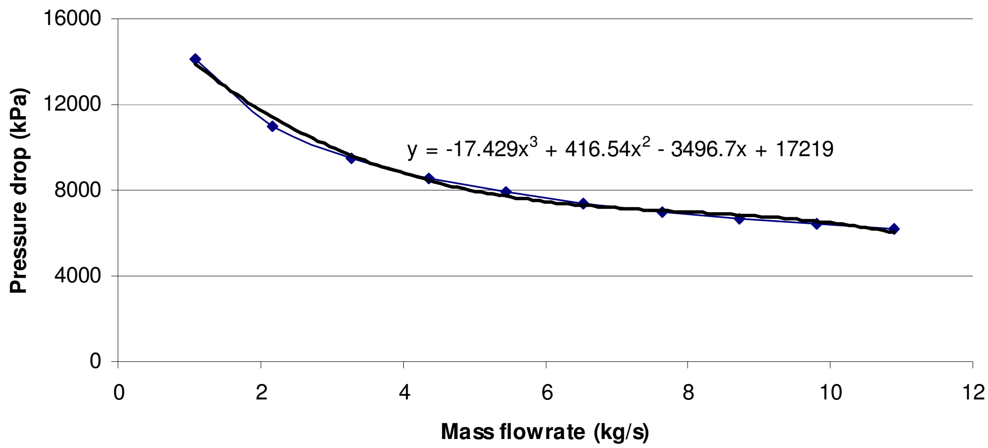


Figure 5.11: Pressure drop correlation for saturated condensate piping.

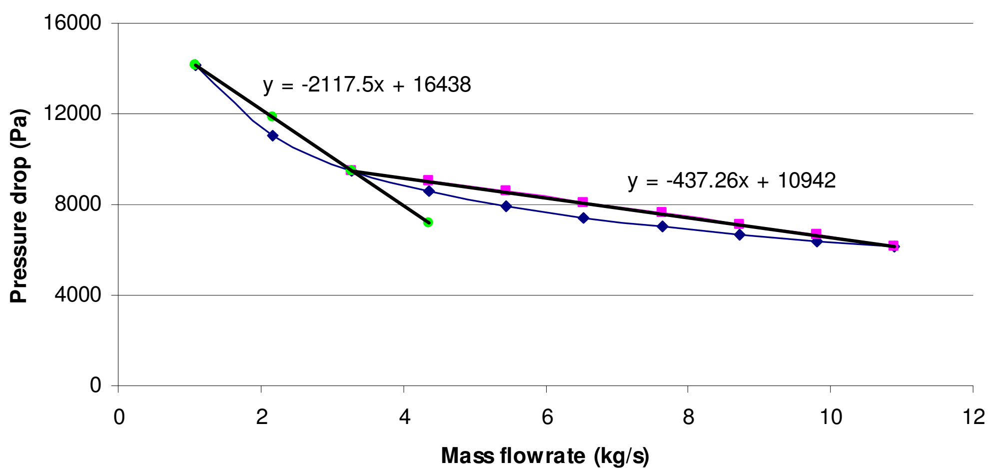


Figure 5.12: Piecewise linear approximation for piping pressure drop.

Using the figures shown in this section, each heat exchanger and piping system is assigned a set of piecewise linear approximations as well as a nonlinear approximation of pressure drop as a function of mass flowrate. These will be used appropriately in the CPA pressure constraints such as Constraint (4.86).

5.3.2 Pressure drop minimisation

Three separate models were established to examine pressure drop in the steam system heat exchanger networks. The first used the piecewise linear approximations for pressure drop and thus formed an MILP. The second used the same MILP model as a starting point for the nonlinear, however more accurate approximations. This created a combination MILP/MINLP. The final model was a purely nonlinear model consequently forming an MINLP.

The models were structured to use the original flowrate minimisation model of Coetzee and Majozi (2008) from Section 4.1 to find the minimum steam flowrate for the network. This flowrate was then used as a parameter in the models which minimised pressure drop by rearranging the same network. With the total flowrate accounted for, the only other decision variable in the system was the number of split heat exchangers. The results varied with the different number of allowed heat exchangers and therefore several runs were done with each of the three models.

The minimum steam flowrate was calculated as 7.68 kg/s, as seen in Section 5.1.1. This was achieved with a single heat exchanger split and as such this single split was allowed for the initial pressure drop minimisation. Table 5.9 shows the results from the three models for comparison. Figure 5.13 then shows the network arrangement for the minimum flowrate, while Figure 5.14 shows the network arrangement exhibiting the minimum pressure drop taken from the combination MILP/MINLP model. It was noted that the minimum flowrate had to be relaxed slightly for the various models to solve in GAMS. The linearised models had to be relaxed by a value of approximately 2%, while the nonlinear model less than 1%.

Table 5.9: Pressure drop from the various models.

Model type	Starting Pressure (kPa)	Pressure drop (kPa)
Type 1: Single MILP	2 550	348.94
Type 2: MILP and MINLP	2 550	323.42
Type 3: Single MINLP	2 550	323.42

From Table 5.9 it can be seen that the pressure drop for the linearised (Type 1) model is higher than for the combined (Type 2) and nonlinear (Type 3) models. It is likely that this is due to the conservative nature of the linearisation. Since the purely nonlinear model exhibits the same pressure drop as the combined model it can be concluded that using the linearised solution as a starting point did not improve the solution.

The networks designed by the MILP/MINLP model were examined. The initial flowrate minimisation network designed without pressure drop constraints is shown in Figure 5.13.

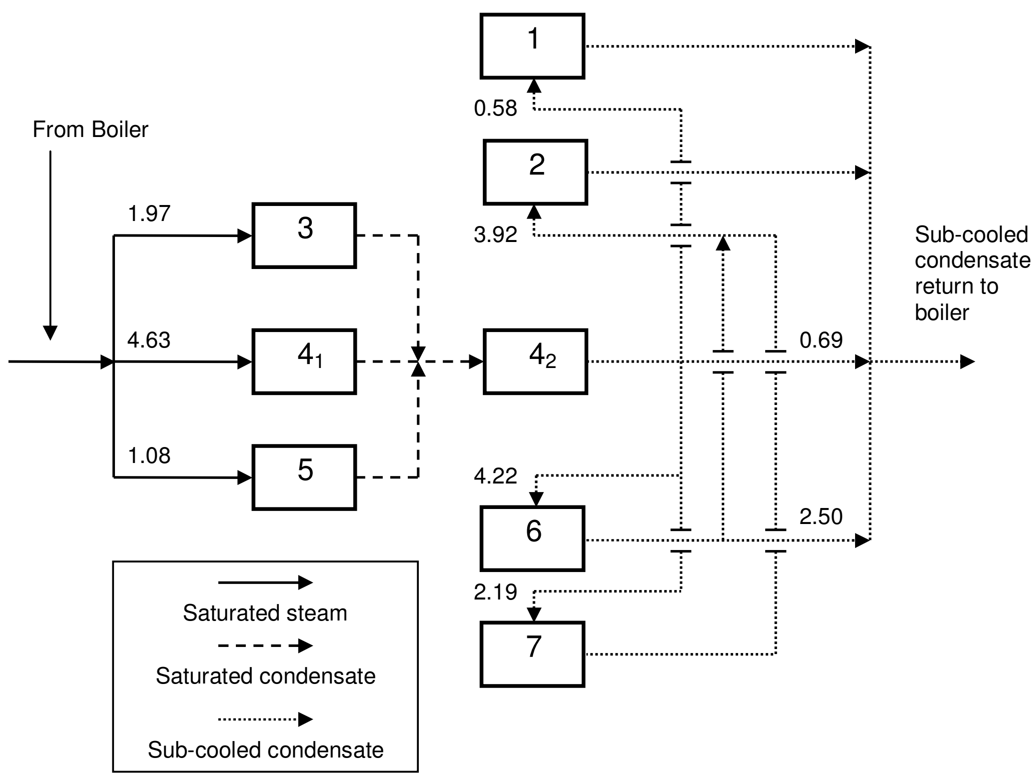
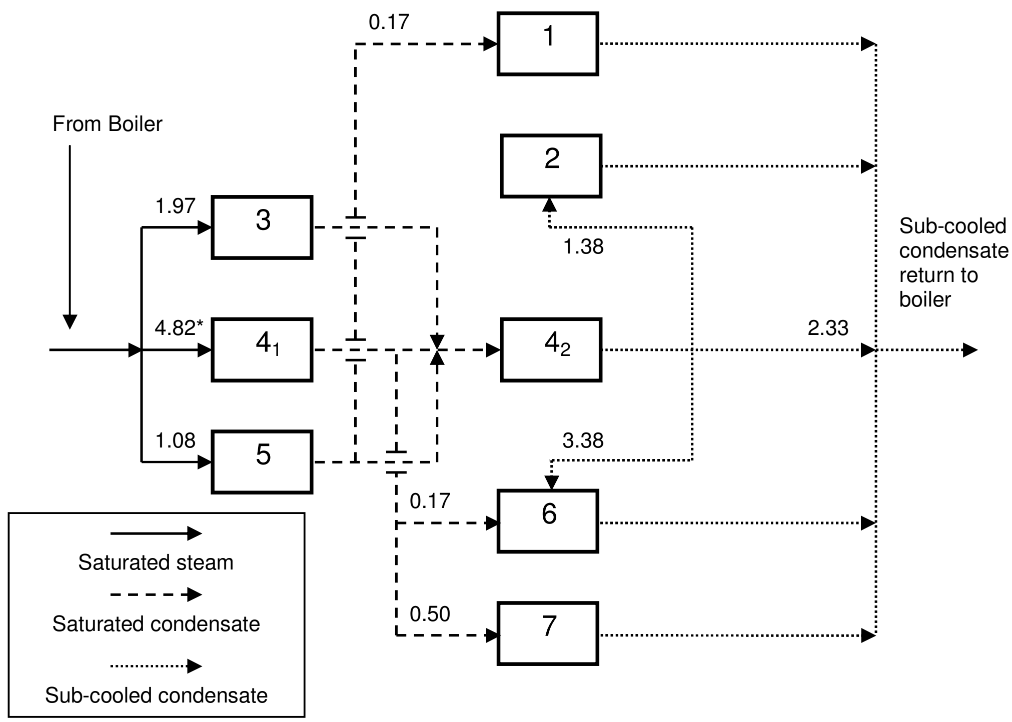


Figure 5.13: Network arrangement for minimum flowrate.

Figure 5.13 is different from the network in Figure 5.2. This is possible since multiple networks can be designed for the same minimum flowrate. Figure 5.14 is the network with the minimum pressure drop. Of the multiple feasible networks the model designs the one that has the lowest pressure drop.



* this value differs from Figure 5.13 due to the slight relaxation mentioned in the text

Figure 5.14: Network arrangement for minimum pressure drop.

If the networks in Figure 5.13 and Figure 5.14 are compared it can be seen that there is less reuse from heat exchanger 4 for the minimum pressure drop case. This is brought about by more saturated condensate being dispersed in Figure 5.14. In Figure 5.13 all of the saturated condensate is sent directly to heat exchanger 4. The reduced reuse causes the pressure drop to be lower. Figure 5.15 shows the critical path of pressure drop for the network in Figure 5.14. The solid lines represent the path of greatest pressure drop and the pressure of the respective nodes in that path are shown in the figure.

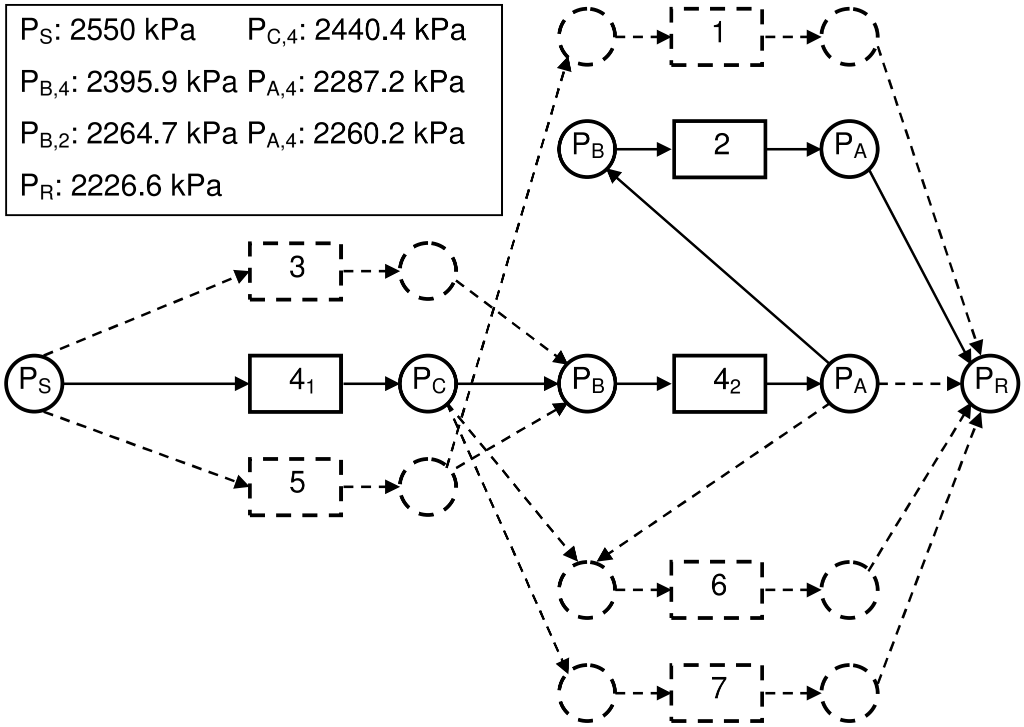


Figure 5.15: Maximum pressure drop path which is minimised.

The effect of varying the number of split heat exchangers was then explored further. By increasing the number of splits it may be possible to further reduce the pressure drop that corresponds to the minimum flowrate. Table 5.10 shows the various models and the pressure drops for the varying number of split heat exchangers. The pressure drop with a single heat exchanger split is repeated from Table 5.9 for comparative purposes. The lowest pressure drop for each model has been highlighted.

By comparing the various models it can be seen that the MILP model shows considerably lower pressure drops for an increase in the number of split heat exchangers. Using these solutions as a starting point for the combination MILP/MINLP model did not show the same trend of low pressure drops. If the combination MILP/MINLP model and the MINLP model are compared it can be noted that in some cases the combination model exhibits lower pressure drops and in others much higher pressure drops. This is most apparent with more allowable split heat exchangers which give more freedom to the system but also introduce more complexity. In terms of the MILP/MINLP and MINLP models' performance the varied results make it unclear which is superior. The minimum pressure drop for the MILP/MINLP and MINLP models are similar to the pressure drop found with a single split.

Table 5.10: Model response to varying heat exchanger splits.

Model type	Number of splits	Pressure drop (kPa)
MILP	1	348.94
	2	238.98
	3	235.45
	4	230.39
	5	229.12
	6	230.66
	7	228.24
MILP/MINLP	1	323.42
	2	319.62
	3	350.52
	4	405.86
	5	434.42
	6	419.75
	7	351.22
MINLP	1	323.42
	2	321.3
	3	319.02
	4	316.67
	5	404.81
	6	430.09
	7	317.73

The number of split heat exchangers was then made a variable in each of the models. The results can be seen in Table 5.11. The MILP model showed the same minimum pressure drop as when the split heat exchangers were varied manually. The MILP/MINLP model showed a pressure drop that was slightly higher than the minimum from Table 5.10, however substantially lower than the value for 4 heat exchanger splits form that table. The addition of the split heat exchanger variable caused the already complex and nonconvex MINLP model to fail to find a starting point for the problem and thus no result could be found.

Table 5.11: Sensitivity analysis with splits as a variable.

Model	Optimum splits	Pressure drop (kPa)
MILP	7	228.24
MILP/MINLP	4	323.71
MINLP	-	-

If Tables 5.10 and 5.11 are compared, it can be concluded that the number of split heat exchangers is a critical variable in the solution of the minimum pressure drop problem. The lowest pressure drop was calculated by varying the number of splits manually, however, for large systems this may be problematic.

When Tables 5.9, 5.10 and 5.11 are compared the feasible solutions from the MILP/MINLP and MINLP models are all fairly comparable. Since an additional heat exchanger is required for each split the capital cost increases dramatically. It is for this reason that the single heat exchanger split network shown in Figure 5.14 is recommended for this case study.

5.3.3 Minimum network pressure drop

In the event that the pressure drop for a system is too large, it may be of use to a designer to relax the minimum steam flowrate to find a more suitable pressure drop. A similar approach has already been implemented in the boiler efficiency section.

As mentioned in Section 4.5.1, the mathematical intricacies of the CPA require that it is solved independently. It is for this reason that the maximum allowable pressure drop for the network cannot simply be set as a parameter for a model that finds the minimum flowrate for that pressure drop. Therefore the minimum flowrate is found beforehand and then relaxed such that the minimum network pressure drop is found. In the event that this pressure drop is below the maximum allowable for the network, then the relaxed steam flowrate can once again be decreased until the minimum pressure drop found matches the maximum allowable for the network. This approach will need to occur in an iterative fashion. Since the pressure drop is a combination of nonlinear relationships to mass flowrate as well as the network structure the pressure drop will not necessarily decrease the more the steam flowrate is compromised.

Table 5.12 shows the minimum pressure drop that was found as well as the amount by which the model compromised the minimum steam flowrate to achieve that pressure drop. This is shown as 'Slack' in the table. The number of allowable heat exchanger splits was varied manually since including the splits as a variable greatly increased the solution time. The number of these splits is also included in Table 5.12. As before, the complexity and nonconvexity of the models containing nonlinear elements led to no starting points being found for several simulations and therefore no feasible solutions.

Table 5.12: Sensitivity analysis results.

Model type	Number of splits	Pressure drop (kPa)	Slack (kg/s)
MILP	1	276.77	0.452
	2	186.72	0.549
	3	177.15	0.571
	4	176.34	0.575
	5	176.26	0.579
	6	178.12	0.581
	7	177.59	0.587
MILP/MINLP	1	340.33	0.095
	2	286.84	-
	3	205.56	1.940
	4	215.54	1.936
	5	215.54	1.936
	6	338.93*	0.968
	7	226.16	1.771
MINLP	1	260.02	0.452
	2	286.84*	-
	3	205.56	1.940
	4	215.54	1.936
	5	218.32	1.936
	6	338.93*	0.968
	7	226.15	1.773

The number of heat exchanger splits was then included as a variable to see if a better solution could be found despite the larger solution time. The results are shown in Table 5.13.

Table 5.13: Sensitivity analysis with splits as a variable.

Model	Optimum splits	Pressure drop (kPa)	Slack (kg/s)
MILP	7	177.59	0.587
MILP/MINLP	5	226.06	1.853
MINLP	5	226.06	1.853

For these results it can be noted that the linearised model has a lower pressure drop than the nonlinear model. Also the combined MILP/MINLP model did not perform better than the purely nonlinear model. This shows that the starting point from the MILP did not improve the MINLP solution. The extra time and resources would tend to discourage users from this option.

5.4 Combined Boiler Efficiency and Pressure Drop

The same case study was used to compare the network using only the boiler efficiency constraints and using the boiler efficiency as well as the pressure drop constraints.

5.4.1 Sensible heat preheater

As before the boiler efficiency was maintained using 79% of the available sensible heat. The original network shown in Figure 5.2 could thus be utilised. This network has one heat exchanger split, thus it was attempted to use a single split for the pressure drop model. By incorporating the pressure drop constraints the minimum pressure drop using the minimum flowrate was found to be 344.41 kPa and the HEN can be seen in Figure 5.16 below. By comparing the networks it can be seen that additional reuse occurs between the units. By varying the number of heat exchanger splits in the pressure drop section it was found that a minimum pressure drop of 344.41 kPa could not be improved. Thus this design will be suggested.

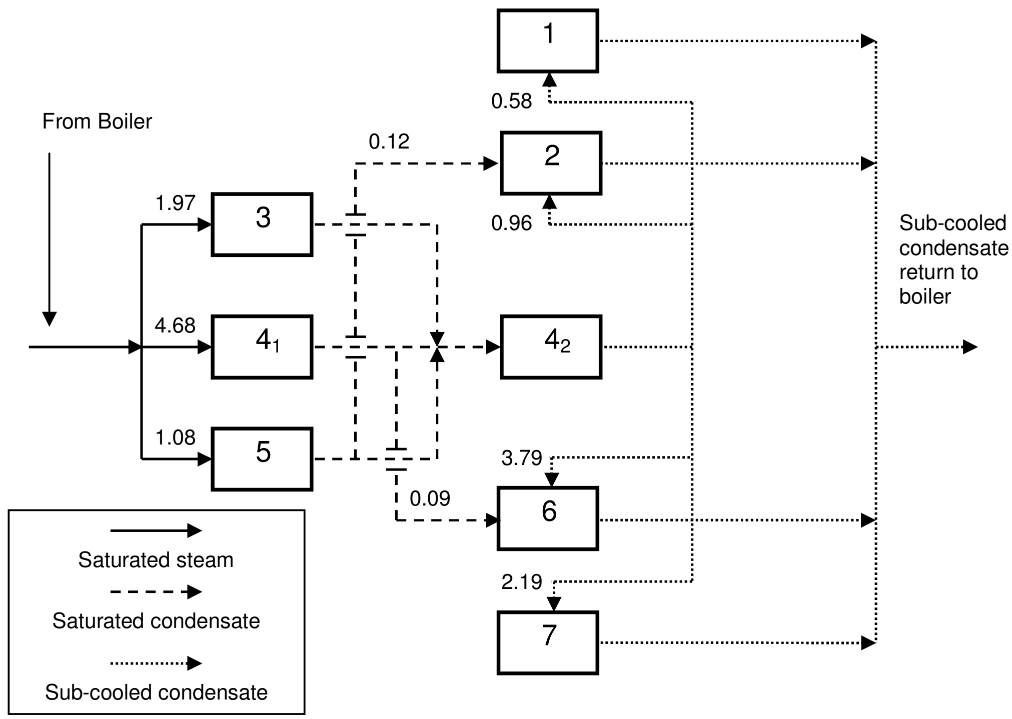


Figure 5.16: Minimum pressure drop whist maintaining boiler efficiency.

If the available sensible heat is limited to 30% the boiler efficiency can be maintained with an increase of 8.3% in the minimum flowrate. The original network can be seen in Figure 5.3. This new minimum flowrate was found using 1 heat exchanger split, thus this value was used for the pressure drop model. No feasible starting point could be found with a single split heat exchanger and as such the solution was infeasible. By increasing the number of splits to 2, a solution of 272.31 kPa was found and the network associated with this can be seen as Figure 5.17 below. The additional heat exchanger did not influence the minimum flowrate. Comparing this network to the original it can be seen that more reuse streams are required and no saturated condensate is returned to the boiler directly. By varying the number of heat exchanger splits in the pressure drop section it was found that the minimum pressure drop of 272.31 kPa could not be improved. Setting the number of heat exchanger splits as a variable resulted in a similar situation as previously mentioned, with an infeasible solution resulting from a lack of a feasible starting point. From this result it can be deduced that manually changing the number of heat exchanger splits improves the performance of the model by removing a largely influential variable.

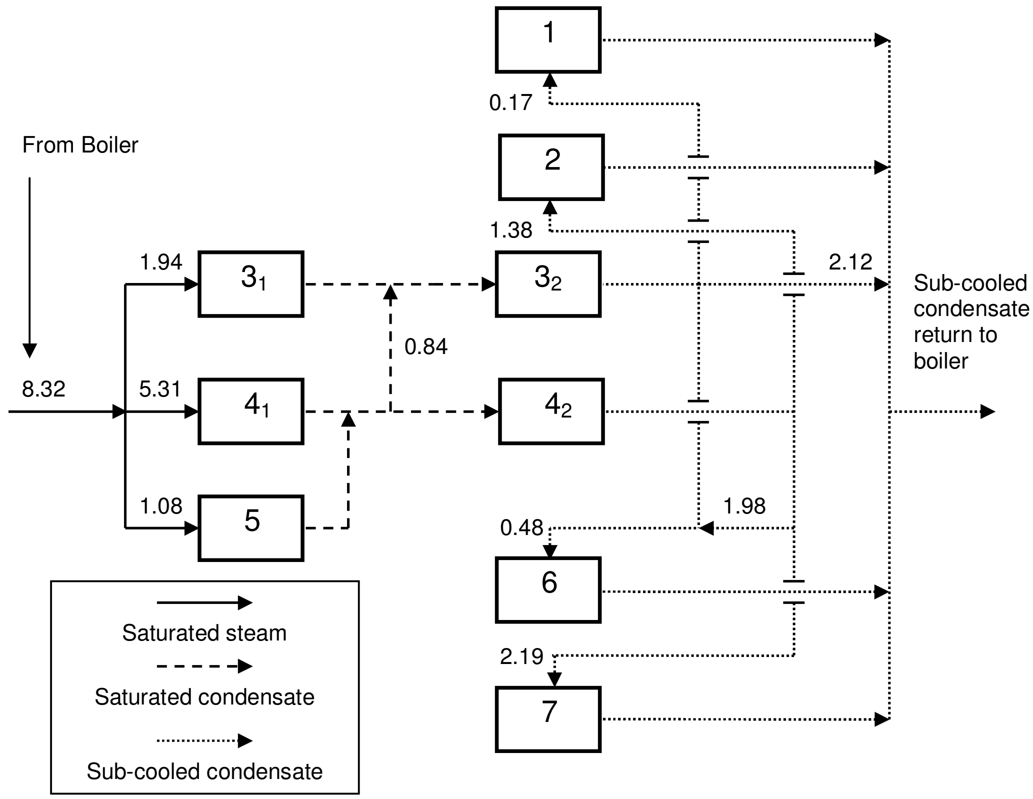


Figure 5.17: Network for minimum pressure drop for reduced sensible heat.

5.4.2 Dedicated preheater

By adding a dedicated heat exchanger to preheat the boiler return stream it was found that the boiler efficiency could be maintained with a flowrate 13.9% higher than the minimum. This network can be seen in Figure 5.4. This minimum flowrate was achieved with 1 heat exchanger split, as well as the dedicated preheater. By minimising the pressure drop using this flowrate and the number of heat exchanger splits the pressure drop was found to be 231.94 kPa and the associated HEN can be seen in Figure 5.18 below. By comparing this to the original network it can be seen that the dedicated preheater is eliminated. Since saturated condensate is recycled to heat exchanger 5 the same number of heat exchangers exist. This creates a situation where the return stream temperature before the pump is very high creating a risk of cavitation. By varying the number of splits for the pressure drop model the minimum pressure drop of 231.94 kPa could not be improved.

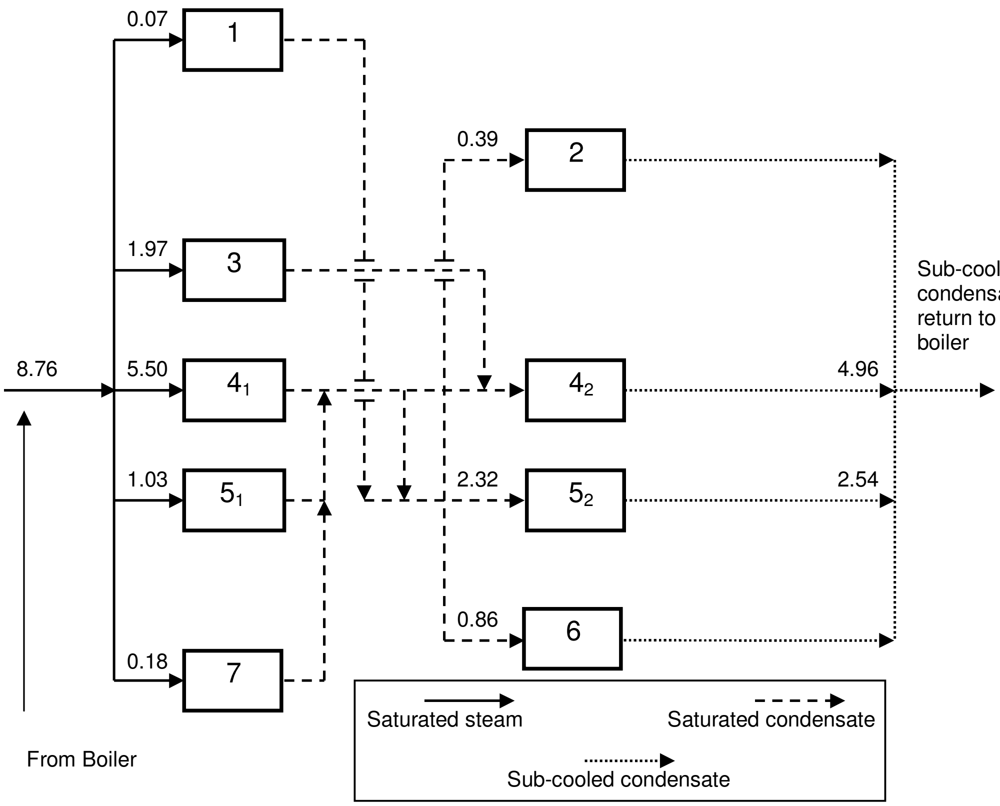


Figure 5.18: Network for extra preheater.

Since the return temperature is very high for the previous solution it was decided to force the model to use the dedicated preheater. The result of this is a lower temperature stream arriving at the pump, which reduces the risk of cavitation. The pumped stream is then preheated to the necessary temperature to maintain the boiler efficiency. This was accomplished by using the same specifications for the preheater as was found in the purely boiler efficiency model. After setting the duty and limiting temperatures a solution was found that utilised the dedicated preheater. With 2 heat exchanger splits the minimum pressure drop was found to be 262.23 kPa. However when the number of heat exchanger splits was increased the lowest pressure drop was achieved with 3 splits and was calculated as 255.40 kPa. The network corresponding to this pressure drop can be seen in Figure 5.19. By using the number of splits as a variable in the formulation the minimum pressure drop was found to be 321.70 kPa. Once again it can be seen that setting the number of splits manually leads to a better solution due to the nonlinearity and nonconvexity of the pressure drop model.

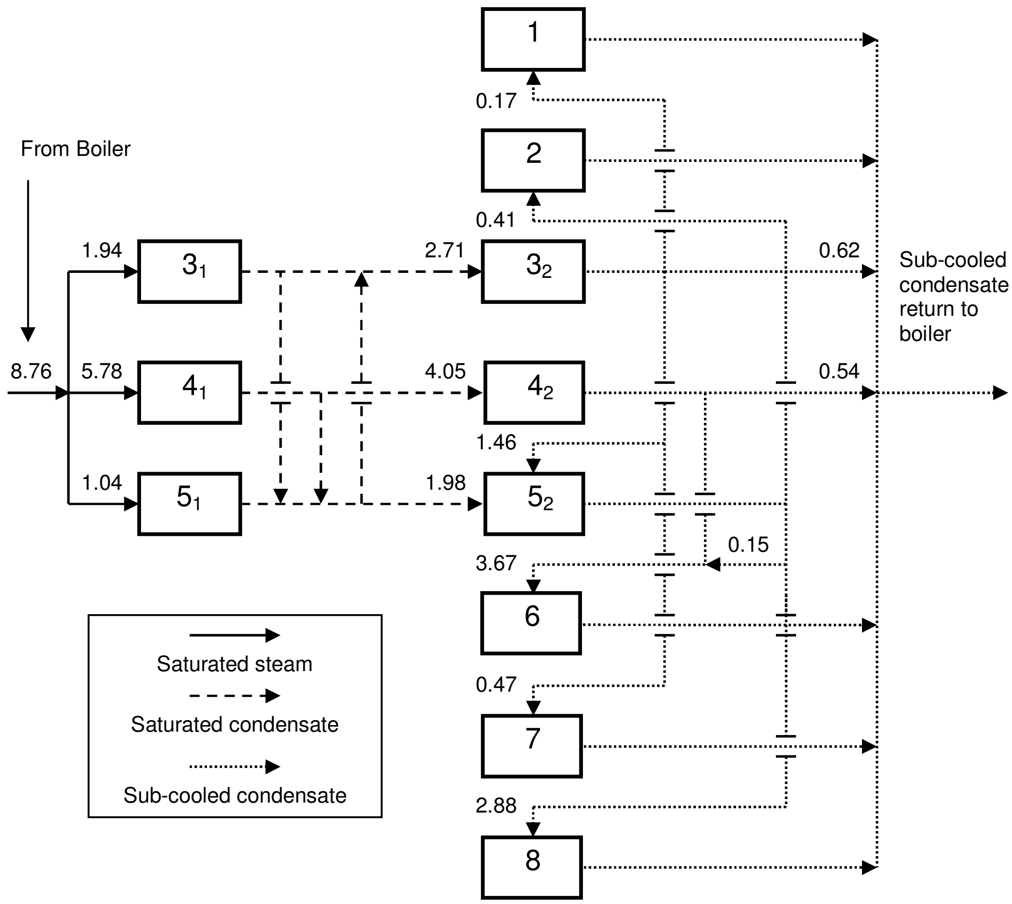


Figure 5.19: Minimum pressure drop network with dedicated preheater.

It is clear from Figure 5.19 that some mathematically feasible choices are poor design choices. For instance the recycle of saturated condensate from heat exchanger 3 to heat exchanger 5, and then from heat exchanger 5 to heat exchanger 3 is unnecessary. This could be eliminated to further save on piping costs. The recycle of subcooled condensate is also very complex. A cost analysis with piping could properly determine if this complexity is necessary for the sake of reduced pressure drop.

5.5 References

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Shang, Z. and Kokossis, A. (2004) A transshipment model for the optimisation of steam levels of total site utility system for multiperiod operation, Computers and Chemical Engineering 28, pages 1673-1688.

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6. Conclusions and Recommendations

The conclusions from the work presented in this dissertation are presented in this chapter. They are arranged in the same order as they appear in Chapter 4. The limitations and recommendations for further work are given at the end of this chapter.

6.1 Single Pressure Level

From reducing the steam flowrate while maintaining boiler efficiency for single steam pressure levels the following conclusions can be made:

- Reducing steam flowrate affects the boiler efficiency by lowering boiler return temperature and flowrate.
- Preheating the return flow to a slightly higher temperature will maintain boiler efficiency for a reduced steam flowrate.
- The system let down valve can be replaced by a heat exchanger that will utilise the sensible heat of the high pressure steam to preheat the boiler feed water.
- In the event of there not being enough sensible heat to maintain the boiler efficiency with the minimum steam flowrate, a compromise in either the boiler efficiency or the minimum steam flowrate must be made, unless another preheating source is used.
- It is occasionally possible to increase boiler efficiency by further preheating the boiler feed water by utilising as much of the sensible heat as possible.

The following conclusions can be made from the results of a case study utilising the formulation of reducing the steam flowrate while maintaining boiler efficiency:

- By reducing the steam flowrate, a saving of 29.6% in steam can be realised.

- This lower flowrate has the effect of reducing the boiler return temperature from 116°C to 78.6°C.
- This reduction in return temperature in turn leads to a 5.9% drop in boiler efficiency.
- A return feed temperature of 117.6°C will maintain the boiler efficiency with the reduced flowrate.
- 79.2% of the HP steam sensible heat is required to preheat the feed stream.
- Thus it is possible to increase the boiler efficiency by up to 1.7% by utilising all of the available sensible heat.

From the case study, the following conclusions can be made for the sensitivity analysis where insufficient sensible heat was supplied to the system, forcing a compromise between the boiler efficiency and the minimum steam flowrate:

- Reducing the available sensible heat to 30% showed how the minimum steam flowrate or boiler efficiency must be changed to satisfy the other.
- By maintaining the boiler efficiency the minimum steam flowrate increased from 7.68 kg/s to 8.32kg/s.
- By maintaining the minimum steam flowrate the boiler efficiency dropped from 63.49% to 61.30%.

The following conclusions can be drawn from the scenario where an additional heat exchanger was added to the system:

- The inclusion of an additional preheating heat exchanger that utilises steam from the steam system was able to maintain the boiler efficiency.
- The minimum steam flowrate had to be increased from 7.68kg/s to 8.76kg/s; a 13.9% increase.

- This is a 19.6% decrease from the original parallel network flowrate.

6.2 Multiple Pressure Levels

The following conclusions can be made about reducing steam flowrate while maintaining boiler efficiency where multiple steam pressure levels are catered for:

- By incorporating the background processes of steam turbines into the HEN and minimising the flowrate, some of the excess exhaust from the turbines can be used in the HEN.
- The system let down valves can be replaced by two heat exchangers that will utilise the sensible heat of the high pressure steam and the exhaust from the high pressure turbine to preheat the boiler feed water.
- In the event of there not being enough sensible heat to maintain the boiler efficiency with the minimum steam flowrate, a compromise in either the boiler efficiency or the minimum steam flowrate must be made.

The following conclusions can be made from the results of a case study utilising the formulation of reducing the steam flowrate while maintaining boiler efficiency:

- By reducing the steam flowrate a saving of 33.2% in steam can be realised.
- This lower flowrate has the effect of reducing the boiler return temperature from 116°C to 63.0°C.
- This reduction in return temperature in turn leads to an 8.2% drop in boiler efficiency.
- A return feed temperature of 119.0°C will maintain the boiler efficiency with the reduced flowrate.

- 60.1% of the HP steam sensible heat and 80% of the MP sensible heat is required to preheat the feed stream.
- Thus it is possible to increase the boiler efficiency by up to 5.2% by utilising all of the available sensible heat.

In the case study the following conclusions can be made for the case where insufficient sensible heat was available for the system which required a compromise between boiler efficiency and steam flowrate:

- Reducing the available sensible heat to 30% showed how the minimum steam flowrate or boiler efficiency must be changed to satisfy the other.
- By maintaining the boiler efficiency the minimum steam flowrate increased from 5.68 kg/s to 6.096 kg/s.
- By maintaining the minimum steam flowrate the boiler efficiency dropped from 63.40% to 60.5%.

From the case study the following conclusions can be drawn from the scenario where an additional heat exchanger was added to the system to maintain the boiler efficiency:

- The boiler efficiency was maintainable without requiring an additional heat exchanger as a result of an increase in saturated condensate return to the boiler.
- The minimum flowrate increased from 5.68 kg/s to 6.48 kg/s; an increase of 14.1%.
- This is still a saving of 23.7% of the original parallel arrangement flowrate.
- This situation results in a higher pumping temperature which could cause cavitation.
- This risk can be eliminated by increasing the number of heat exchangers and dedicating energy to heat the return stream after pumping has occurred.

6.3 Pressure Drop

The following conclusions can be drawn from the minimisation of pressure drop through the heat exchanger network:

- Given that multiple networks can exhibit the same minimum flowrate the network that exhibits the lowest pressure drop amongst these should be found.
- Pressure drop is dependent on the network structure and thus requires a means of finding the pressure drop based on the structure as well as the elements causing pressure drop.
- The Critical Path Algorithm can find the largest pressure drop in the network.
- With the largest pressure drop known it can be minimised using mathematical modelling and optimisation.

From the case study the following conclusions can be made:

- For the given minimum flowrate the lowest pressure drop that could be achieved was 323.42 kPa.
- The minimum pressure drop was achieved with 3 heat exchanger splits.
- The minimum pressure drop of the system where the flowrate was relaxed was found to be 205.56 kPa.
- This minimum also required 3 heat exchanger splits and resulted in a flowrate relaxation of 1.940 kg/s. This implies that the flowrate increased by 1.940 kg/s from the minimum flowrate that was originally obtained.

From the case study where the boiler efficiency and pressure drop constraints were combined, the following conclusions were drawn:

- The boiler efficiency constraints with the sensible heat preheater resulted in a higher pressure drop for the system. This was found to be 344.41 kPa.

- This result was achieved with a single heat exchanger split.
- The boiler efficiency constraints with the dedicated preheater resulted in a pressure drop of 231.94 kPa.
- This was achieved with a single heat exchanger split as well as the dedicated preheater.
- The dedicated preheater results in a lower pressure drop when boiler efficiency and pressure drop are considered. This is due to the complexity of the relationship between pressure drop and network structure.

6.4 Recommendations

From this work the following recommendations and are made:

- A cost analysis would be of great value since maintaining boiler efficiency requires extra heat exchangers and piping.
- A more detailed boiler efficiency model is needed to verify the results of this study as the regression parameters used were not for the boiler specific to this case study.
- A more accurate linearisation of the pressure drop functions could possibly lead to better solutions.

7. Nomenclature

7.1 Sets

I = {i or j | i or j = 1, 2, …, I} is the set of heat exchangers

L = {l | l = hp, mp, or lp} is the set of various saturated steam pressure levels

7.2 Parameters

A	heat transfer area for heat exchanger
a	regression parameter
BP	large pressure to compensate for nodes that do not exist
b	regression parameter
c_p	heat capacity, (kJ/kg.k)
D_i	pipe inside diameter, (m)
d_i	inside tube diameter, (m)
d_o	outside tube diameter, (m)
F^U	maximum capacity of the boiler (kg/s)
F_{turb}	flowrate of turbine condensate, (kg/s)
h_{sup}	enthalpy of superheated HP steam, (kJ/kg)
h_{sat}	enthalpy of saturated HP steam, (kJ/kg)
h_{sup}^{HP}	enthalpy of superheated HP steam, (kJ/kg)
h_{sup}^{MP}	enthalpy of superheated MP steam, (kJ/kg)
h_{sat}^{HP}	enthalpy of saturated HP steam, (kJ/kg)
h_{sat}^{MP}	enthalpy of saturated MP steam, (kJ/kg)
L	pipe length, (m)
m	number of allowable split heat exchangers due to variable pressure levels
m_p	fluid mass flowrate through pipes, (kg/s)

m_t	fluid mass flowrate through tube side of heat exchanger, (kg/s)
n	number of allowable heat exchanger splits
n_{tp}	number of tube passes in heat exchanger
N_t	number of tubes in heat exchanger
N_p^{EX}	pressure drop factor for pipes
N_{t1}	pressure drop factor for friction loss
N_{t2}	pressure drop factor for sudden expansions, contractions and flow reversals
Q_i	duty of heat exchanger i, (kW)
Q_{fuel}	energy contained in fuel, (kW)
Q_{loss}	energy losses inside the boiler, (kW)
Q_{steam}	energy gained by steam in the boiler, (kW)
q	latent and superheated sensible heat of HP steam, (kJ/kg)
Tin_i^L	limiting utility inlet temperature for heat exchanger i, (°C)
$Tout_i^L$	limiting utility outlet temperature for heat exchanger i, (°C)
T_{turb}	temperature of turbine condensate, (°C)
ρ	density of fluid, (kg/m^3)
μ	viscosity of fluid, (N.s/m^2)
λ	latent heat of steam, (kJ/kg)
θ	fraction of sensible heat used for preheating

7.3 Binary Variables

$$x_i = \begin{cases} 1 \text{ if heat exchanger } i \text{ receives heat from condensate} \\ 0 \text{ otherwise} \end{cases}$$

$$y_{i,l} = \begin{cases} 1 \text{ if heat exchanger } i \text{ receives heat from steam from level } l \\ 0 \text{ otherwise} \end{cases}$$

$$y_{A,B} = \begin{cases} 1 \text{ if connection between node A and B exists} \\ 0 \text{ otherwise} \end{cases}$$

7.4 Continuous Variables

F	steam load raised by the boiler, (kg/s)
FF_l	fixed flowrate for turbine at pressure level l, (kg/s)
Fin_i	total flowrate entering heat exchanger i, (kg/s)
$Fout_i$	total flowrate leaving heat exchanger i, (kg/s)
FR	total return flow to the boiler, (kg/s)
FR_i	condensate returning to the boiler form heat exchanger i, (kg/s)
$FRR_{j,i}$	reused/recycled condensate from heat exchanger j to heat exchanger i, (kg/s)
FS	total saturated steam flowrate to the heat exchanger network, (kg/s)
$L_{j,i}$	subcooled condensate reuse from heat exchanger j to heat exchanger i, (kg/s)
M	mass flow to the boiler, (kg)
MP	flowrate of medium pressure steam through preheater, (kg/s)
n_b	boiler efficiency
PA	pressure of node after condensate heat exchanger, (kPa)
PB	pressure of node before condensate heat exchanger, (kPa)
PC	pressure of node after condenser, (kPa)
PP	flowrate of process pressure steam through preheater, (kg/s)
PR	pressure of final node for boiler return stream, (kPa)
PS	source node from boiler, (kPa)
Q_i^L	portion of heat exchanger duty catered for by condensate, (kW)
Q_i^S	portion of heat exchanger duty catered for by steam, (kW)
$Q_{preheat}$	heat added to boiler return stream by various preheaters, (kW)

slack	slack variable used in objective functions
$SL_{j,i}$	saturated condensate reuse/recycle from heat exchanger *j* to heat exchanger *i*, (kg/s)
$SL_{j,i,l}$	saturated condensate reuse/recycle from heat exchanger *j* to heat exchanger *i* at pressure level *l*, (kg/s)
SS_i	saturated steam flowrate to heat exchanger *i*, (kg/s)
$SS_{i,l}$	saturated steam flowrate to heat exchanger *i* at pressure level *l*, (kg/s)
T_{boil}	temperature of boiler return flow, (°C)
T_{proc}	outlet temperature from HEN, (°C)
T_{pump}	combined temperature of process and turbine condensate, (°C)
T_{ret}	initial return temperature used to construct Figure 2.3, (°C)
V_p	volumetric flowrate through pipes, (m^3/s)
V_t	volumetric flowrate through tube side of heat exchanger, (m^3/s)
ΔP_p	pressure drop through pipes, (kPa)
ΔP_t	pressure drop through the tube side of heat exchanger, (kPa)
ΔT_{min}	global minimum approach temperature for the HEN, (°C)
ΔT_{sat}	temperature difference between boiler return and saturated HP steam, (°C)
$\Delta T_{sa,tl}$	temperature difference between boiler return and saturated HP steam, (°C)

Appendix A

This section of the Appendix is intended to show how the unknown terms relating to the heat exchanger and piping pressure drop correlations have been derived for the pressure drop mathematical model.

A.1 Heat Exchanger Guidelines

Constraint (2.10) contains the following heat exchanger design based variables; heat transfer area A, the number of tube passes n_{tp}, the number of tubes N_t, the inside tube diameter d_i and the outside tube diameter d_o.

For the various heat exchangers in the relevant case study the heat transfer area is determined using the general equation for heat transfer defined in Sinnot (2005), shown in Constraint (A.1)

$$Q = UA\Delta T_m \tag{A.1}$$

In Constraint (A.1) Q is the relevant duty for the stream to be heated, U is the overall heat transfer coefficient, A is the heat transfer area and ΔT_m is the log mean temperature difference. The stream duty is known for the case studies. An approximate overall heat transfer coefficient is taken from Sinnot (2005). This value is 1000 W/m^2.K for the condensers and 800 W/m^2.K for the condensate heat exchangers. The ΔT_m for condensers is determined using Constraint (A.2) as defined by Sinnot (2005).

$$\Delta T_m = \frac{t_2 - t_1}{\ln\left(\dfrac{T_{sat} - t_1}{T_{sat} - t_2}\right)} \tag{A.2}$$

In Constraint (A.2) T_{sat} is the saturated steam temperature, t_2 is the cold stream inlet temperature and t_1 is the cold stream outlet temperature. The ΔT_m for the condensate heat exchangers is assumed to be the minimum global approach temperature, ΔT_{min} for the case study so as to give a conservative approximation for the area. With the Q, U and ΔT_m known the heat transfer area can be calculated using Constraint (A.1).

This area must then be made up by the outside area of the tubes in the heat exchanger. The area is a function of the area of one tube multiplied by the number of tubes, as stated by Nie (1998) and shown in Constraint (A.3)

$$A = N_t \pi d_o L \tag{A.3}$$

In Constraint (A.3) L is the length of a tube. With A known, the variables N_t, L and do must be selected. There are several design guidelines given in Sinnot (2005) that aid in the selection of these variables. One of the major considerations for heat exchangers is the fluid velocity. Low fluid velocities often lead to fouling while high fluid velocities increase the pressure drop. Velocity is calculated by Constraint (A.4), shown in Nie (1998).

$$u = V \frac{4}{\pi d_i^2} \frac{n_{tp}}{N_t} \tag{9.4}$$

The number of tube passes for are assumed to start at 2 for the smaller duties and increasing up to 16 for the larger duties. Sinnot (2005) indicates generally accepted velocity ranges for fluids and gases in pipes, which can be assumed equivalent to heat exchanger tubes. These ranges are shown in Table A.1.

Table A.1: Fluid velocity ranges.

Fluid	Velocity (m/s)
Liquids	1-3
Gases and Vapours	15-30

The lower limit is in place to prevent fouling. Since steam and steam condensate contains very little impurities this lower limit is assumed to be flexible for the approximate heat exchanger design. Since the pressure drop through the steam pipes is assumed to be negligible the only steam flowrate to be considered is inside the condensers. Thus the gas and vapour flowrate range is used simply as a guideline.

The relation between the inside and outside pipe diameter is assumed to be a ratio of 0.8, since this is the ration indicated by the most commonly chosen pipe diameters in Sinnot (2005).

A relationship between the tube length and the number of tubes exists in the form of the bundle diameter to tube length relationship defined in Sinnot (2005). The bundle diameter is calculated using the number of tubes and the tube pitch. A square pitch is assumed for all heat exchangers.

By alternating the inside diameter, number of tubes, tube length and number of tube passes while maintaining the heat transfer area and bearing the design guidelines in mind the various heat exchangers are designed for the grassroot case. In the retrofit case the actual heat exchanger dimensions can be used. It must be emphasised that the objective is not to design an optimal heat exchanger but simply to simulate the pressure drop for a realistic heat exchanger such that the pressure drop can be included in the flowrate minimisation framework.

A. 2 Piping Guidelines

Constraint (2.14) is a fairly simple piping pressure drop estimation. Since an economic design guideline has already been used to derive the constraint the only choice left is the pipe length. In the model the various types of pipe length are varied. The recycle and reuse pipes are assumed to be longer than the pipes joining the heat exchangers to the central condensate return hub.

Appendix B

This section of the Appendix includes a description of the linearisation techniques used in this dissertation. Two linearisations are described. The first is applicable for the bilinear product of two continuous variables and the technique is illustrated by Quesada and Grossmann (1995). The second is applicable for the bilinear product of a continuous variable and a binary variable and this method is explained by Glover (1975).

B.1 Relaxation Linearisation Technique Described by Quesada and Grossmann (1995)

Let

$xy = \Gamma$, where x and y are both continuous variables in the formulation. In the case of this dissertation *x* is a temperature and *y* is a flowrate.

with each variable displaying the following bounds

$$X^L \leq x \leq X^U \tag{B.1}$$

$$Y^L \leq y \leq Y^U \tag{B.2}$$

each of the bounds expressed as individual equations

$$x - X^L \geq 0 \tag{B.3}$$

$$X^U - x \geq 0 \tag{B.4}$$

$$y - Y^L \geq 0 \tag{B.5}$$

$$Y^U - y \geq 0 \tag{B.6}$$

performing the following multiplications

$$\text{B.3 x B.6} \Rightarrow \Gamma = xy \leq xY^U + X^L y - X^L Y^U \tag{B.7}$$

$$\text{B.4 x B.5} \Rightarrow \Gamma = xy \leq xY^L + X^U y - X^U Y^L \tag{B.8}$$

$$\text{B.3 x B.5} \Rightarrow \Gamma = xy \geq xY^L + X^L y - X^L Y^L \tag{B.9}$$

$$\text{B.4 x B.6} \Rightarrow \Gamma = xy \geq xY^U + X^U y - X^U Y^U \tag{B.10}$$

This linearisation technique is not exact. It does however create a convex solution space where the linearised model is solved. This solution is then used as a starting point for the exact, nonlinear model. To implement this technique Constraints (B.7) to (B.10) are included in the formulation for the linear model and all the bilinear terms are replaced with Γ. If the solution to the nonlinear model equals that of the linear model it can be concluded that a globally optimal solution exists. If the solutions differ then it can be concluded that the solution for the exact, nonlinear model is feasible but not globally optimal.

B.2 Glover (1975) Transformation

Let

$xy = \Omega$, where $x \in \Re$ and $y \in [0,1]$

the following restrictions also hold

$$0 \le \Omega \le 0 \text{ when } y = 0 \tag{B.11}$$

$$x \le \Omega \le x \text{ when } y = 1 \tag{B.12}$$

the following bounds are applicable

$$M^L \le \Omega \le M^U \tag{B.13}$$

then the following constraints on Ω hold

$$M^L y \le \Omega \le M^U y \tag{B.14}$$

$$x - M^U(1-y) \le \Omega \le x - M^L(1-y) \tag{B.15}$$

Constraints (B.14) and (B.15) are then added to the formulation and the bilinear term xy is replaced with the new linearisation variable Ω. This linearisation technique is exact and will therefore result in a globally optimal solution if all of the other constraints in the formulation are linear.