Steam System Network Synthesis Using Process Integration

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Steam boilers are used to generate steam in order to meet the duty requirements of cold process streams. The most common heat exchanger network layout associated with the boiler is a completely parallel design, where the duty of each cold process stream is met by an external hot utility stream. This observation suggests that the flow rate of steam needed for the system can be reduced, while maintaining the required duty, simply by introducing series connections to the layout of the network (Kim, J. K.; Smith, R. Cooling water system design. *Chem. Eng. Sci*. **2001**, *56*, 3641−3658). Phase change of saturated steam to saturated liquid plays a vital role in the targeting method as well as the design of the network layout, since the saturated liquid (and resulting reuse liquid) is also used to meet the cold process duty requirements. A hybrid graphical and mathematical technique for targeting and network synthesis is presented. The main objective of the technique is to reduce steam flow rate without compromising the duty requirements of the process heat exchangers. In order to assess the advantage of a hybrid technique, a case study is used where a steam savings of 29.6% is obtained, compaired with using just saturated steam. We further present a complete mathematical technique to demonstrate the advantage of the graphical targeting concept in solution time improvement.

1. Introduction

Large quantities of water are used by many chemical industries daily. There are two main factors that contribute to the high usage of water, namely, cold water being used as external cold utility as well as the source of steam. Furthermore, fresh water is needed for mass exchange operations.

Linnhoff and Hindmarsh[2] developed a methodology based on pinch analysis. Their method exploited the benefit of maximizing process−process integration, thereby minimizing the duties that have to be met by the external utilities. Furthermore, Linnhoff and Hindmarsh[2] developed a systematic approach to design the heat exchanger network after the maximum process−process integration target has been found. This method, which is mainly used for the reduction of the heat duty for the external utilities has become known as heat integration.

Following the method developed by Linnhoff and Hindmarsh,[2] El-Halwagi and Manousiouthakis[3] developed a method to reduce the contaminant load that needed to be removed by external mass separating agents, by maximizing process−process mass integration. A minimum allowable composition difference value is used throughout between each lean and rich process stream. This is to ensure that the mass exchange driving force is sufficient, as to ensure practical feasibility. However, for each lean process stream the corresponding equilibrium data is required, which is often difficult to obtain.

Wang and Smith[4] developed a method for wastewater minimization in water using operations, where only one lean stream is used, i.e., water. Since only one lean stream is used, there is no need for lean stream scales, as in the case of El-Halwagi and Manousiouthakis'[3] work. The limiting water profile was introduced by Wang and Smith,[4] where the curve incorporates the process constraints directly, i.e., concentration driving forces as well as other constraints due to corrosion limitations, etc. Therefore, several minimum allowable concentration differences are allowed throughout the network. With the method by Wang and Smith,[4] water and wastewater is minimized while the process streams remain unchanged.

Although the heat and mass integration problems have, to a very large extent, been treated in a dichotomous manner, there are cases of particular interest where they occur simultaneously. Only recently have methods been developed to integrate mass and heat simultaneously.[1,5–7] Of particular interest for this paper is the work of Kim and Smith.[1]

In the experimental work done by Bernier,[8] it was shown that as the ratio of cooling water to air flow rate in a cooling tower decreased, the coefficient of performance (COP) of the tower increased. It must be noted that the duty that had to be met by the cooling water remained the same. Therefore, the temperature of the returning cooling water to the tower increases as the flow rate decreased. Furthermore, in the experiment conducted by Bernier,[8] the flow rate of the air in the cooling tower was kept constant, therefore, only the cooling water flow rate was decreased.

Using the understanding that decreasing the cooling water flow rate (for a fixed duty) increases the COP, Kim and Smith[1] developed a systematic method to achieve this effect, for a given cooling water system. The cooling water system includes the cooling tower and the network of operations that use cooling water. The method entails a graphical technique based on the water pinch method.[4] It should be noted that the method developed by Kim and Smith[1] is employed after process−process integration has been applied; therefore, the duty that has to be met by the cooling tower is fixed.

Since the flow rate of the cooling water (external cold utility) can be reduced for a fixed duty, the question arises if the flow rate of steam as an external hot utility can be reduced for a fixed duty, bearing in mind that in the latter case, one has to consider phase change that does not occur in the former case. In this paper, a graphical targeting technique is developed to obtain the minimum flow rate of saturated steam used as an external hot utility. A linear programming (LP) model is developed to help obtain the network layout after the steam flow rate target has been set using a graphical technique. Furthermore,

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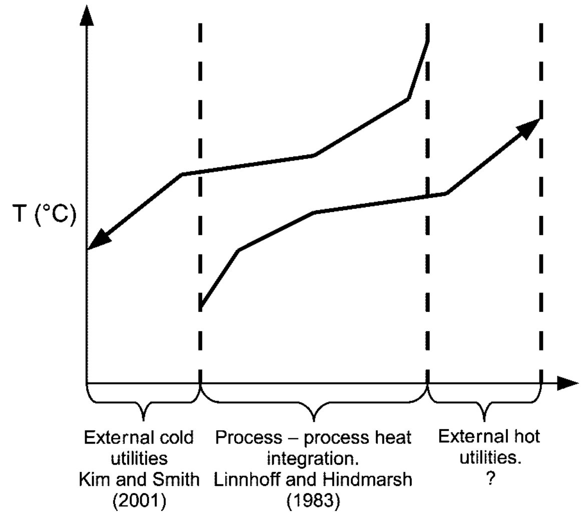

Figure 1. Hot and cold process composite curves.

a mixed integer linear programming model (MILP) is presented in which the target as well as the network layout are obtained simultaneously. The main reason for developing the MILP model is to compare the CPU times between the hybrid graphical technique (which includes the LP model) and the MILP model. Comparing the CPU times between the models illustrates the fact that the hybrid graphical technique required less computing power, which becomes a significant factor when dealing with large-scale problems.

2. Motivation for the Study

Looking at Figure 1, one can see that for hot and cold composite curves, optimization of the process−process section is achieved through heat integration.[2] Furthermore, the external cold utilities section can readily be optimized using the cooling water systems design method of Kim and Smith,[1] particularly when cooling is supplied by cooling water from a cooling tower. However, there is no equal method for the hot utility section in Figure 1.

The network layout of steam that is used as an external hot utility is generally of a parallel design, even for systems that have been designed following pinch analysis. The reason for this is that implicit in pinch analysis is the assumption that all the heat exchangers associated with external duty requirements are directly connected to the energy supply, e.g., steam boiler or hot oil circuit. The main reason for this is that when steam levels and their duties are determined with the grand composite curve, only the latent heat of the steam is used to transfer heat to cold process streams. This concept results in energy losses since some of the condensate still has sufficient energy to supply some of the processes in most instances. Therefore, by allowing condensate reuse at sufficiently high energy/heat levels, one can reduce the overall steam demand. This paper presents a systematic approach to this effect.

Reducing the steam flow rate results in savings in the capital cost for fundamental design, and debottlenecks the boiler for retrofit designs. From Figure 2,[9] it is evident that if the steam flow rate is reduced, the cost of the steam boiler is also reduced significantly. Therefore, in fundamental design, capital costs can be reduced directly when designing for the steam network layout. In retrofit design, the existing steam boiler is debottlenecked, which implies that steam will be available from the existing steam boiler for new heat exchangers which require steam as a hot utility, thereby reducing capital costs.

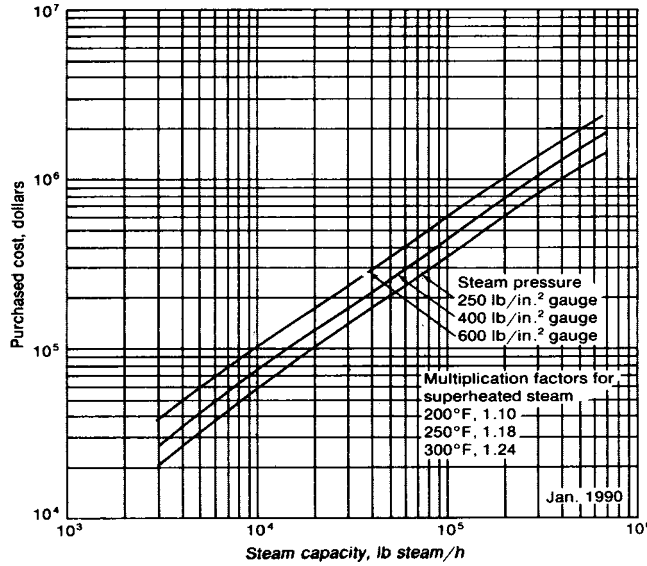

Figure 2. Steam boiler cost.[9]

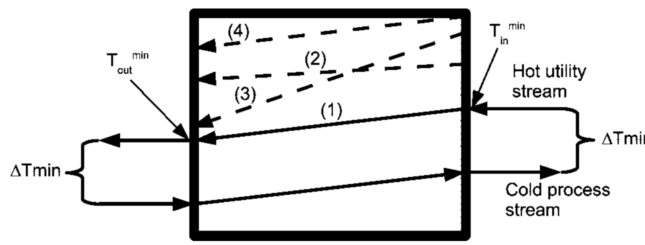

Figure 3. Heat exchanger with a hot utility and cold process stream.

Traditionally, optimization of the steam boiler entails increasing the efficiency of the boiler in isolation from the associated heat exchanger network. However, in this work it is demonstrated that by optimizing the steam system as one entity instead of individual components, better results are obtained in terms of the overall steam demand. Furthermore, the optimization technique for the overall steam system is presented. In the context of this research, the steam system refers to the combination of a steam boiler and heat exchangers that require heating by steam. The aim is to show that the steam flow rate can be significantly reduced by the consideration of an integrated system.

3. Problem Statement

The problem addressed in this paper can be stated as follows. Given

(i) a set of heat exchangers,
(ii) the fixed duties of each heat exchanger,
(iii) the limiting data for each heat exchanger, i.e., minimum inlet/outlet temperatures of the hot utility, and
(iv) the minimum driving force $\Delta T_{\min}$ for the overall network, determine the minimum amount of steam required to satisfy the heat exchanger network, as well as the steam utility network layout without compromising the heat duty requirement. It must be noted that the duties of the cold process streams, which must be met by external hot utilities, are fixed. However, even though the duties of the cold process streams are fixed, different flow rates of the external hot utilities can be obtained to meet the cold process stream duty (Figure 3).

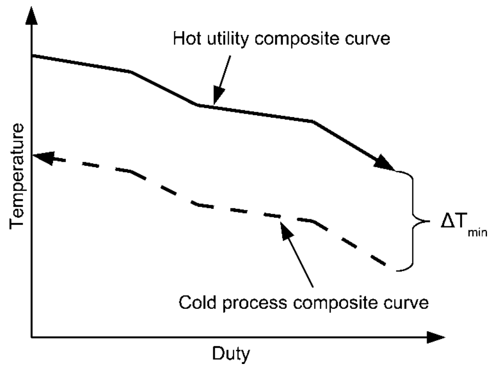

Figure 4. Hot utility composite curve.

4. Methodology

The methodology presented in this paper consists of two steps, i.e., targeting and network design. These two steps are explained in detail below.

4.1. Targeting. Hot utility streams are used to supply energy to the cold process streams in order to increase the temperature of the cold streams. Therefore, thermodynamically, the utility streams must be at a higher temperature than the cold process streams at all times. This is to ensure that there will always be a driving force of energy from the hot utility streams to the cold process streams. When heat exchangers are designed, a minimum temperature difference (ΔT_{min}) between the utility and process stream is specified, as well as the inlet and outlet temperature limits of the cold process stream. Therefore, the temperature range of a utility stream is guided by the ΔT_{min} value for a particular heat exchanger, as seen in Figure 3.

Furthermore, from Figure 3, it is evident that hot utility line number 1 is the border line, since the line is at the limiting inlet and outlet temperatures allowed by ΔT_{min}. The other hot utility lines above the border line are all valid; therefore, the border line divides the heat exchanger into feasible and infeasible regions. The area above the border line is feasible, whereas the area below the line is infeasible.

When there are several heat exchangers each with its own ΔT_{min} value, the maximum ΔT_{min} value is selected as the global minimum temperature difference. The above simplification is not necessary, however, since the individual ΔT_{min} values of the heat exchangers can be used without compromising the targeting and design method. The global ΔT_{min} value, as well as the temperature limits of all the heat exchangers, can be used to construct the hot utility composite curve as seen in Figure 4. The hot utility composite shows visually the minimum allowable temperatures one can target for on a temperature vs duty (TH) diagram. Therefore, the hot utility composite curve divides the TH diagram into two regions (Figure 5), namely, region 1 (the feasible region) and region 2 (the infeasible region). Targeting for the minimum steam flow rate may only be done in the feasible region, since the ΔT_{min} value and heat exchanger temperature limits are not violated. However, targeting in the infeasible region violates the ΔT_{min} value as well as the heat exchanger temperature limits.

Superheated steam at high pressure is generated by steam boilers. In general, the superheated steam is sent to run turbines, the exhaust of which is saturated steam. In our study, we consider the use of saturated steam as a hot utility. The reason for this is that we want to take advantage of the latent heat associated with the saturated steam. Equation 1 shows the energy supplied by the saturated steam.

$$Q_{SS} = \dot{m}_{SS}\lambda \tag{1}$$

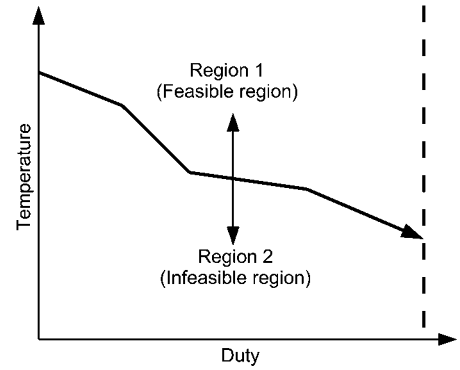

Figure 5. Hot utility composite curve dividing the TH diagram into two regions.

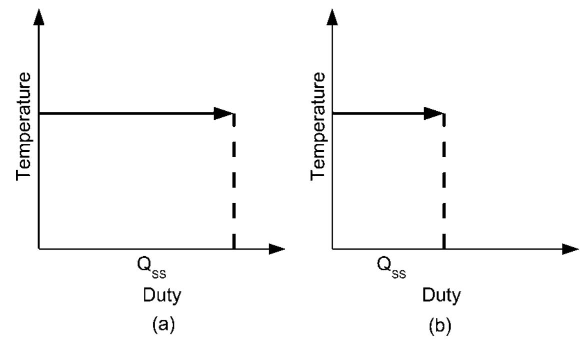

Figure 6. Representation of saturated steam on a TH diagram.

In eq 1, Q_{SS} is the energy supplied by the saturated steam, $\dot{m}_{SS}$ is the saturated steam flow rate, and λ is the latent heat of vaporization of the saturated steam.

Since the saturated steam will phase change into saturated liquid, the temperature at which the saturated steam was supplied will remain the same. Therefore, on a TH diagram, the latent heat of vaporization that the saturated steam transfers is represented as a straight horizontal line, which we will call the saturated steam line. The length of the saturated steam line indicates the quantity of latent heat supplied from the saturated steam. For example, looking at Figure 6, the saturated steam in diagram a supplies more latent heat than the saturated steam in diagram b, since the length of the saturated steam line in a is longer than that of the saturated steam line in b. The value of $\dot{m}_{SS}$ determines the length of the saturated steam line for the same temperature. Therefore, the higher the flow rate of water, the more latent heat is supplied from the saturated steam and vice versa. Consequently, in Figure 6, the flow rate of steam in diagram a is more than that in diagram b.

The saturated liquid resulting from the saturated steam is then used further to supply heat to the remaining cold process streams. The saturated liquid that transferred energy becomes hot liquid below saturation temperature when it exits the heat exchanger. Depending on the energy content, the hot liquid is reused to supply energy to other cold process streams. Therefore, the sensible heat that can be supplied by the saturated and hot liquid is given by eq 2.

$$Q_L = \dot{m}_L c_p (T_s - T_t) \tag{2}$$

In eq 2, Q_L is the energy supplied by the saturated liquid and reuse liquid, c_p is the specific heat capacity of the water, $\dot{m}_L$ is the water flow rate, T_s is the supply temperature, and T_t is the target temperature.

Since the temperature does not stay the same when using saturated liquid and reuse liquid, the saturated liquid and reuse

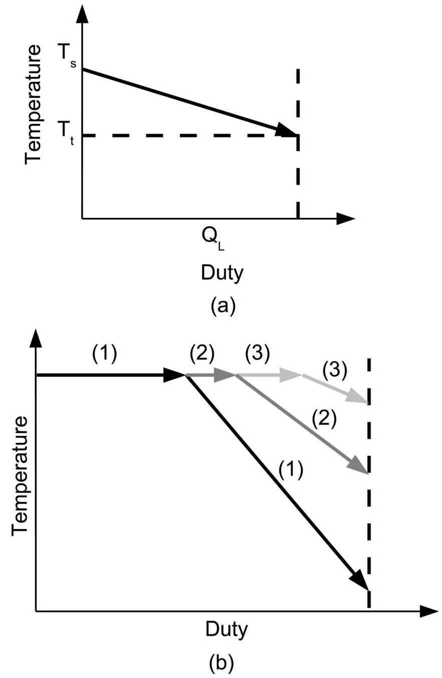

Figure 7. (a) Saturated liquid and reuse liquid represented as a straight line; (b) changing the flow rate of one line influences the other line.

liquid are therefore represented as a diagonal line on a TH diagram, as shown in Figure 7a, which we will call the hot liquid line. The gradient of the hot liquid line is equal to the inverse of the flow rate. This implies that the gradient of the hot liquid line gives an indication of the quantity of the hot liquid flow rate.

Note that, as stated previously, the saturated liquid results from the saturated steam losing its latent heat. Therefore, the flow rate of the hot liquid line is, in essence, the same as the flow rate of the saturated steam line, i.e., $\dot{m}_{SS} = \dot{m}_L$. Since both lines have the same flow rate, by changing the flow rate of one line the other line will adjust to the new flow rate, as shown in Figure 7b. In Figure 7b the flow rate of graph 1 is less than the flow rates of graphs 2 and 3. Also, the flow rate of graph 2 is less than the flow rate of graph 3. Equations 1 and 2 are combined to form eq 3, where the total duty of the cold process streams is met by saturated steam as well as hot liquid. Figure 8 depicts the final diagram when targeting for the minimum flow rate by using saturated steam as well as hot liquid. Q_{SS} is the duty that is supplied by the saturated steam, while Q_L is the duty that is supplied by the saturated and hot liquid. The two duties, Q_{SS} and Q_L, are separated by the border line, which is the line where the saturated steam phase changes into saturated liquid.

$$\dot{m} = \frac{Q}{\lambda + c_p(T_s - T_t)} \tag{3}$$

In eq 3, Q is the energy supplied by the saturated steam and hot liquid and $\dot{m}$ is the combined flow rate of saturated steam and resulting water.

4.2. Graphical Targeting Method. To obtain the minimum flow rate, one must target for a pinch point on the hot utility composite curve. The shape of the hot utility composite curve determines where the pinch point will occur. In some cases, determining where the pinch point will be on the hot utility composite curve can be difficult, as seen in Figure 9. From Figure 9, we do not know exactly where the pinch point on the hot utility composite curve will occur. This is further compli-

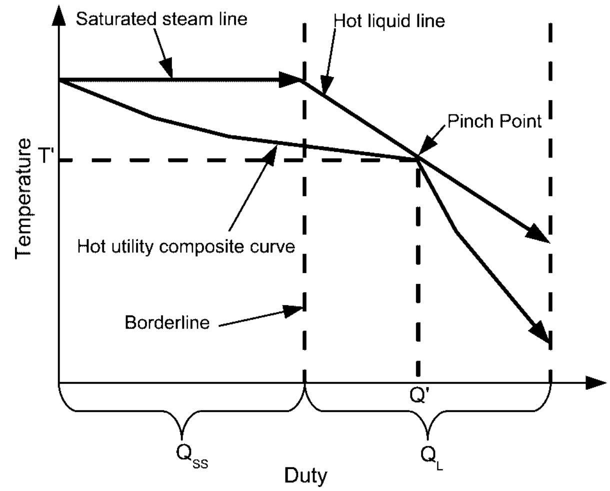

Figure 8. Saturated steam and hot liquid are used to target for the minimum flow rate.

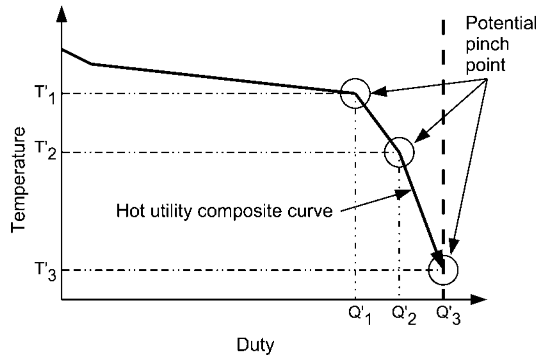

Figure 9. Hot utility composite curve determines where the pinch point will occur.

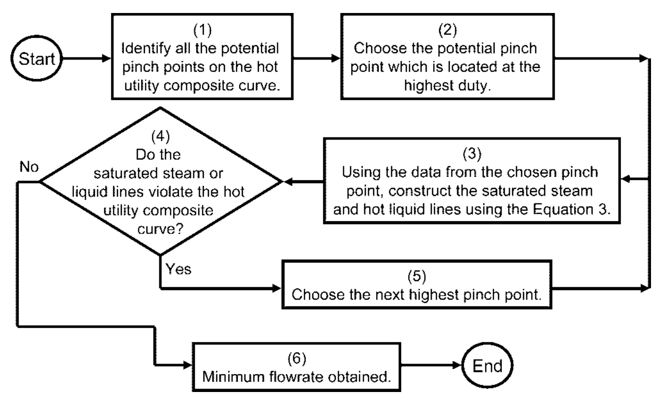

Figure 10. Algorithm to use when targeting for the minimum flow rate.

cated by the fact that saturated steam and hot liquid lines influence each other, as aforementioned. Figure 10 shows an algorithm that can be used when targeting for the minimum flow rate.

The algorithm in Figure 10 ensures that a minimum flow rate will be obtained, regardless of the shape of the hot utility composite curve. The designer first has to identify all the possible pinch point locations on the hot utility composite curve (see Figure 9). Thereafter, the potential pinch point with the highest duty is chosen first, i.e., this will be the point where the hot utility composite curve ends. The reason we start with the point having the highest duty is that the target temperature (T_o) will be at the minimum value. Therefore, the temperature

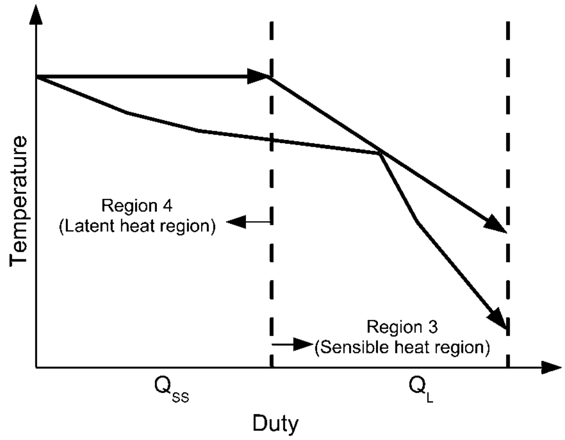

Figure 11. Latent heat and sensible heat region.

difference will be the highest to ensure a minimum flow rate according to eq 3.

The designer uses the data (duty and temperature) from the chosen pinch point to obtain the hot liquid flow rate ($\dot{m}$) via eq 3. With the obtained flow rate, the saturated steam and hot liquid lines can be constructed on the graph. The designer inspects the constructed lines to see whether they violate the hot utility composite curve. Violation in the content of this paper implies that the targeting line goes below the hot utility composite curve, i.e., region 2 in Figure 5.

If there is no violation, then the minimum flow rate has been obtained. However, if the saturated steam and liquid lines violate the hot utility composite curve, the designer has to choose a new potential pinch point. This is done by choosing the potential pinch point on the left-hand side of the current chosen pinch point. The same method as described above is followed with the newly chosen pinch point, until the designer discovers the pinch point that yields the minimum flow rate without any violation.

4.2.1. Network Synthesis. In order to satisfy the target set using the graphical method described above, we need to determine the corresponding network layout of the HEN. We know that saturated steam and saturated liquid are used in the HEN. Furthermore, we know we must first use the available energy from the saturated steam before we use the energy available from the saturated liquid and hot liquid below the saturation point. In this section, the method to obtain the network layout, after using the graphical technique to target for the minimum flow rate, is shown.

Network Synthesis in the Latent Heat Region. To help better understand the layout of the HEN, Figure 11 shows how the TH diagram is divided into two additional regions. Recall that the hot utility composite curve divided the diagram into the feasible and infeasible region (Figure 5). The border, where saturated steam transforms to saturated liquid, divides the diagram into region 3 (the sensible heat region) and region 4 (the latent heat region). In region 3, heat transfer takes place through sensible heat, whereas in region 4 heat transfer involves latent heat, i.e., phase change. By exploiting the structure of Figure 11, a HEN that meets the target steam flow rate can be designed.

The network layout in region 4 will always be a parallel layout, since latent heat is transferred in this area. This implies that the layout for all the heat exchangers that are situated in this region will be parallel, thereby reducing the complexity of solving for the HEN layout. More accurately, all the heat exchangers in this region are directly connected to the saturated steam boiler. However, the network layout in region 3 can be parallel and/or series since sensible heat is transferred in this area. Determining the network layout in region 3 is certainly not as straightforward as in region 4. Consequently, a LP model was developed in order to obtain a network layout in the sensible heat region to meet the targeted flow rate.

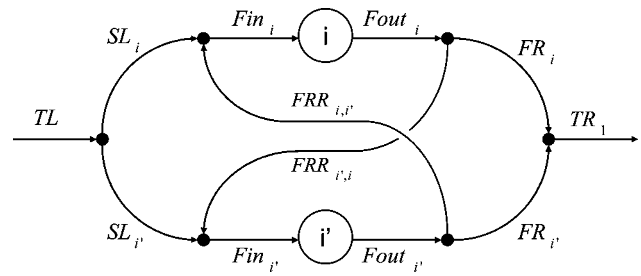

Figure 12. Superstructure for the sensible heat region.

Network Synthesis in the Sensible Heat Region. Figure 12 shows the superstructure for the network design in the sensible heat region. The superstructure shows that every inlet stream to heat exchanger i is comprised of a recycle/reuse stream from other heat exchanger(s) i', which is below saturation temperature, and saturated liquid from saturated steam. Similarly, the outlet stream from said heat exchanger i is comprised of recycle/reuse streams to other heat exchangers i' and return condensate stream to the boiler.

Constraints. Equation 4 stipulates that the total flow rate of saturated liquid is equal to the amount used by heat exchangers in the sensible heat region. Equation 5 stipulates that the total flow rate of the return condensate stream to the boiler is comprised of the return streams from each heat exchanger. The mass coming into the system must be equal to the mass going out of the system if no accumulation occurs, as shown in eq 6. The slack variables S_j^+ and S_j^- are introduced to account for deviations which could arise from targeting errors. Ideally, these slack variables should be zero, hence the objective function, eq 7.

$$\mathrm{TL} + S_1^+ - S_1^- = \sum_{i \in I} \mathrm{SL}_i \quad \forall\, i \in I \tag{4}$$

$$\mathrm{TR}_1 + S_2^+ - S_2^- = \sum_{i \in I} \mathrm{FR}_i \quad \forall\, i \in I \tag{5}$$

$$\mathrm{TL} = \mathrm{TR}_1 \tag{6}$$

$$\mathrm{Min(OBJ)} = \sum_{j=1,2} \left(S_j^+ + S_j^-\right) \tag{7}$$

As aforementioned, the total inlet flow rate to heat exchanger i is equal to the saturated liquid entering the unit, plus the summation of reused hot liquid from other heat exchangers i', as given in eq 8. Furthermore, the total outlet flow rate from heat exchanger i is equal to the returned liquid to the steam boiler, plus the summation of reused hot liquid to other heat exchangers i', as given in eq 9. The conservation of mass for heat exchanger i is given by eq 10.

$$F(\mathrm{in})_i = \mathrm{SL}_i + \sum_{i' \in I} \mathrm{FRR}_{i',i} \quad \forall\, i,\ i' \in I \tag{8}$$

$$F(\mathrm{out})_i = \mathrm{FR}_i + \sum_{i' \in I} \mathrm{FRR}_{i,i'} \quad \forall\, i,\ i' \in I \tag{9}$$

$$F(\mathrm{in})_i = F(\mathrm{out})_i \quad \forall\, i \in I \tag{10}$$

Equation 11 ensures that total inlet flow rate to heat exchanger i does not exceed the design flow rate. To obtain the maximum

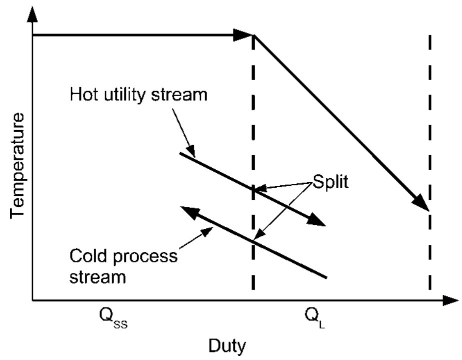

Figure 13. Splitting of cold process stream.

flow rate of hot liquid that can be supplied to a heat exchanger i, eq 12 is used.

$$F(\mathrm{in})_i \le F(\mathrm{in})_i^{\mathrm{U}} \quad \forall\, i \in I \tag{11}$$

$$F(\mathrm{in})_i^{\mathrm{U}} = \frac{Q_i}{c_{\mathrm{p}}\left(T(\mathrm{in})_i^{\mathrm{L}} - T(\mathrm{out})_i^{\mathrm{L}}\right)} \quad \forall\, i \in I \tag{12}$$

In a situation where local recycle is forbidden, eq 13 is relevant.

$$\mathrm{FRR}_{i,i'} = 0 \quad \forall\, i, i' \in I \quad i' = i \tag{13}$$

Equation 14 gives the energy balance across heat exchanger i. The inlet temperature into heat exchanger i is given by eq 15.

$$Q_i = F(\mathrm{in})_i c_{\mathrm{p}}(T(\mathrm{in})_i - T(\mathrm{out})_i) \quad \forall\, i \in I \tag{14}$$

$$T(\mathrm{in})_i = \frac{\sum_{i' \in I}(\mathrm{FRR}_{i',i} T(\mathrm{out})_{i'}) + \mathrm{SL}_i T}{F(\mathrm{in})_i} \quad \forall\, i, i' \in I \tag{15}$$

Equations 4–15 constitute the overall mathematical model. However, eqs 14 and 15 contain nonconvex bilinear terms which entail computational difficulties. As a result, it is necessary to try and linearize eqs 14 and 15, as described below.

Both eqs 14 and 15 have the same bilinear term, namely, $F(\mathrm{in})_i T(\mathrm{in})_i$. Equation 15 also contains the bilinear term $\mathrm{FRR}_{i',i} T(\mathrm{out})_i$. Therefore, by substituting eq 15 in eq 14, the bilinear term $F(\mathrm{in})_i T(\mathrm{in})_i$ is eliminated. Furthermore, by setting the outlet temperature of the hot utility streams of the heat exchangers equal to the minimum outlet temperature, the problem is reduced to a LP problem. Savelski and Bagajewicz[10] proved this to be necessary condition for minimizing flow rate, albeit in wastewater minimization. Therefore, by using the same analogy, at an optimal solution in the reduction of the hot utility flow rate, the outlet temperature of the hot utility streams will be at the minimum value. Therefore, eqs 14 and 15 are replaced with eq 16 in the model. Consequently, the overall model without eqs 14 and 15 is an LP model.

$$Q_i = c_{\mathrm{p}} \sum_{i' \in I} (\mathrm{FRR}_{i',i} T(\mathrm{out})_{i'}^{\mathrm{L}}) + c_{\mathrm{p}} \mathrm{SL}_i T - F(\mathrm{in})_i c_{\mathrm{p}} T(\mathrm{out})_i^{\mathrm{L}} \quad \forall\, i, i' \in I \tag{16}$$

The objective function is still eq 7.

4.2.2. Stream Splitting. When targeting for the minimum flow rate, it is most likely that a cold process and hot utility stream will be split between the latent heat and sensible heat regions, as seen in Figure 13. This implies that a fraction of the duty of the split cold process stream will have to be met by saturated steam, while the rest is met by hot liquid.

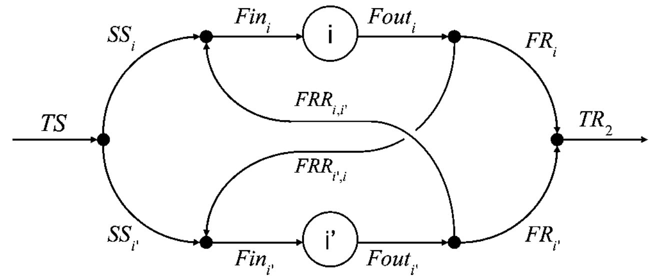

Figure 14. Superstructure for the latent and sensible heat region.

A split increases the capital cost, since two heat exchangers with additional piping have to be used instead of one. A split can be eliminated though, by increasing the water flow rate. As the water flow rate increases, the border between the latent heat and sensible heat region moves to the right on the TH diagram. Therefore, the flow rate can be increased until the total duty of the split stream lies in the latent heat region. We can only increase the water flow rate to eliminate a split, since we have already targeted for the minimum flow rate, which implies that we are at the pinch point. Lowering the water flow rate further will result in violating the $\Delta T_{\min}$ value.

5. Simultaneous Targeting and Synthesis of the Steam System

In this section we demonstrate that targeting and design can be achieved simultaneously using a mathematical approach. However, the disadvantage of using a pure mathematical model for targeting and design is that the solution becomes a black box approach for the user; i.e., the user enters the required data and the program yields a solution. Therefore, the user does not have an in-depth understanding of the problem and design. Furthermore, complex problems can require a lot of CPU time to solve, since the size and complexity of the problem increases sharply with the number of heat exchangers. The complexity mainly arises from the inherent binary dimensions in the problem.

One of the benefits of using a mathematical model to target and design the network simultaneously is that solutions for small problems can be obtained in very short times. Using the graphical approach for targeting by hand takes time and can contain errors from reading off the graph. Furthermore, introducing additional constraints to the targeting and design is easier to implement with a mathematical model than with the graphical approach.

Figure 14 shows the superstructure for the network design in the latent and sensible heat region. It is worth noting that although this superstructure has the same structure as that shown in Figure 12, the variables involved are rather different.

Constraints. To obtain the maximum flow rate of saturated steam or recycle/reuse hot liquid that can be supplied to a heat exchanger i, eqs 17 and 18 are used respectively.

$$\mathrm{SS}_i^U = \frac{Q}{\lambda} \quad \forall\, i \in I \tag{17}$$

$$\mathrm{FRR}_i^{\mathrm{U}} = \frac{Q_i}{c_{\mathrm{p}}\left(T(\mathrm{in})_i^{\mathrm{L}} - T(\mathrm{out})_i^{\mathrm{L}}\right)} \quad \forall\, i \in I \tag{18}$$

Equation 19 stipulates that the total flow rate of saturated steam from the steam boiler, is equal to the overall amount of saturated steam used by each heat exchanger i, while eq 20 stipulates that the total amount of hot liquid leaving all heat exchangers i, to return to the steam boiler, is equal to the total steam input into the HEN. Equation 21, therefore, gives the mass conservation for the whole superstructure.

$$\mathrm{TS} = \sum_{i \in I} \mathrm{SS}_i \quad \forall\, i \in I \tag{19}$$

$$\mathrm{TR}_2 = \sum_{i \in I} \mathrm{FR}_i \tag{20}$$

$$\mathrm{TS} = \mathrm{TR}_2 \tag{21}$$

The total inlet flow rate to heat exchanger i is equal to either saturated steam entering the unit or the total reused hot liquid from other heat exchangers i', as given in eq 22. Furthermore, the total outlet flow rate from heat exchanger i is equal to the returned liquid to the steam boiler, plus the summation of reused hot liquid to other heat exchangers i', as given in Equation 9. The conservation of mass for heat exchanger i is given by Equation 10.

$$F(\mathrm{in})_i = \mathrm{SS}_i + \sum_{i' \in I} \mathrm{FRR}_{i',i} \quad \forall\, i,\, i' \in I \tag{22}$$

The hot liquid that is reused in heat exchanger i' from heat exchanger i is either saturated liquid (if heat exchanger i was supplied by saturated steam) or below saturation point (if heat exchanger i was supplied by hot liquid), as given by eq 23.

$$\mathrm{FRR}_{i,i'} = \mathrm{SL}_{i,i'} + L_{i,i'} \quad \forall\, i,\, i' \in I \tag{23}$$

The saturated liquid that is reused from heat exchanger i to i' cannot exceed the flow rate of saturated steam that entered heat exchanger i, as seen in eq 24. Likewise, eq 25 states that the liquid that is reused from heat exchanger i to i' cannot exceed the flow rate of hot liquid that entered heat exchanger i .

$$\sum_{i' \in I} \mathrm{SL}_{i,i'} \le \mathrm{SS}_i \quad \forall\, i \in I \tag{24}$$

$$\sum_{i' \in I} L_{i,i'} \le \sum_{i' \in I} \mathrm{FRR}_{i',i} \quad \forall\, i,\, i' \in I \tag{25}$$

Equation 26 ensures that total saturated steam flow rate to heat exchanger i does not exceed the maximum flow rate obtained in eq 17. Furthermore, eq 27 ensures that the total flow rate of hot liquid to heat exchanger i does not exceed the maximum flow rate obtained in eq 18.

$$\mathrm{SS}_i \le \mathrm{SS}_i^{\mathrm{U}} y_i \quad \forall\, i \in I \tag{26}$$

$$\sum_{i' \in I} \mathrm{FRR}_{i',i} \le \mathrm{FRR}_i^{\mathrm{U}} x_i \quad \forall\, i,\, i' \in I \tag{27}$$

To ensure that heat exchanger i does not recycle hot liquid back to itself, eq 28 is implemented.

$$L_{i,i'} = 0 \quad \forall\, i,\, i' \in I \quad i' = i \tag{28}$$

Equation 29 gives the energy balance across heat exchanger i, which is supplied by either saturated steam (Q_i^{SS}) or hot liquid (Q_i^{HL}). The duty that is supplied by the saturated steam is given by eq 30, while the duty supplied by the hot liquid is given by eq 31.

The duty in eq 31 is obtained by the difference of the energy that comes into the heat exchanger i from the energy leaving the heat exchanger. The energy coming into heat exchanger i is comprised of saturated liquid and hot liquid reused from other heat exchangers and their respective outlet temperatures. The duty that is exchanged in the heat exchanger is the difference between the energy content of the exit stream and inlet streams into the heat exchanger.

Table 1. Cold Process Stream Data for the Example

cold process stream	T_{target} (°C)	T_{supply} (°C)	duty (kW)
1	45	25	135
2	45	25	320
3	215	209	3620
4	185	79	12980
5	207	207	1980
6	70	44	635
7	70	44	330
total			20000

Table 2. Hot Utility Data for the Example

hot utility stream	$T_{\mathrm{target}}^{\mathrm{min}}$ (°C)	$T_{\mathrm{supply}}^{\mathrm{min}}$ (°C)	duty (kW)
1	35	55	135
2	35	55	320
3	219	225	3620
4	89	195	12980
5	217	217	1980
6	54	80	635
7	54	80	330
total			20000

$$Q_i = Q_i^{\mathrm{SS}} + Q_i^{\mathrm{HL}} \quad \forall\, i \in I \tag{29}$$

$$Q_i^{\mathrm{SS}} = \mathrm{SS}_i \lambda \quad \forall\, i,\, i' \in I \tag{30}$$

$$\mathrm{Q}_i^{\mathrm{HL}} = \sum_{i' \in I} (c_{\mathrm{p}} \mathrm{SL}_{i',i} T) + \sum_{i' \in I} (c_{\mathrm{p}} L_{i',i} T(\mathrm{out})_{i'}) - (c_{\mathrm{p}} F(\mathrm{out})_i T(\mathrm{out})_i) \quad \forall\, i, i' \in I \tag{31}$$

In order to ensure that the duty of heat exchanger i is not split between the sensible and latent heat regions, eq 32 is implemented. Equation 32 ensures that heat exchanger i is supplied by either saturated steam or hot liquid, but not both.

$$y_i + x_i \le 1 \tag{32}$$

If we want to allow a maximum of one split to occur, eqs 33 and 34 are introduced into the formulation. Equations 33 and 34 ensure that a maximum of one heat exchanger split can occur. Therefore, the duty of only one heat exchanger i can be supplied by saturated steam as well as hot liquid, while the remaining heat exchangers duties may only be supplied by either saturated steam or hot liquid. This implies that the heat exchanger, in which the split occurs, will lie in the latent heat as well as in the sensible heat region. In physical terms, this implies heat exchanger i will be split into two heat exchangers, i_1 and i_2, supplied by saturated steam and hot liquid, respectively.

$$\sum_{i \in I} y_i + \sum_{i \in I} x_i \ge |i| \tag{33}$$

$$\sum_{i \in I} y_i + \sum_{i \in I} x_i \le |i| + 1 \tag{34}$$

Equation 31 is the only nonlinear equation. To transform eq 31 to a linear equation, the outlet temperatures of the heat exchangers are set to their minimum values in accordance with the observation made in Section 4.2.1 of this paper. Therefore, eq 31 becomes linear as seen in eq 35, which implies that we obtain an overall MILP model.

$$Q_i^{\mathrm{HL}} = \sum_{i' \in I} (c_{\mathrm{p}} \mathrm{SL}_{i',i} T) + \sum_{i' \in I} \left(c_{\mathrm{p}} L_{i',i} T(\mathrm{out})_{i'}^{\mathrm{L}}\right) - \left(c_{\mathrm{p}} F(\mathrm{out})_i T(\mathrm{out})_i^{\mathrm{L}}\right) \quad \forall\, i,\, i' \in I \tag{35}$$

The objective function is the minimization of the total flow rate of saturated steam (TS) supplied to the heat exchangers,

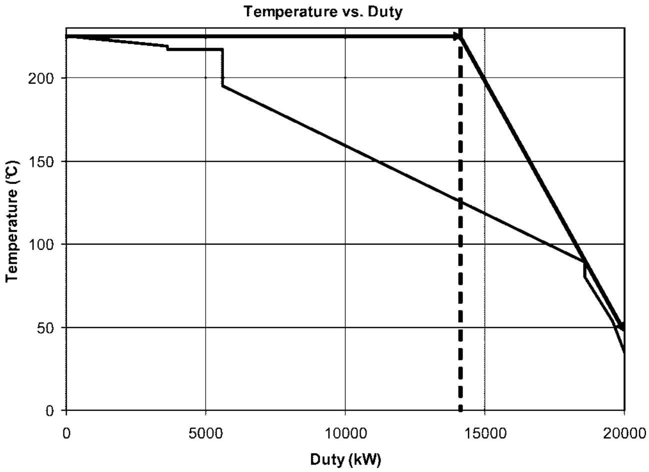

Figure 15. Graphical targeting for the minimum water flow rate.

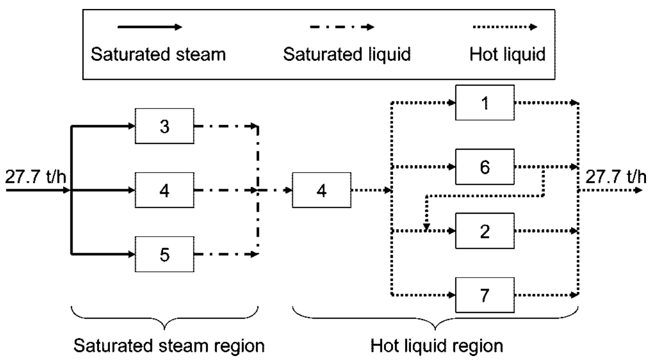

Figure 16. Heat exchanger network layout using the graphical method and LP model.

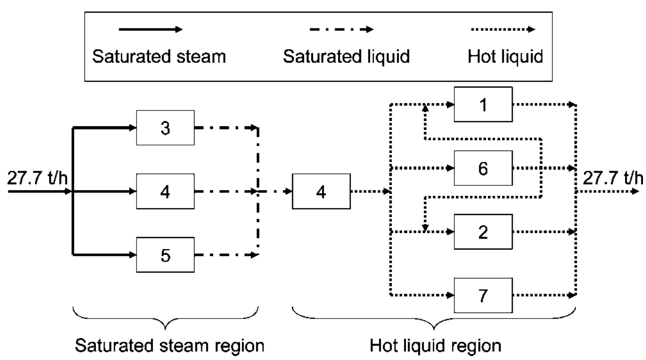

Figure 17. Heat exchanger network layout using the MILP model.

without violating any of the operational constraints within the network as given in eq 36.

$$\mathrm{Min}(Z) = \sum_{i \in I} \mathrm{TS}_i \tag{36}$$

6. Case Study

In order to demonstrate the applicability of the developed hybrid graphical targeting and LP design method, as well as the MILP model, a case study from a South African petrochemical company is presented. The process produces high-value phenols, cresols, and xylenols and includes liquid−liquid extraction. The reboilers of the columns used for separation throughout the process require saturated steam as an external hot utility.

The cold process stream data that will be used for the case study is given in Table 1. Saturated steam is provided at 225 °C (25.5 bar) with a latent heat capacity of 1834.3 kJ/kg. The specific heat capacity of the resulting saturated liquid is 4.30 kJ/kg °C. The global ΔT_{min} value is 10°C, therefore, the limiting hot utility data is obtained as seen in Table 2.

With the use of the graphical technique, the hot utility data in Table 2 is used to construct a hot utility composite curve on a TH diagram. Thereafter, the pinch point must be obtained by using the developed algorithm in Figure 10. Figure 15 shows the pinch point obtained by using the algorithm, corresponding to the minimum steam flow rate. Furthermore, from Figure 15, the saturated steam has to supply 14 MW of energy, while the saturated liquid and reuse liquid has to supply 5.9 MW of energy.

If only saturated steam was used as a hot utility, i.e., assuming a parallel design, the flow rate would be 39.2 t/h. However, by the use of the methodology described above, the flow rate needed is only 27.6 t/h, reducing the original flow rate by 29.6%. After targeting for the minimum flow rate, the final network layout was obtained by using the LP model, as seen in Figure 16. The CPU time for the LP model is 0.001 s. The computer that was used to solve the model was a Pentium 3.4 GHz with 1 GB of RAM, and the solver used was GAMS CPLEX 9.

The MILP model was also used to target for the minimum flow rate, as well as obtaining a network layout, as seen in Figure 17. The same flow rate of 27.6 t/h was achieved with the MILP model, however, the network layout obtained from the MILP model slightly differs from that obtained by using the graphical targeting and LP model. This indicates the known fact that for the optimal flow rate several different network layouts can be obtained, each one making the minimum flow rate possible without violating any of the constraints. The CPU time of the MILP model is 0.062 s. The computer that was used to solve the model was a Pentuim 3.4 GHz with 1 GB of RAM and the solver used was GAMS CPLEX 9.

It is evident when comparing the CPU times of the LP and MILP models that the LP model solves faster than the MILP model. This is due to numerous reasons, one being that the MILP model contains binary variables, which increases the complexity of the structure of the problem. Furthermore, the MILP model has to obtain the minimum steam flow rate, whereas the minimum steam flow rate is already specified in the LP model through the graphical targeting technique. Also, the MILP model has to cater for the latent as well as the sensible heat regions, hence, an increase in the size of the problem. On the other hand, the LP model only has to cater for the sensible heat region. As the amount of data for a problem increases (more process streams), the CPU time of the MILP model tends to increase significantly, due to binary variables. Therefore, the hybrid graphical targeting and LP model is very viable for complex problems, since targeting and the network layout in the latent heat region is first obtained before the LP model is used to obtain the network layout in the sensible heat region.

7. Conclusions

A novel hybrid graphical targeting and LP design technique using pinch analysis has been developed in this paper, where the minimum saturated steam (external hot utillity) flow rate is obtained by exploiting the phase change of saturated steam. Since the designer has to cater for two phases on the graph, an algorithm was developed to assist the designer in obtaining the pinch point without violating the hot utility composite curve.

With regards to the design of the network after targeting for the minimum flow rate, it was found that the network layout in the latent heat region will always be parallel. However, as obaining the network layout in the sensible heat region could be quite complex, an LP model was developed to obtain the network layout.

An MILP model was developed where targeting and network design were achieved simultaneously. However, the network layout obtained by the MILP differs slightly from that obtained by the LP model, suggesting multiple network layouts are possible for the same flow rate target.

An case study using real-life data illustrated the applicability of the developed methodology. The original steam flow rate was reduced by 29.6%, showing that saturated steam as an external hot utility can be reduced for a given fixed duty.

Worthy of mention, however, is that the proposed steam reduction procedure does not consider the efficiency of the boiler, which could be affected by the returned condensate temperature. Such continuation of the proposed investigation would be an interesting future step in the design of optimal steam system networks.

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Nomenclature

Sets

I = heat exchangers

Parameters

Q_i = duty of the heat exchanger i (kW)

$T(\text{out})_i^{\text{L}}$ = minimum outlet temperature out of heat exchanger i (°C)

$T(\text{in})_i^{\text{L}}$ = minimum inlet temperature to heat exchanger i (°C)

FRR_i^{U} = maximum flow rate of hot liquid to heat exchanger i (kg/s)

SS_i^{U} = maximum flow rate of saturated steam to heat exchanger i (kg/s)

Scalars

c_p = specific heat capacity of the hot liquid (kJ/kg °C)

T = temperature of the saturated liquid (°C)

λ = latent heat of the saturated steam (kJ/kg)

TR_1 = total flow rate of hot liquid returning to the steam boiler (kg/s)

TL = total saturated liquid supplied to the heat exchangers (kg/s)

Binary Variables

x_i = 1 if heat exchanger i is supplied by hot liquid, 0 otherwise

y_I = 1 if heat exchanger i is supplied by saturated steam, 0 otherwise

Continuous Variables

$F(\text{in})_i$ = total flow rate entering heat exchanger i (kg/s)

$F(\text{out})_i$ = total flow rate leaving heat exchanger i (kg/s)

$\text{FRR}_{i,i'}$ = liquid reused from heat exchanger i to heat exchanger i' (kg/s)

$L_{i,i'}$ = hot liquid reused from heat exchanger i to heat exchanger i' (kg/s)

$\text{SL}_{i,i'}$ = saturated liquid reused from heat exchanger i to heat exchanger i' (kg/s)

SL_i = flow rate of saturated liquid entering heat exchanger i (kg/s)

SS_i = flow rate of saturated steam entering heat exchanger i (kg/s)

FR_i = flow rate of hot liquid returning to the steam boiler from heat exchanger i (kg/s)

$T(\text{in})_i$ = temperature of the hot liquid entering heat exchanger i (°C)

$T(\text{out})_i$ = temperature of the hot liquid leaving heat exchanger i (°C)

Q_i^{SS} = duty of heat exchanger i supplied by saturated steam (kW)

Q_i^{HL} = duty of heat exchanger i supplied by hot liquid (kW)

TR_2 = total flow rate of hot liquid returning to the steam boiler (kg/s)

TS = total flow rate of saturated steam supplied to the heat exchangers (kg/s)

S_1^+ = slack variable for undershooting TL (kg/s)

S_1^- = slack variable for overshooting TL (kg/s)

S_2^+ = slack variable for undershooting TR_1, (kg/s)

S_2^- = slack variable for overshooting TR_1 (kg/s)

OBJ = total error allowance (kg/s)

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