Level-by-Level Flowsheet Synthesis Methodology for Thermal System Design

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A new level-by-level methodology for flowsheet synthesis is presented. In the first level, an initial superstructure is constructed to form a master model. Optimization of the master model gives an initial optimal flowsheet, which indicates major structural features of a design, interconnections between subsystems and also optimal main parameters. In the second level this initial flowsheet is analyzed using exergy analysis to identify relevant modification options. The promising options are used to update the initial superstructure, which is optimized to obtain an improved design. The results from the above design process determine the flowsheet of overall system and provide targets for design of subsystems. Design of subsystems forms the third design level. The predetermined targets may not be met in the third level due to use of simplistic models in the previous design levels. The master model is updated and then optimized again. The overall design process terminates when designs for subsystems can meet the design targets determined by the master model.

Introduction

The main objective in process synthesis is to generate the most economic flowsheet structure and parameters to satisfy given requirements. The most difficult part of a design problem is determination of an optimal structure among a large number of possible alternatives. Due to strong interactions between subsystems, decomposition of an overall problem into design of several subproblems separately will lead to nonoptimal solutions.

Essentially, the design process is sequential. It starts with determination of objectives and high-level conceptual designs, which are evaluated technically and economically in feasibility analysis. Then flowsheet synthesis is carried out, in which several alternatives are obtained, and they are analyzed and evaluated. As a result, the most viable flowsheet is determined. This flowsheet will form a basis for a more detailed design of individual elements in the flowsheet. The flowsheet synthesis stage is of paramount importance, since the decisions made during this stage will dictate the overall viability of a design. It is in this stage that major trade-offs, including thermal efficiency, capital cost, and operational flexibility, are determined. Optimization of these trade-offs determines the targets or requirements for subsequent design of subsystems. Additionally, as the design process moves toward more detailed design, the problem becomes better defined and more options appear. Thus it is logical to reformulate the problem definition and start the design process again at a higher level.

To reflect the nature of a design process, a synthesis methodology for thermal systems design is presented here. The methodology combines the physical insights of thermodynamic analysis, the level-by-level strategy of hierarchical design procedure, and the computational power of mathematical programming. The new method consists of three design levels and one updating mechanism. In the first level, major structural issues and parameters are considered, and they are optimized using a simplified overall plant model, namely the master model. In this model subsystems are modeled as aggregate models (that is, in low level of detail) and an initial flowsheet is generated by optimizing this model. In the second level, thermodynamic analysis is employed to identify beneficial structural changes, which are then implemented to the master model superstructure, subject to further optimization. This process can continue until no desir-

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able improvement to the flowsheet can be made. As a result, solution from the master model provides design targets, parameter constraints, and the relevant structural options to be considered in the third level of design, which is the conceptual design of each subsystem. This stage of design can be carried out using tools specifically developed for each subsystem. If for any reason the conceptual design cannot meet the specifications suggested by the master model, an updating mechanism is employed, which either introduces additional constraints or includes alternative practical designs in the master model for further optimization and analysis.

Review of Flowsheet Synthesis Techniques

Hierarchical decomposition

Douglas (1988) introduced a hierarchical procedure for chemical process design, where heuristic rules are used to guide the search direction. At each decision step, more detail is added to the flowsheet. The steps can be summarized as follows:

Step 1. Batch vs. continuous.
Step 2. Input–output structure of the flowsheet.
Step 3. Recycle structure of the flowsheet.
Step 4. Separation system synthesis.
Step 5. Heat recovery network.

The decision hierarchy is useful in guiding a design from a simple to a more detailed flowsheet. In the early stages of the design process the amount of data available is small, and gradually more detail is generated as the design process goes further. However, the major limitation is that the procedure cannot take into account the interactions between different design levels due to its sequential nature.

Mathematical programming

To simultaneously explore the benefits of both parameter and structural changes, and to address complex trade-offs, the best way would be to build a general superstructure including all possible options and then optimize it to give an optimal design. Mixed-integer linear programming (MILP) and mixed-integer nonlinear programming (MINLP) formulations have been used for general flow-sheet synthesis (Grossmann, 1985) and for synthesizing utility systems for process industries (Papoulias and Grossmann, 1983; Grossmann and Kravanja, 1995; Bruno et al., 1998).

However, the general superstructure approach can result in a very complex combinatorial problem, since too many candidate options need to be considered, which will cause computational difficulty. Many design options are either technically or economically infeasible and many of them are redundant options.

Combined methods

Mizsey and Fonyo (1990) proposed a process synthesis methodology, which attempts to combine hierarchical decomposition (Douglas, 1988) with mathematical optimization techniques. The hierarchical approach is adopted first where heuristics are used to generate initial flowsheets. Mathematical optimization of continuous variables are performed on the initial flow sheets with a fixed structure to generate a detailed design.

Daichendt and Grossmann (1997) introduced an approach for flowsheet synthesis, where hierarchical decomposition (Douglas, 1988) is used together with MINLP programming in both flowsheeting and detailed design stages. At each stage, the entire flowsheet is optimized using a multilevel tree search. Since the cost increases monotonically, at each level uneconomic branches of the search tree can be removed, which reduces the computational task.

Kravanja and Grossmann (1997) presented a multilevel-hierarchical approach to MINLP synthesis of flowsheets, which in general utilizes a similar approach as the one by Daichendt and Grossmann (1997). The flowsheet is postulated at different levels of detail, depending upon whether the model is used for targeting or design. The overall synthesis strategy follows hierarchical strategy starting from reactor design to separation system design and so on.

The main advantage of these approaches when compared to solving a single large MINLP problem is that smaller subproblems are solved at each level of optimization. However, the main limitation of these approaches is that they do not fully take into account interactions between the upstream subsystems and the detailed models. This is due to the fact that during the detailed design of a subsystem, the structures of other subsystems upstream of the subsystem are already fixed.

Problem Definition

The design problem to be addressed in this article is stated as follows. Given the building blocks of the thermal system, for example, optional fuels, boilers, turbines, and heat exchangers, the objective is to generate a good conceptual design for an overall plant, maximizing annual profit while satisfying the given design constraints.

The main characteristics of the problem and the challenges for solving it are discussed in more detail as follows.

Interactivity

The major subsystems of a thermal system include the boiler system, turbine system, and feedwater heating system, which are strongly interlinked. This interconnectivity makes it difficult to decompose the overall design problem into separate subproblems without sacrificing the solution quality. To maintain optimality, all the subsystems must be optimized simultaneously, but this will inevitably increase the size of the problem, and hence the difficulty of solving the problem since the design task of any subsystem is a complex task in itself.

Structural complexity

Thermal system design is a flowsheet synthesis problem, which includes location and sizing of boilers, heat recovery steam generators, steam and gas turbines, and heat exchangers. The candidate units can have numerous possible connections among them. The number of combinations can grow exponentially when a design problem becomes more complicated. Since the tendency in thermal system design is to move toward more efficient and often more complex cycles, the combinatorial problem becomes more acute.

Design options and constraints

In the early stages of design, it is virtually impossible to fully specify both all the possible design options and constraints to be considered. This problem has been recently addressed by Stephanopoulos (1998). For example, constraints involving control and complexity issues cannot be properly addressed before the more detailed design of the subsystems. Also if too many constraints are imposed at the initial flowsheeting stage, the solution space may be unnecessarily restricted, and promising designs can be missed. Constraints therefore should be introduced only when they are considered to be relevant.

New Methodology

Essential features

1. The method should take advantage of the insights provided by thermodynamics for suggesting improvements, while simultaneously utilizing the power of mathematical programming for addressing complex interactions and trade-offs.
2. The method should provide a transparent design procedure. This is essential to enable a designer to monitor and control the design process. This can be achieved by adopting a stepwise methodology in the spirit of hierarchical methods. The steps provide a logical procedure that is easy to follow. The designer can interact with the design process at any time to introduce or reject design changes based on practical constraints.
3. Last but not least, the method should be able to reduce the complexity of the problem by considering only relevant design options without sacrificing optimality, so that the design method can be applied to large practical problems.

Overview of the methodology

Based on the benefits and the shortcomings of reviewed approaches, a new methodology for the design of thermal systems is proposed (Figure 1).

In the beginning the amount of available data is very limited, so we only consider major design options, for example, the number of steam generation levels, the basic configuration of the steam system, and gas turbine size. An initial superstructure captures the major design options, and process units are modeled using aggregate models and simplified cost equations. Optimization of this superstructure model (master model) gives a base-case flowsheet, where key variables and major structural options are optimized from the overall plant context.

Once the base flowsheet is obtained, thermodynamic analysis is applied in order to improve the base-case flowsheet. Since operational cost is directly linked to the thermodynamic performance of a plant, it is desirable to locate the inefficiencies and find ways to improve them by introducing structural changes.

The change options proposed from the thermodynamic analysis are added to the initial superstructure and the master model is modified accordingly. The modified master model

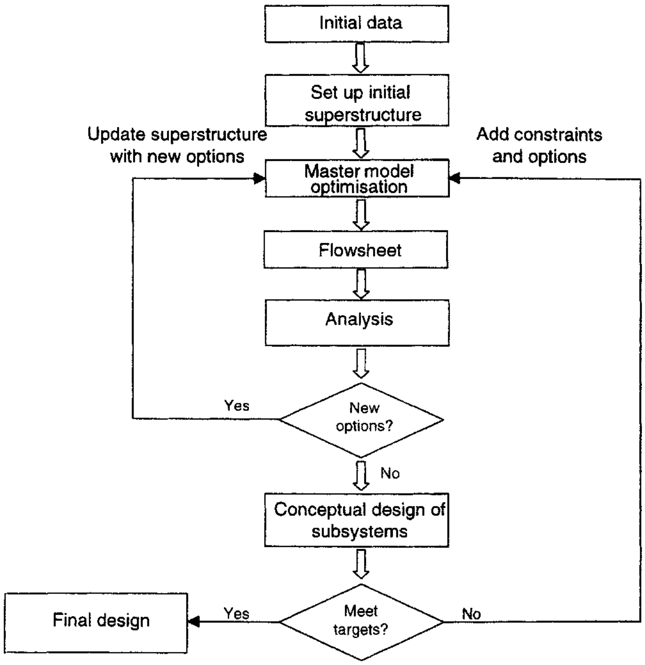

Figure 1. Overall design methodology.

is optimized again to maximize the overall profit by trading off the benefit and capital investment for these newly introduced change options. Some of the proposed design changes may not be retained in the optimized design due to their economic inferiority.

The preceding optimization and analysis procedure is repeated until no desirable improvement can be made. As a result, the master model optimization determines the optimal flowsheet in respect of major structural features and parameters (such as heat loads of various process units), and these are used as design requirements or targets for the design of subsystems. At this stage specialist software and techniques can be employed to carry out the design task. This allows a more rigorous evaluation of the performance and the cost for the individual subsystems.

The difference in level of detail between master model optimization and conceptual design is illustrated in Figure 2 using a heat recovery steam generator (HRSG) as an example.

In the master model optimization a black-box HRSG model is used to determine how much and at which pressure levels steam is to be produced without addressing the specifications and internal connections of heat exchangers. In the conceptual design stage, however, a heat exchanger network (HEN) superstructure is constructed using the results (steam mass flows and temperatures) from the master model optimization as constraints. Then the network structure and heat loads for individual exchangers are optimized. In this example, the master model optimization found generation of low-pressure (LP) steam uneconomical, and thus eliminated it from the HRSG. It was therefore excluded from the HRSG superstructure for the HEN design.

It may be possible that all the requirements derived from the master model cannot be satisfied due to the increased complexity or the prohibitive cost of the design of subsys-

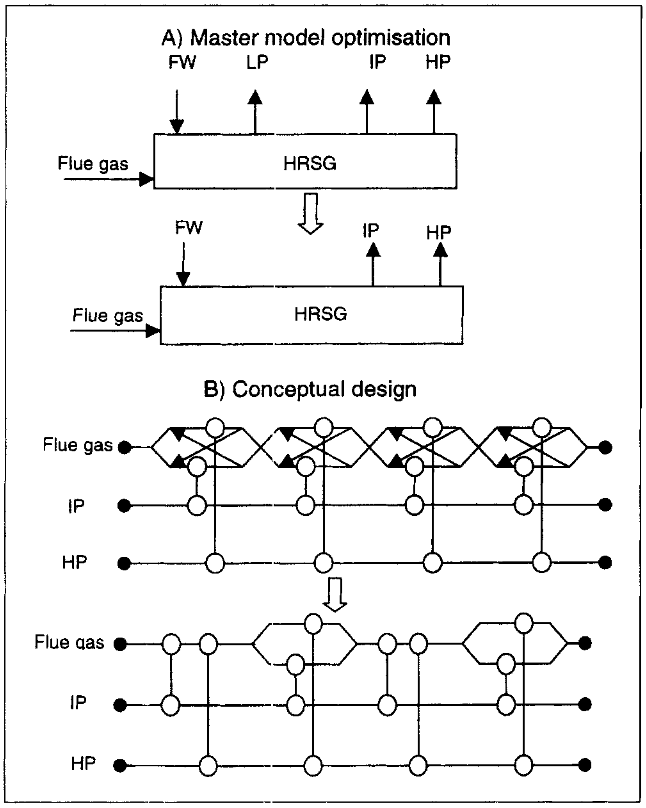

Figure 2. Design level complexity.

tems, control issues, and so on. Should this happen, a feedback mechanism is provided to resolve this problem. It is likely that some constraints or design options were not considered in previous master model optimization, either because the designer was unaware of them or they were deliberately left out to simplify the problem. The conceptual design stage will identify the significance of these constraints and options, and they should be added to the master model. The master model is then optimized again and a new conceptual design is carried out using the new requirements.

Mathematical formulation of the master model

General. The master model is an MINLP model. The non-linear terms come from area calculations for heat exchangers, unit operation performance models and energy balances. The binary variables are used to consider the existence or nonexistence of equipment and connections between equipment. Furthermore binary variables are also to select the optimal set of operating parameters. To formulate the model, the equations for the master model are given as follows. Nomenclature for the master model can be found in Appendix A.

Formulation. Mass and Energy Balances.

$$\sum_{j \in \mathrm{IN}(i,j)} m_{j,k} - \sum_{j \in \mathrm{OUT}(i,j)} m_{j,k} = 0 \quad \forall k \in \mathrm{K}, i \in (\mathrm{I}-\mathrm{GT}) \tag{1}$$

$$\sum_{j \in \mathrm{IN}(i,j)} m_{j,k} \cdot h_{j,k} - \sum_{j \in \mathrm{OUT}(i,j)} m_{j,k} \cdot h_{j,k} + \mathrm{Q}_{i,k} - W_{i,k} = 0 \quad \forall k \in \mathrm{K}, i \in (\mathrm{I}-\mathrm{GT}) \tag{2}$$

$$\mathrm{Q}_{i,k} = 0 \quad \forall k \in \mathrm{K}, i \in \mathrm{COMB, COMP, EXP, FWD, FWI, MIX, PUMP, SPLI, TUR} \tag{3}$$

$$\mathrm{W}_{i,k} = 0 \quad \forall k \in \mathrm{K}, i \in \mathrm{BX, BOIL, COMB, COOL, FWD, FWI, HR, MIX, SPLI} \tag{4}$$

$$\mathrm{Q}_{i,k} = \mathrm{Q}_i^{\mathrm{fix}} \cdot y_k^p \quad \forall k \in \mathrm{K}, i \in \mathrm{FIXQ} \tag{5}$$

$$\mathrm{W}_{i,k} = \mathrm{W}_i^{\mathrm{fix}} \cdot y_k^p \quad \forall k \in \mathrm{K}, i \in \mathrm{FIXW} \tag{6}$$

$\mathrm{Q}^{\mathrm{fix}}$ and $\mathrm{W}^{\mathrm{fix}}$ are multiplied by binary variable y^p to ensure that if parameter set k is selected, $Q = Q^{\mathrm{fix}}$ and $W = W^{\mathrm{fix}}$. Otherwise, Q and W are set to zero so that the energy balance (Eq. 2) still holds for those parameter sets, which are not selected.

Thermal Properties.

$$T_{j,k} = \tilde{T}_{j,k},\ h_{j,k} = \tilde{h}_{j,k} \quad \forall k \in \mathrm{K}, j \in (\mathrm{FP}-\mathrm{FUEL}) \tag{7}$$

$$h_{j,k} = \mathrm{LHV}_j \quad \forall k \in \mathrm{K}, j \in \mathrm{FUEL} \tag{8}$$

$$h_{j,k} = a_j^h \cdot \mathrm{T}_{j,k} + b_j^h \quad \forall k \in \mathrm{K}, j \in \mathrm{NP} \tag{9}$$

The specific enthalpies and temperatures for water and steam are defined as discrete parameters $(\tilde{h},\tilde{T})$ instead of continuous variables. The purpose is to eliminate complex nonlinear terms. Each parameter set can represent process conditions at different live steam pressures, reheat pressures, superheat temperatures, and so on. The parameter set, which gives the best overall performance, is selected from the optimization.

Temperature Assignments. These equations are used to assign values to inlet and outlet temperatures of some process units. These temperatures are then correlated with other variables in later defined equations.

$$\mathrm{T}_{i,k}^{\mathrm{in}} = T_{j,k} \quad \forall i \in \mathrm{COMP, EXP, HR, SPLI}, j \in \mathrm{IN}(i,j), k \in \mathrm{K} \tag{10}$$

$$\mathrm{T}_{i,k}^{\mathrm{out}} = T_{j,k} \quad \forall i \in \mathrm{COMP, EXP, GT, HR, SPLI}, j \in \mathrm{OUT}(i,j), k \in \mathrm{K} \tag{11}$$

$$\mathrm{T}_{i,k}^{fwin} = T_{j,k} \quad \forall i \in \mathrm{FWI}, j \in FW\mathrm{IN}(i,j), k \in \mathrm{K} \tag{12}$$

$$\mathrm{T}_{i,k}^{fwout} = T_{j,k} \quad \forall i \in \mathrm{FWI}, j \in FW\mathrm{OUT}(i,j), k \in \mathrm{K} \tag{13}$$

$$\mathrm{T}_{i,k}^{\mathrm{bleed}} = T_{j,k} \quad \forall i \in \mathrm{FWI}, j \in \mathrm{BLEED}(i,j), k \in \mathrm{K} \tag{14}$$

$$\mathrm{T}_{i,k}^{\mathrm{cout}} = T_{j,k} \quad \forall i \in \mathrm{FWI}, j \in \mathrm{COUT}(i,j), k \in \mathrm{K} \tag{15}$$

Equipment Performance and Cost Equations: Steam Turbines.

$$c_{t,k}^{\mathrm{tur}} = \alpha_{t,k}^{\mathrm{tur}} \cdot \sum_{i \in TC(i,t)} \mathrm{W}_{i,k} + \beta_{t,k}^{\mathrm{tur}} \cdot y_k^P \quad \forall k \in \mathrm{K}, t \in \mathrm{T} \tag{16}$$

Following Chou and Shih (1987), steam turbines are decomposed into several expansion sections with given isentropic efficiencies. The capital cost of steam turbines is calculated for each complete turbine. The coefficients (α^{tur}, β^{tur}) in the capital cost function can be assigned with a value for each parameter set to reflect the effect of varying pressure ratio and steam temperature on capital cost. If a parameter set k is not selected, mass flow through a turbine, or any other process unit, is set to zero (Eq. 45), and energy balance (Eq. 2) makes sure that work output of a turbine is also zero.

Gas Turbines.

$$W_{i,k} = W_i^{gt} \cdot y_{i,k}^{gt} \qquad \forall k \in \mathrm{K},\ i \in GT \tag{17}$$

$$Q_{i,k}^{\text{fuel}} = Q_i^{gt} \cdot y_{i,k}^{gt} \qquad \forall k \in \mathrm{K},\ i \in GT \tag{18}$$

$$T_{i,k}^{\text{out}} = T_i^{gt} \cdot y_{i,k}^{gt} \qquad \forall k \in \mathrm{K},\ i \in GT \tag{19}$$

$$\sum_{j \in \mathrm{OUT}(i,j)} m_{k,j} = m_i^{gt} \cdot y_{i,k}^{gt} \qquad \forall k \in \mathrm{K},\ i \in GT \tag{20}$$

$$c_{i,k}^{\text{cap}} = c_i^{gt} \cdot y_{i,k}^{gt} \qquad \forall k \in \mathrm{K},\ i \in GT. \tag{21}$$

Gas turbines are given as discrete options with their main data, work output, fuel consumption and exhaust temperature, and flow, provided as parameters. Binary variable y^{gt} is used for selecting a gas turbine from the set of available gas turbines.

Compressors and Expanders.

$$T_{i,k}^{\text{out}} = T_{i,k}^{\text{in}} \cdot r_{i,k}^{((\gamma_i - 1)/(\gamma_i \eta_i^{\pi}))} \qquad \forall k \in \mathrm{K},\ i \in \mathrm{COMP} \tag{22}$$

$$T_{i,k}^{\text{out}} = \frac{T_{i,k}^{\text{in}}}{r_{i,k}^{(\eta_i^{\pi}(\gamma_i - 1)/\gamma_i)}} \qquad \forall k \in \mathrm{K},\ i \in \mathrm{EXP} \tag{23}$$

$$c_{i,k}^{\text{cap}} = \alpha_i \cdot W_{i,k} + \beta_i \cdot y_k^p \qquad \forall k \in \mathrm{K},\ i \in \mathrm{COMP, EXP} \tag{24}$$

Pumps.

$$c_{i,k}^{\text{cap}} = \alpha_i \cdot \sum_{j \in IN(i,j)} m_{j,k} + \beta_i \cdot y_k^p \qquad \forall k \in \mathrm{K},\ i \in \mathrm{PUMP}. \tag{25}$$

Feedwater Heaters (Indirect Contact)

$$Q_{i,k}^{fw} = \sum_{j \in FW\mathrm{OUT}(i,j)} m_{j,k} \cdot h_{j,k} - \sum_{j \in FW\mathrm{IN}(i,j)} m_{j,k} \cdot h_{j,k} \qquad \forall k \in \mathrm{K},\ i \in \mathrm{FWI} \tag{26}$$

$$\Delta \mathrm{T1}_{i,k} = \mathrm{T}_{i,k}^{\text{cout}} - \mathrm{T}_{i,k}^{fw\text{in}} \qquad \forall k \in \mathrm{K},\ i \in \mathrm{FWI} \tag{27}$$

$$\Delta \mathrm{T2}_{i,k} = \mathrm{T}_{i,k}^{\text{bleed}} - \mathrm{T}_{i,k}^{fw\text{out}} \qquad \forall k \in \mathrm{K},\ i \in \mathrm{FWI} \tag{28}$$

$$\Delta \mathrm{T}_{i,k}^{LM} = \left[\Delta \mathrm{T1}_{i,k} \cdot \Delta \mathrm{T2}_{i,k} \cdot \left(\frac{\Delta \mathrm{T1}_{i,k} + \Delta \mathrm{T2}_{i,k}}{2} \right) \right]^{1/3} \qquad \forall k \in \mathrm{K},\ i \in \mathrm{FWI} \tag{29}$$

$$\mathrm{A}_{i,k} = \frac{\mathrm{Q}_{i,k}^{fw}}{\mathrm{U}_i \cdot \Delta \mathrm{T}_{i,k}^{LM}} \qquad \forall k \in \mathrm{K},\ i \in \mathrm{FWI} \tag{30}$$

$$c_{i,k}^{\text{cap}} = \alpha_i \cdot \mathrm{A}_{i,k} + \beta_i \cdot y_k^p \qquad \forall k \in \mathrm{K},\ i \in \mathrm{FWI}. \tag{31}$$

The user fixes the temperature differences ($\Delta T1, \Delta T2$) for individual feedwater heaters. The logarithmic mean temperature is estimated using Chen's correlation (1987).

Feedwater Heaters (direct contact).

$$c_{i,k}^{\text{cap}} = \alpha_i \cdot \sum_{j \in \mathrm{IN}(i,j)} m_{j,k} + \beta_i \cdot y_k^p \qquad \forall k \in \mathrm{K},\ i \in \mathrm{FWD}. \tag{32}$$

Heat Recovery Units. The heat recovery units, which are boilers and heat recovery steam generators (HRSG), are modeled using the principle shown in Figure 3. For the flue gas (hot) curve, the mass flow, temperatures, and enthalpies are variables. For steam and water streams, properties are selected from several sets of parameters and mass flows are variables. A cold composite curve represents the different heating duties in a heat recovery unit (Figure 3). The curve represents a single steam level HRSG with reheat. Each of the sections in the heat recovery unit is modeled as a process unit. In this example the first process unit has two heating duties assigned to it, namely superheat and reheat. The model considers any heat recovery unit as a cooler, since it cools down the process stream (flue gas), and therefore it is assigned with a negative heat load.

$$Q_{i,k} + \sum_{i \in \mathrm{REC}(i \in \mathrm{BX},i)} \mathrm{Q}_{i,k} = 0 \qquad \forall k \in \mathrm{K},\ i \in \mathrm{HR} \tag{33}$$

$$T_{i,k}^{\text{out}} \geq \mathrm{T}_{i,k}^{\text{maxin}} + \Delta T_i^{\text{min}} - (1 - y_k^p) \cdot \Gamma \qquad \forall k \in \mathrm{K},\ i \in \mathrm{HR} \tag{34}$$

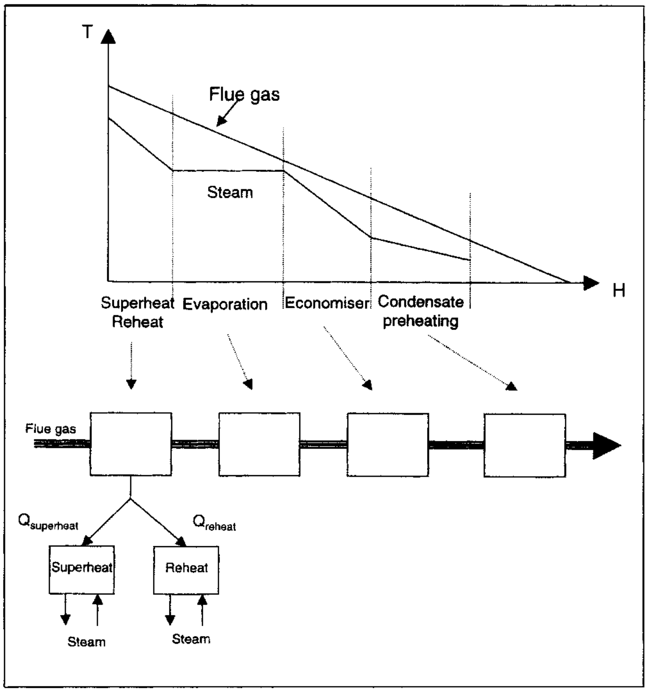

Figure 3. Division of HRSG sections.

$$T^{\mathrm{in}}_{i,k} \geq T^{\mathrm{maxout}}_{i,k} + \Delta T^{\mathrm{min}}_{i} - (1 - y^{p}_{k}) \cdot \Gamma \qquad \forall k \in \mathrm{K},\, i \in \mathrm{HR} \tag{35}$$

$$\Delta T1_{i,k} = T^{\mathrm{out}}_{i,k} - T^{\mathrm{maxin}}_{i,k} \qquad \forall k \in \mathrm{K},\, i \in \mathrm{HR} \tag{36}$$

$$\Delta T2_{i,k} = T^{\mathrm{in}}_{i,k} - T^{\mathrm{maxout}}_{i,k} \qquad \forall k \in \mathrm{K},\, i \in \mathrm{HR} \tag{37}$$

$$\Delta T^{\mathrm{LM}}_{i,k} = \left[\Delta T1_{i,k} \cdot \Delta T2_{i,k} \cdot \left(\frac{\Delta T1_{i,k} + \Delta T2_{i,k}}{2} \right) \right]^{1/3} \qquad \forall k \in \mathrm{K},\, i \in \mathrm{HR} \tag{38}$$

$$c^{\mathrm{cap}}_{i,k} = \alpha_i \cdot \left(\frac{-Q_{i,k}}{\Delta T^{\mathrm{LM}}_{i,k}} \right) + \phi_i \cdot \sum_{i \in \mathrm{REC}(i \in BX, i)} \sum_{j \in \mathrm{IN}(l,j)} m_{j,k} + \psi_i \cdot \sum_{j \in \mathrm{IN}(i,j)} m_{j,k} \qquad \forall k \in \mathrm{K},\, i \in \mathrm{HR}. \tag{39}$$

The capital cost function is simplified from Foster-Pegg (1986), where cost is a function of effective heat-transfer area and mass flows of steam and flue gas.

Combustor. A combustor is modeled as a mixer, where the mixing ratio between air and fuel is controlled by the following equation. The fuel/air ratio (f^f) determines the minimum air requirement for any given fuel.

$$\sum_{j \in \mathrm{FUI}(i,j)} m_{j,k} \leq \sum_{j \in \mathrm{AIN}(i,j)} m_{j,k} \cdot f^f_j \qquad \forall k \in \mathrm{K},\, i \in \mathrm{COMB} \tag{40}$$

Simple Boiler. In addition to a heat recovery unit, steam can be generated in a boiler. In this simple boiler model, the fuel consumption (Q^{fuel}) and capital cost are based on the heat load of the generated steam:

$$Q^{\mathrm{fuel}}_{i,k} = \frac{Q_{i,k}}{\eta_i} \qquad \forall k \in \mathrm{K},\, i \in \mathrm{BOIL} \tag{41}$$

$$c^{\mathrm{cap}}_{i,k} = \alpha_i \cdot Q_{i,k} + \beta_i \cdot y^p_k \qquad \forall k \in \mathrm{K},\, i \in \mathrm{BOIL}. \tag{42}$$

Splitters.

$$T^{\mathrm{in}}_{i,k} = T^{\mathrm{out}}_{i,k} \qquad \forall k \in \mathrm{K},\, i \in \mathrm{SPLI}. \tag{43}$$

Logical Constraints: Selection of Steam Parameters.

$$\sum_{k \in K} y^p_k = 1 \tag{44}$$

$$m_{j,k} - y^p_k \cdot \Gamma \leq 0 \qquad \forall j \in J,\, k \in \mathrm{K}. \tag{45}$$

A binary variable y^p is assigned for each parameter set k. Only one of the sets will be selected. For those parameter sets that are not selected, the mass flow of all streams is set to zero. Since all variables in objective function are functions of mass flow, this formulation ensures that the objective function reflects the impact of nonzero mass flows associated with the selected parameter set.

Minimum Flow. A minimum flow, which is modeled using a binary variable y^f, can be set for streams. For parameter sets not selected, the constraint (Eq. 46) is inactivated:

$$m_{j,k} \geq m^{\mathrm{min}}_j \cdot y^f_j - (1 - y^p_k) \cdot \Gamma \qquad \forall k \in \mathrm{K},\, j \in MF \tag{46}$$

$$m_{j,k} - y^f_j \cdot \Gamma \leq 0 \qquad \forall k \in \mathrm{K},\, j \in MF. \tag{47}$$

Minimum Temperature. Minimum temperature constraint can be used to enforce, for example, a minimum allowable stack temperature:

$$T_{j,k} \geq T^{\mathrm{min}}_j - (1 - y^p_k) \cdot \Gamma \quad \forall k \in \mathrm{K},\, j \in TM. \tag{48}$$

Optional Process Units. Optional process units are selected using binary variable y^o. If an optional process unit is not selected, the mass flow through the unit is forced to be zero:

$$\sum_{j \in \mathrm{IN}(i,j)} m_{j,k} - y^o_{i,k} \cdot \Gamma \leq 0 \qquad \forall k \in \mathrm{K},\, i \in OM \tag{49}$$

$$y^p_k - y^o_{i,k} \geq 0 \qquad \forall k \in \mathrm{K},\, i \in OM. \tag{50}$$

Gas Turbine Selection.

$$\sum_{k \in K} \sum_{i \in GT} y^{gt}_{i,k} = 1 \tag{51}$$

$$y^p_k - y^{gt}_{i,k} \geq 0 \qquad \forall k \in \mathrm{K},\, i \in GT. \tag{52}$$

Selection of a gas turbine for any parameter set is controlled by the binary variable y^{gt}. Only one gas turbine can be selected, and for the parameter sets not selected y^{gt} must be set to zero.

Objective Function.

$$\mathrm{OBJ} = \sum_{k \in K} \sum_{i \in I} W_{i,k} \cdot c^e \cdot t - \sum_{k \in K} \left(\sum_{j \in \mathrm{FUEL}} m_{j,k} \cdot h_{j,k} \cdot c^f_j + \sum_{i \in \mathrm{BOIL}} Q^{\mathrm{fuel}}_{i,k} \cdot c^f_i \right) \cdot t - f^a \cdot \sum_{k \in K} \left(\sum_{i \in I} c^{\mathrm{cap}}_{i,k} + \sum_{t \in T} c^{\mathrm{tur}}_{t,k} \right). \tag{53}$$

The objective of the model is to maximize the annual profit of the total plant. The first part of the objective is the annual revenue generated by selling electricity. Please note that in the formulation all the power-consuming equipment, like pumps, have negative values for work. The second part is the annual operating cost of the plant, and the third part represents the annualized capital expenditure of process units. This objective function can deal with two other design scenarios. For a fixed power output, the preceding objective function becomes the minimization of capital cost. Alternatively, for a given capital investment, the objective function tends to maximize power output with minimum operating cost.

Remarks for the Model. Equation 1–53 constitute the master model, which can be used for the most common types of thermal systems. Additional equations depicting the behavior of other types of equipment can be added to extend the model.

The preceding master model is a MINLP formulation that is solved on a 200-MHz Pentium PC using a Dicopt + + solver (Viswanathan and Grossmann, 1990) in a GAMS modeling environment (Brooke et al., 1992). The number of binary variables for a given problem is the summation of parameter sets, the number of optional modules multiplied by the number of parameter sets, the number of flows with minimum flow rate, and the number of optional gas turbines multiplied by the number of parameter sets. For a typical problem the overall number of binary variables is between 40 and 100. The number of continuous variables can be several thousands, depending on the problem complexity. The CPU time for solving the master model on a Pentium PC is typically between one and five minutes per run.

Thermodynamic analysis

Thermodynamic analysis is used in this approach to identify thermal inefficiencies and to suggest design changes to improve the initial flow sheet obtained by the master model. It is based on second law analysis by means of a graphical tool (Zheng et al., 1996).

Combined Pinch and Exergy Representation. Combined pinch and exergy representation (CPER) (Zheng et al., 1996) is a graphical representation of the heat transfer system in a power plant, which is obtained by plotting the Carnot factor ($\eta_c = 1 - T_0/T$) against enthalpy (Figure 4). The shaded area represents the exergy losses (ΔEx_{HEN}) incurred from heat transfer processes.

Figure 4 shows the CPER of a conventional Rankine cycle. The overall exergy supplied to the power plant (ΔEx_{fuel}) comes from the fuel, some of which is lost throughout the system. In the furnace, exergy losses occur due to heat transfer between the flue gas and the steam generation and between the flue gas and the combustion air. In the feedwater heaters, exergy losses occur in the process of heat transfer between the feedwater and bleed streams. Another part of exergy loss occurs in the condenser. Since all the exergy losses occur in the HEN, they are represented by the term ΔEx_{HEN}. Subtracting all the exergy losses (ΔEx_{HEN}) from the total exergy supplied by the fuel (ΔEx_{fuel}) gives the exergy gain for power generation (ΔEx_{power}). The turbine system receives the ΔEx_{power} and transforms it into actual shaftwork. Clearly, the smaller the ΔEx_{HEN}, the larger the ΔEx_{power} for power generation.

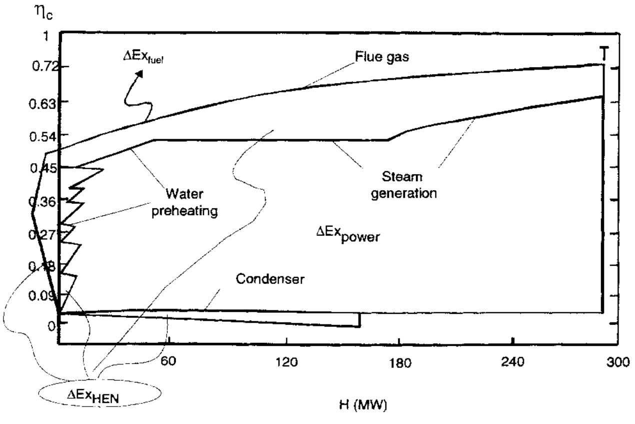

Figure 4. Combined pinch and exergy representation.

From the CPER, a designer is able to visualize the performance of each process and identify the inefficient parts that have big exergy losses. Introducing structural changes to the system can reduce some of these exergy losses. For example, losses can be reduced in the feedwater heating section by introducing additional heaters. Similarly in boilers or heat recovery units introduction of additional steam levels or reheaters can reduce losses.

The main purpose of using thermodynamic analysis in the overall procedure is to readily identify promising options, which are used to update the superstructure of the master model. Thus the designer can start with a relatively simple superstructure and update it according to analysis results until no desirable improvement can be made.

Conceptual design of subsystems

General. The models for doing the conceptual design of subsystems differ widely depending on the specific design task. This stage is separated from the master model optimization. In this stage, specialist software and techniques can be used to carry out the design task. For example, heat exchanger network design software can be used to design the heat recovery units. In the master model we use simplified cost equations, which are accurate enough for high-level decision making. In the design of subsystems, however, we are employing more rigorous nonlinear cost equations. Here are examples of how the conceptual design of subsystems could be done.

Heat Recovery Units. Heat recovery units, which include boilers and heat recovery steam generators, are modeled following a similar approach as that used in HEN design. For each of the recovery units, a superstructure can be built that contains all the heat exchangers assigned by the master model to a specific heat recovery unit (e.g., Figure 2 for HRSG). The objective is to obtain a network configuration with a minimum cost satisfying the given requirements determined from the master model optimization.

Methods for solving HENs can be found in literature (e.g., Yee and Grossmann, 1990; Ciric and Floudas, 1991; Zhu et al., 1995). It should be noted that UMIST software (SPRINT, 1995) with the method proposed by Zhu et al. (1995) is used in this work. The capital cost of heat exchangers is calculated using Foster-Pegg's (1986) correlation as follows.

$$c_i^{\text{cap}} = a_i \cdot \left(\frac{Q_i}{\Delta T_i^{LM}} \right)^{0.8} + 21.276 \cdot m_i^{\text{steam}} + 1.184 \cdot (m^{fg})^{1.2} \qquad \forall i \in \text{BX}, \quad (54)$$

where coefficient a_i is 13.0 for any other type of exchanger except evaporators that have a value of 6.5; m^{steam} is the steam flow through the exchanger (kg/s); and m^{fg} is the flue gas flow (kg/s).

Feedwater Preheat Train and Steam Turbine. The design of the feedwater heating train may not be achieved by using

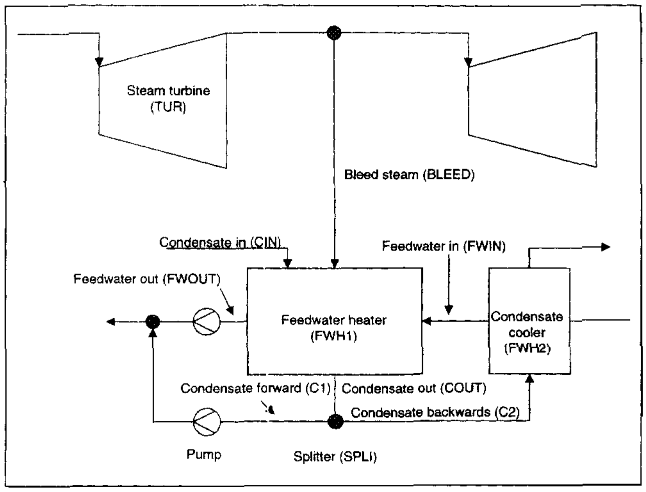

Figure 5. Feedwater heater model.

standard HEN software, since the heaters can be either the indirect or direct contact type. Also steam turbines providing the bleed steam for heating should be designed at the same time that the configuration of feedwater heaters is determined. The optimization is to obtain the optimal temperature differences in the heaters, and decide the type of heaters. We have developed an optimization model that can be used to model direct contact and indirect contact heat exchangers in connection with steam turbines. Figure 5 contains the basis for the following mathematical formulation. The design model for this task is an MINLP model. The formulation of this design model is given as follows. Nomenclature can be found in Appendix A.

Formulation: Mass and Energy Balances:

$$Q_i = 0 \qquad \forall i \in \text{FWH, PUMP, SPLI, TUR} \tag{55}$$

$$W_i = 0 \qquad \forall i \in \text{COOL, FWH, SPLI.} \tag{56}$$

Thermal Properties.

$$T_j^{\text{sat}} = \tilde{T}_j^{\text{sat}} \qquad \forall j \in \text{FP} \tag{57}$$

$$h_j = -0.024\times 10^{-3}\cdot T_j^3 + 9.47\times 10^{-3}\cdot T_j^2 - 5.217\times T_j + 27.567 \qquad \forall j \in \text{NP}. \tag{58}$$

Saturation temperatures for bleed streams are provided as parameters. Since temperature differences in the heat exchangers are variables, the correlation function for calculating the enthalpy of water is required (Eq. 58). This correlation provides good accuracy over a wide range of temperatures (Manninen, 1999).

Temperature Assignments. Similarly to Eqs. 10–15, the following two equations assign values for bleed-steam temperature and saturation temperature:

$$T_i^{\text{bleed}} = T_j, \qquad TS_i^{\text{bleed}} = T_j^{\text{sat}} \qquad \forall i \in \text{FWH}, \quad j \in \text{BLEED}(i,j). \tag{59}$$

Feedwater Heater Mass Flow Constraints.

$$\sum_{j\in FWIN(i,j)} m_j = \sum_{j\in FWOUT(i,j)} m_j \qquad \forall i \in \text{FWH2} \tag{60}$$

$$\sum_{j\in \text{COUT}(i,j)} m_j \geq \sum_{j\in \text{BLEED}(i,j)} m_j + \sum_{j\in C\text{IN}(i,j)} m_j - \left(1-y_i^{fwh}\right)\cdot\Gamma \qquad \forall i \in \text{FWH1} \tag{61}$$

$$\sum_{j\in \text{COUT}_{(i,j)}} m_j \leq \sum_{j\in \text{BLEED}(i,j)} m_j + \sum_{j\in C\text{IN}(i,j)} m_j \qquad \forall i \in \text{FWH1} \tag{62}$$

$$\sum_{j\in COUT(i,j)} m_j \leq y_i^{fwh}\cdot\Gamma \qquad \forall i \in \text{FWH1} \tag{63}$$

$$\sum_{j\in C1(i,j)} m_j \leq y_i^{c}\cdot\Gamma \qquad \forall i \in \text{FWH1} \tag{64}$$

$$\sum_{j\in C2(i,j)} m_j \leq \left(1-y_i^{c}\right)\cdot\Gamma \qquad \forall i \in \text{FWH1}. \tag{65}$$

For condensate coolers ($i \in FWH2$), the feedwater amount does not change. On the other hand, for feedwater heaters ($i \in FWH1$), this depends on the type of heat exchanger. The type of heat exchanger (i.e., direct or indirect contact) for the feedwater heater is controlled by a binary variable y^{fwh}. When it takes a value of one, the feedwater heater is of the indirect contact type; otherwise, it is direct type.

Feedwater Heater Temperature Constraints.

$$\Delta T1_i = T_i^{\text{cout}} - T_i^{fwin} \qquad \forall i \in \text{FWH} \tag{66}$$

$$\Delta T2_i = \frac{TS_i^{\text{bleed}} + T_i^{\text{bleed}}}{2} - T_i^{fwout} \qquad \forall i \in \text{FWH} \tag{67}$$

$$\Delta T_i^{LM} = \left[\Delta T1_i\cdot\Delta T2_i\cdot\left(\frac{\Delta T1_i + \Delta T2_i}{2}\right)\right]^{1/3} \qquad \forall i \in \text{FWH2} \tag{68}$$

$$\Delta T_i^{LM} = \left[\Delta T1_i\cdot\Delta T2_i\cdot\left(\frac{\Delta T1_i + \Delta T2_i}{2}\right)\right]^{1/3} + \left(1-y_i^{fwh}\right)\cdot\Gamma \qquad \forall i \in \text{FWH1} \tag{69}$$

$$\Delta T1_i \geq \Delta T_i^{\min 1} \qquad \forall i \in \text{FWH} \tag{70}$$

$$\Delta T2_i \geq \Delta T_i^{\min 2} \qquad \forall i \in \text{FWH2} \tag{71}$$

$$\Delta T2_i \geq \Delta T_i^{\min 2} - \Gamma\cdot y_i^{fwh} \qquad \forall i \in \text{FWH1}. \tag{72}$$

For direct-contact heat exchangers, temperature difference and hence logarithmic mean temperature can be zero. Therefore a large number is added in Eq. 69 to avoid numerical problems in area calculations.

Capital Cost Equations.

$$c_i^1 = 1.06\cdot A_i^{0.68} \qquad \forall i \in \text{FWH} \tag{73}$$

$$c_i^2 = 13.33\cdot\left(\sum_{j\in IN(i,j)} m_j\right)^{0.62} \qquad \forall i \in \text{FWH1} \tag{74}$$

$$c_i^3 \le c_i^1 \qquad \forall i \in \text{FWH1} \tag{75}$$

$$c_i^3 \ge c_i^1 - \left(1 - y_i^{fwh}\right) \cdot \Gamma \qquad \forall i \in \text{FWH1} \tag{76}$$

$$c_i^3 \le y_i^{fwh} \cdot \Gamma \qquad \forall i \in \text{FWH1} \tag{77}$$

$$c_i^4 \le c_i^2 \qquad \forall i \in \text{FWH1} \tag{78}$$

$$c_i^4 \ge c_i^2 - y_i^{fwh} \cdot \Gamma \qquad \forall i \in \text{FWH1} \tag{79}$$

$$c_i^4 \le \left(1 - y_i^{fwh}\right) \cdot \Gamma \qquad \forall i \in \text{FWH1} \tag{80}$$

$$c_i^{\text{cap}} = c_i^3 + c_i^4 \qquad \forall i \in \text{FWH1} \tag{81}$$

$$c_i^{\text{cap}} = c_i^1 \qquad \forall i \in \text{FWH2}. \tag{82}$$

Auxiliary variables $c^1 - c^4$ are introduced to determine whether the capital cost for the indirect- or direct-contact type is to be considered for a specific heat exchangers. Values for c^1 and c^2 are obtained from Garrett (1989) and Bruno et al. (1998), respectively.

Pumps.

$$W_i = -0.11 \cdot \sum_{j \in \text{IN}(i,j)} F_j \cdot dp_i \qquad \forall i \in PUMP \tag{83}$$

$$c_i^{\text{cap}} = 4.72 \cdot \left(\sum_{j \in \text{IN}(i,j)} F_j \right)^{0.42} \qquad \forall i \in PUMP \tag{84}$$

$$\sum_{j \in C1(i,j)} F_j = \sum_{j \in C1(i,j)} m_j \qquad \forall i \in \text{FWH1} \tag{85}$$

$$\sum_{j \in FW\text{OUT}(i,j)} F_j \le \sum_{j \in FW\text{OUT}(i,j)} m_j \qquad \forall i \in \text{FWH1} \tag{86}$$

$$\sum_{j \in FW\text{OUT}(i,j)} F_j \ge \sum_{j \in FW\text{OUT}(i,j)} m_j - y_i^{fwh} \cdot \Gamma \qquad \forall i \in \text{FWH1} \tag{87}$$

$$\sum_{j \in FW\text{OUT}(i,j)} F_j \le \left(1 - y_i^{fwh}\right) \cdot \Gamma \qquad \forall i \in \text{FWH1}. \tag{88}$$

The shaftwork requirement for a pump is a function of the specific volume of water (estimated here at 1.1 dm^3/kg), flow through the pump (F), and the pressure rise in the pump (dp). Since the unit for dp is bar, the specific volume is multiplied in order to have the shaftwork in kW. The capital cost correlation is from Garrett (1989). For the condensate pumps, variable F equals the mass flow. For feedwater pumps, however, the flow must be zero if the heat exchanger is of indirect type (y^{fwh} equals one).

Steam Turbine.

$$c^{\text{tur}} = \alpha^{\text{tur}} \cdot \left(\sum_{i \in \text{TUR}} W_i \right)^{0.7}. \tag{89}$$

Objective Function.

$$\text{OBJ} = \sum_{i \in I} W_i \cdot c^e \cdot t - f^a \cdot \left(\sum_{i \in I} c_i^{\text{cap}} + c^{\text{tur}} \right). \tag{90}$$

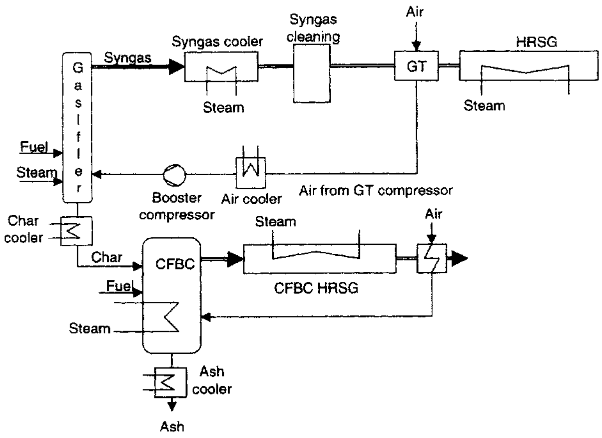

Figure 6. Simplified flowsheet of AGGT plant.

The objective of the model is to maximize the annual profit of the system, which consists of a steam turbine and feedwater heaters. The first part of the objective is the annual revenue generated by selling electricity. The second part is the annualized capital expenditure. Equations 1, 2, 7, 26, 30, 43 and 55–90 form the complete model for the design of the feedwater preheat train.

Case Study

A case study involving the design of the steam system for an air gasification gas turbine (AGGT) plant is used to show the procedure and demonstrate the methodology. A flowsheet of the AGGT system is shown in Figure 6. The aim of this design problem is to design a steam cycle that optimally fits this given AGGT system. The design task is a large and challenging one due to the existence of several steam generators, potential existence of several steam generation levels, and so on.

In the system in Figure 6, fuel is gasified in the gasifier in the presence of air and steam. The syngas is cooled down in a syngas cooler, cleaned, and used as fuel in the gas turbine (GT). The GT exhaust is utilized in a separate HRSG. The gas turbine compressor supplies air for the gasifier through an air cooler. Char residue from the gasifier is cooled in a char cooler and fed with additional fuel to a circulating fluidized-bed boiler (CFBC). Fuel gas from the CFBC goes through a heat-recovery generator (CFBC HRSG) and is cooled to the stack temperature in an air preheater. Ash from the CFBC goes through an ash cooler and is disposed of. Steam can be generated by several pieces of equipment as shown in Figure 6. Appendix B summarizes the main assumptions and constraints for this case study.

The duties of the gasifier, gas turbine, and CFBC unit, together with the associated gas flow rates, were specified as fixed ground rules for this study. They were sized appropriately for a gas turbine of output 200 MW, corresponding to a total gross generation (gas turbine + steam turbine) of about 450 MW. The capacity of the steam turbine was not specified in advance, but was calculated as part of the optimization of the steam side of the cycle.

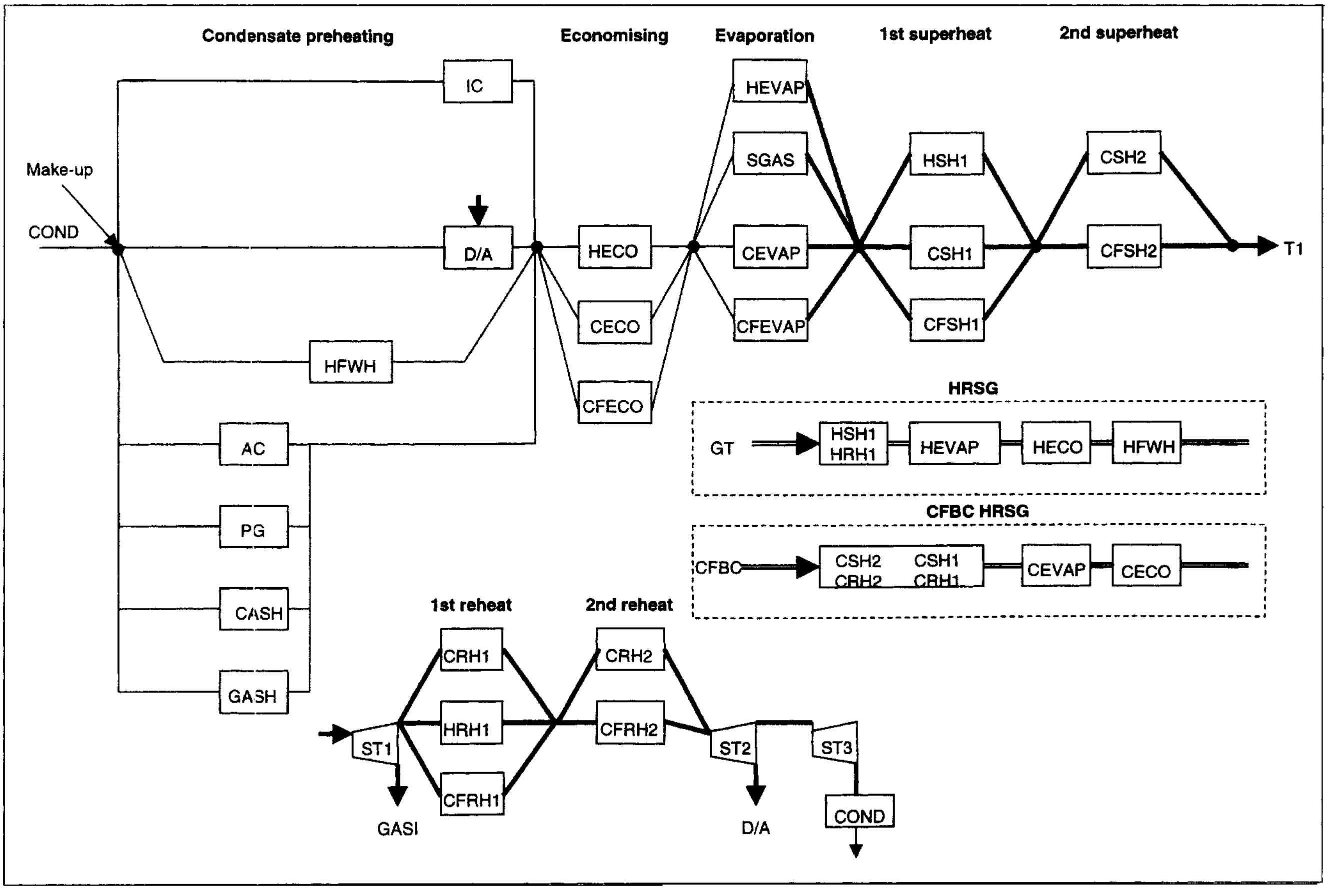

Figure 7. Initial superstructure.

Since the gasification system is fixed, we are effectively maximizing the incremental profit generated by the steam cycle. Note that the capital cost annualization factor for the calculations is 0.125, and the price of sold electricity is \$20/MWh.

The optimization was carried out on a 200-MHz Pentium PC using GAMS/Dicopt++(Viswanathan and Grossmann, 1990).

Initial flowsheet synthesis

First we investigate a steam cycle with one pressure level and steam reheat. The possible connections between the process units are postulated to construct the initial superstructure (Figure 7). The main constraints, assumptions, and notations for the initial superstructure are given in Appendix B. The parameter sets are given in Table 1.

These specific pressures are selected on the grounds that they reflect typical steam conditions for a subcritical reheat steam plant. For each parameter set, temperatures and enthalpies for streams consisting of water and steam are determined by using steam tables.

The master model is formulated based on the initial superstructure. The initial master model has around 2400 continuous and 6 binary variables. Optimization of the master model gives the results described in Table 2.

Among the 6 candidate parameter sets, set 5 is selected from the optimization. The initial flow sheet, which is a subset of the initial superstructure, is shown in Figure 8. It should be noted that the heat-recovery units and condensate preheaters are modeled in a low level of detail, and the conceptual design for these process units is considered later.

Improvement of the initial flowsheet

The CPER of the base-case design is shown in Figure 9. The feedwater curve runs parallel to the flue-gas curve, as highlighted in the figure. This indicates that a second steam-

Table 1. Parameter Sets for the Master Model

Pressure	Set 1	Set 2	Set 3	Set 4	Set 5	Set 6
HP (bar)	160	140	180	160	140	180
Reheat (bar)	42	35	45	42	35	45
Deaerator (bar)	4	4	4	1.7	1.7	1.7

Table 2. Results from Initial Flowsheet Synthesis

Steam turbine output (MW)	251.7
Overall capital cost (\$1,000)	54,879
Overall profit (\$1,000/yr)	33,412
HP/reheat/deaerator pres. (bar)	140/35/1.7 (set 5)
CPU time (s)	7.9

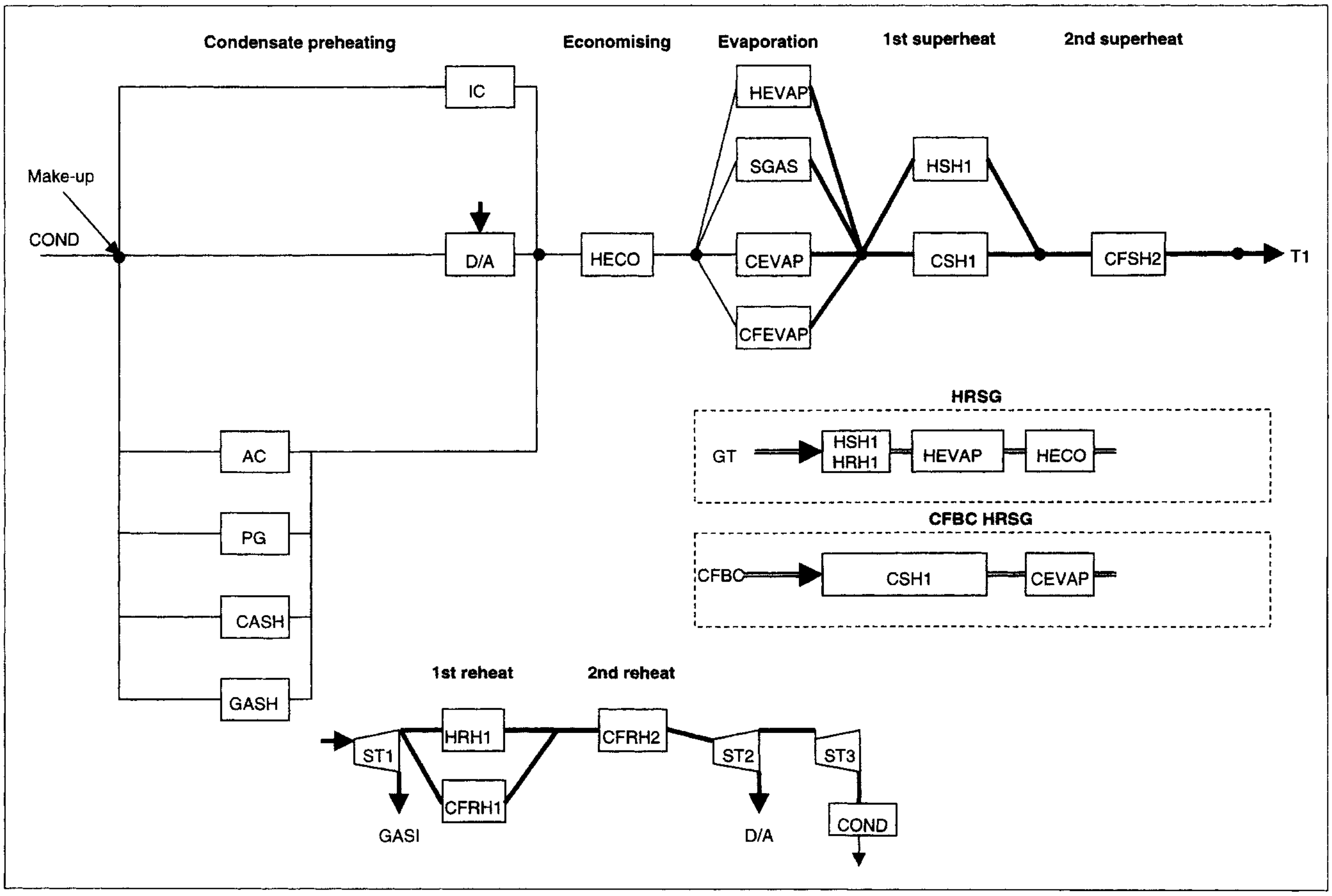

Figure 8. Initial flowsheet.

generation level is not likely to bring any additional benefits, since it would increase the area between the hot and cold composites and therefore increase losses. The same applies to using steam to preheat feedwater above the deaerator temperature. Additional condensate heaters may have the benefit of bringing the cold curve closer to the hot curve in the lower temperature region.

Based on this observation, three condensate heaters using bleed steam were placed so that the water heating duty was equally distributed among them. This is widely recognized as giving the best thermal performance (El-Wakil, 1984). They are assumed to be of the indirect type with no condensate cooling. The terminal temperature difference (TTD) between the saturation temperature of the bleed steam and the outlet temperature of the condensate was fixed at 3°C. The condensates from the heaters do not contribute to the condensate preheating at this stage.

Three of the coolers have high enough temperatures to be used for reasons of economy, which might also help to reduce losses. This option is added to the master model together with three condensate heaters. Figure 10 shows the updated superstructure with all the added options. It should be noted that additional turbine sections need to be added due to the introduction of three condensate preheaters. The master model size has thus increased to around 2800 continuous and 42 binary variables.

Optimization of the modified master model gives the results (Table 3). It can be seen that significant improvement has been achieved from this optimization compared with the results in Table 2. The CPER of the improved flow sheet is shown in Figure 11. The reassignment of the cooler duties and introduction of heaters had the anticipated effect of bringing the two curves much closer, as highlighted in the figure, and thus reducing exergy losses in the preheating and economizing sections.

However, the optimization does not allocate any evaporation duty to the CFBC. Generally speaking, fluidized-bed boiler evaporator duty should be at least 25% of the overall duty (Basu and Fraser, 1991). This design aspect is identified after the present master model optimization. Thus the feed-

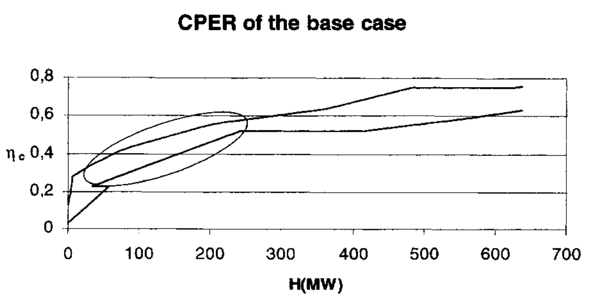

Figure 9. CPER of the base case.

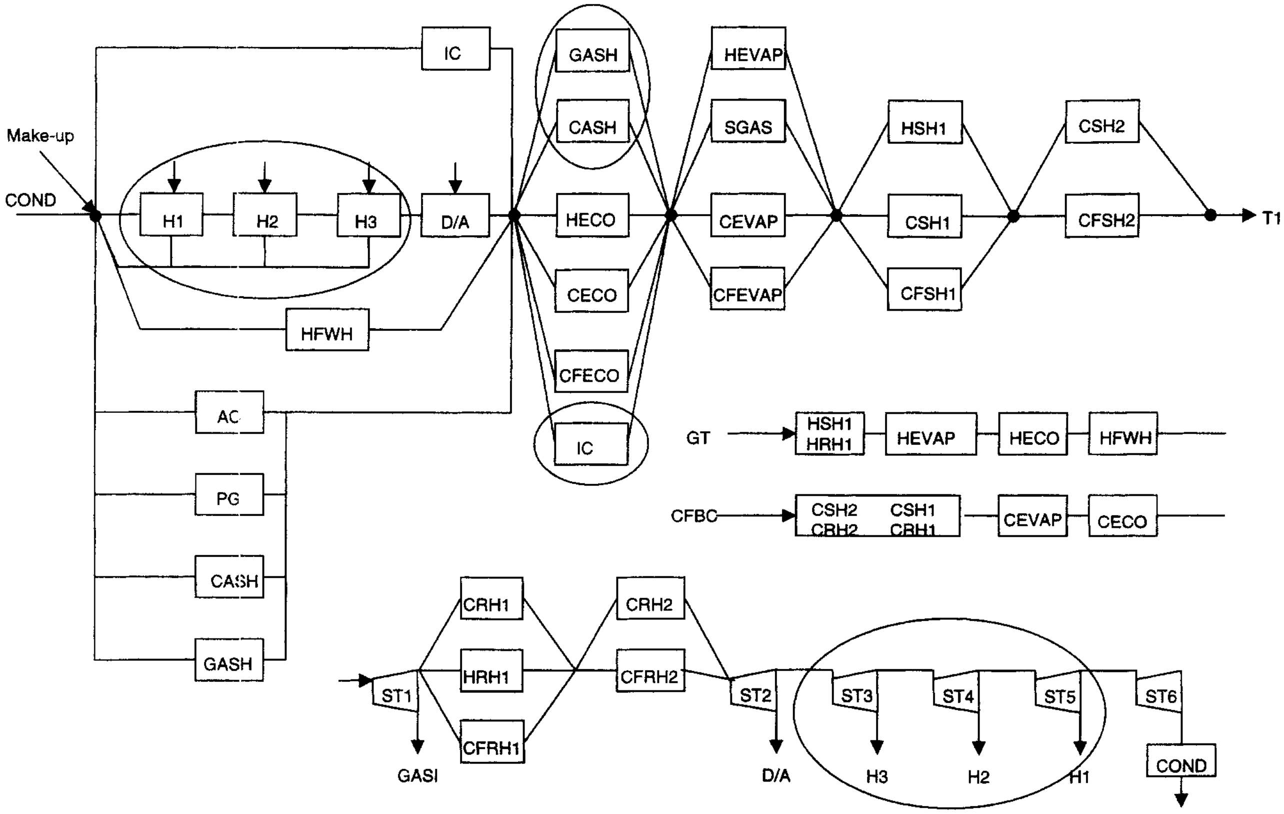

Figure 10. Updated superstructure.

back mechanism is employed and an additional constraint is introduced to the master model to ensure that the minimum evaporation requirement is met.

Optimization of the constrained master model allocates the minimum required evaporation duty to the fluidized-bed boiler. At the same time it also allocates two high-temperature duties, namely secondary superheat and reheat, to the CFBC. At this stage, it is realized for this case study these two heaters cannot feasibly be placed together in the CFBC. To account for this, additional logical constraints are introduced to make these two heaters mutually exclusive. This increases the number of binary variables to 54.

Optimization of the further constrained master model resolves the problem by reallocating secondary superheat to the CFBC HRSG. The evaporation flow in the CFBC HRSG is less than 7 kg/s, however, and to avoid small flow rates in the heat exchangers, a minimum flow constraint of 10 kg/s is imposed on all boiler heat exchangers. This option increases the number of binary variables to 156.

Table 3. Results of Improved Flow Sheet

Steam turbine output (MW)	262.1 (+10.4)
Capital cost ($1,000)	55,247 (+368)
Profit ($1,000/yr)	35,034 (+1,622)
HP/reheat/deaerator pres. (bar)	140/35/1.7 (set 5)
CPU time (s)	6.3

Finally, optimization of the newly constrained master model balances the flows in such a manner that the flow constraints are met. The results of the previous optimization runs with three iteration steps using the feedback mechanism are shown tin Table 4.

Main Steam Parameters. After deciding the major structural issues, the main steam parameters are optimized. In flow-sheet synthesis a choice from six widely spaced parameter sets was enough to decide structural issues and to estimate steam conditions. In parameter optimization the steam conditions are refined by choosing from closer spaced parameter sets. This optimization does not change the steam pressures, but changes approach temperature (temperature dif-

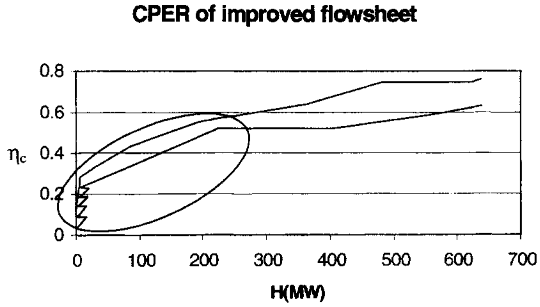

Figure 11. CPER of the improved flowsheet.

Table 4. Results of Constrained Optimization

	CFBC Min. Evapor. Duty	CFBC Superheat and Reheat	Min. Flow for Boiler Duties
Steam turbine output (MW)	262.1 (+0.0)	262.1 (0.0)	262.1 (+0.0)
Capital cost ($1,000)	57,793 (+2,546)	58,984 (+3,737)	59,079 (+3,832)
Profit ($1,000/yr)	34,716 (−318)	34,567 (−467)	34,555 (−479)
CPU time (s)	6.1	21.2	71.1

Table 5. Results from Steam Parameter Optimization

Steam turbine output (MW)	262.1 (+0.0)
Capital cost ($1,000)	58,751 (−328)
Profit ($1,000/yr)	34,596 (+41)
CPU time (s)	22.7

ference between saturation temperature and the inlet temperature of feedwater to evaporators) from 0°C to 20°C. The effect on economics of this change can be seen in Table 5.

Although the output remains virtually unchanged, the profit increases due to a slightly lower capital cost. This is mainly caused by the increased temperature differences in the economizers.

Summary for the Master Model Optimization. In addition to the structural issues and the parameter optimization, the optimization results of the master model also indicate the distribution of heating duties among the boilers and heat-recovery units. These are summarized in Table 6, and the optimized flowsheet is shown in Figure 12. The main design parameters and economic performance of the improved design (Figure 12) are summarized in Tables 7 and 8.

Table 6. Optimal Heat Duties of the Units (MW)

	HRSG	CFBC HRSG	CFBC	Syngas Cooler	FWH Train	Coolers
Condensate preheating					53.5	7.8
Economizer	173.5					25.9
Evaporation	31.8	14.1	35	114.3		
1st Superheat	83.1		25.0			
2nd Superheat		47.2				
1st Reheat			44.6			
2nd Reheat			35.5			

Table 7. Main Parameters from Master Model Optimization

Steam turbine output (MW)	262.1
Steam pres. (HP/reheat/deaerator) (bar)	140/35/1.7
Superheated steam flow (kg/s)	186.5
Reheated steam flow (kg/s)	175.5
Condensate flow through condensate heating train (kg/s)	158.0

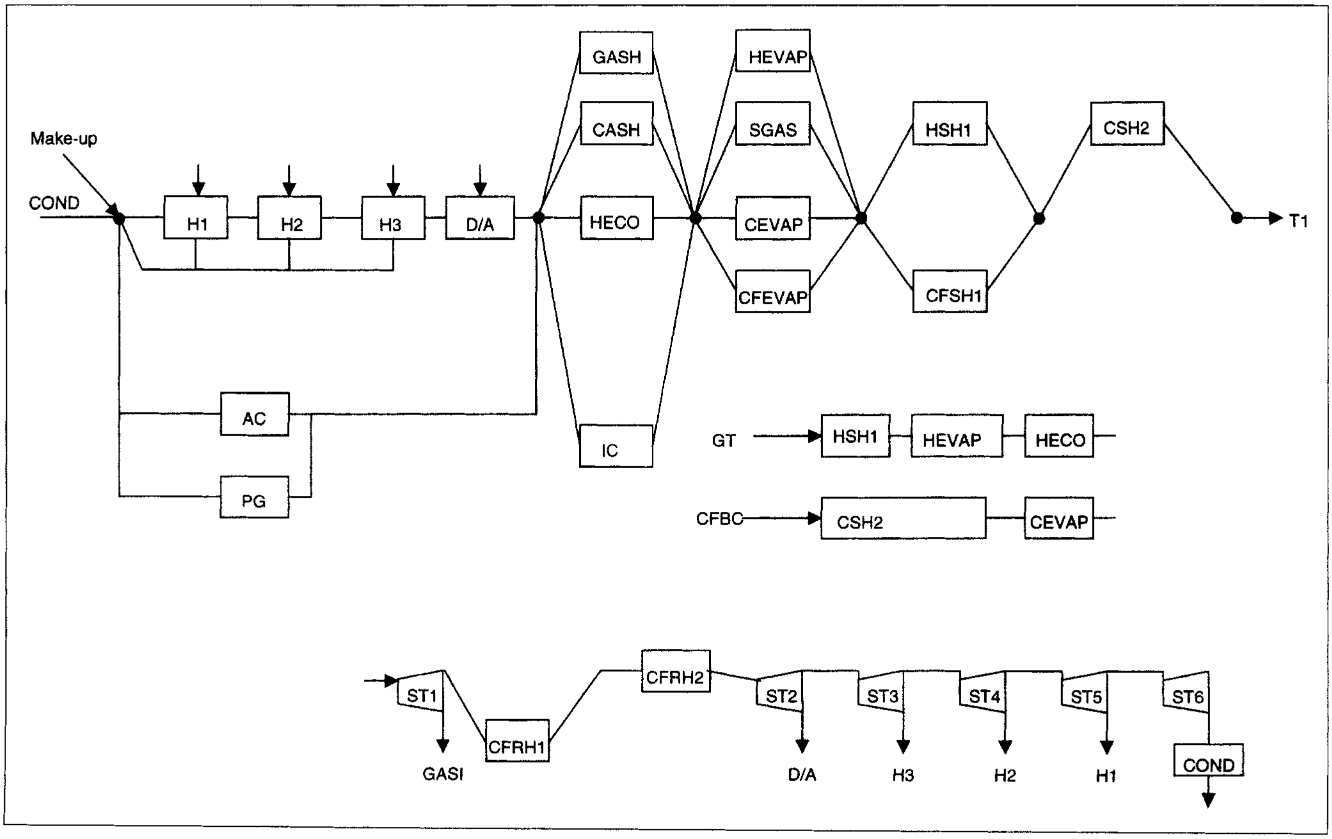

Figure 12. Final flowsheet.

Table 8. Economic Parameters from Master Model Optimization

Revenue ($1,000/yr)	41,940
Total capital cost ($1,000/yr)	7,344
Profit ($1,000/yr)	34,596
Capital cost of HRSG ($1,000)	28,887
Capital cost of CFBC HRSG ($1,000)	6,094
Capital cost of steam turbine ($1,000)	16,622
Capital cost of condensate preheaters ($1,000)	1,230
Capital cost of coolers ($1,000)	5,918

Conceptual design for subsystems

The master model determines the overall duties of the subsystems but not their internal structural and parameter arrangement. Here we need to design the following subsystems:

1. HRSG
2. CFBC HRSG
3. Condensate heating train and steam turbine

In the case of designing HRSGs (items 1 and 2), the total enthalpy changes of water and steam streams and their inlet and outlet temperatures are specified by those determined in the master model. The objective is to design HRSGs with a minimum capital cost for the heat exchangers in the HRSGs.

The condensate preheating train is strongly linked to the steam turbine, and hence they must be considered simultaneously. The steam inlet conditions, condensing temperature, and final feedwater temperature are fixed. The objective is to maximize the annual profit for the design of item 3.

The way to solve these subsystems is to build a superstructure for each subsystem first, and then optimize the superstructure to give an optimal solution. The optimized structures for HRSG and CFBC HRSG are given in Figure 13.

Condensate Preheating Train and Steam Turbine. A superstructure containing the intermediate and low-pressure sections of the steam turbine and the condensate preheaters is constructed based on the principle shown in Figure 5. The main design aspects to be optimized are

1. Heater type (direct or indirect contact)
2. TTD of the heaters
3. Condensate return
4. Structure of preheat train and heater duties.

The optimization of the superstructure is performed to find the most economic arrangement, and the optimal structure is illustrated in Figure 14. Optimized results are given in Table 9. The most profitable arrangement is to have the heaters as direct contact heaters despite the work penalty (around 60 kW per pump) caused by having additional pumps.

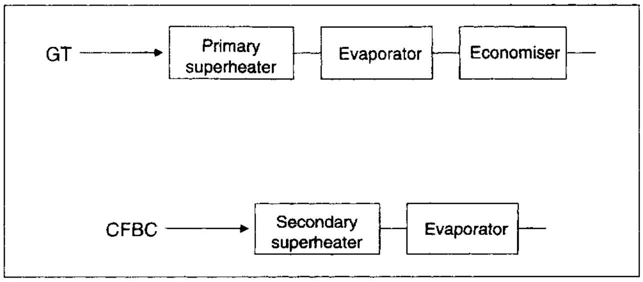

Figure 13. Conceptual design for HRSG and CFBC HRSG.

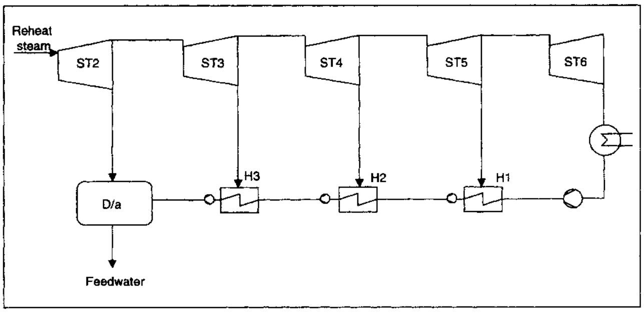

Figure 14. Conceptual design for condensate preheating train.

Summary of Conceptual Design. The subsystems were able to satisfy the targeted heat loads without producing overly complicated designs. Had this not been the case, we would again introduce additional constraints to the master model and optimize it again to get the new optimal heat-load distribution. The main economic parameters after the final stage of the design are summarized in Table 10.

The increase in capital cost for the heat-recovery units is caused by the fact that the cost for individual heat exchangers needs to be calculated, while in the master model some of the exchangers were lumped together.

Conclusions

A methodology for flowsheet synthesis of thermal systems has been presented. The methodology combines the physical insights of thermodynamic analysis and the computational power of mathematical programming.

The main challenge for solving flowsheet synthesis problems is to identify all the relevant design options in order to

Table 9. Results from Condensate Preheat Optimization

Steam turbine output (MW)	262.4 (+0.3)
Pump work penalty (MW)	0.2
Net output (MW)	262.2 (+0.1)
TTD of heater 1 (°C)	0.0
TTD of heater 2 (°C)	0.0
TTD of heater 3 (°C)	0.0
Capital cost of heaters and pumps ($1,000)	1,026
CPU time (s)	4.8

Table 10. Economic Parameters from Conceptual Design

Steam turbine output (MW)	262.2 (+0.1)
Revenue ($1,000/yr)	41,952 (+12)
Total capital cost ($1,000/yr)	7,302 (−42)
Profit ($1,000/yr)	34,650 (+54)
Capital cost of HRSG ($1,000)	29,109 (+222)
Capital cost of CFBC HRSG ($1,000)	6,444 (+350)
Capital cost of steam turbine ($1,000)	15,918 (−704)
Capital cost of condensate preheaters ($1,000)	1,026 (−204)
Capital cost of coolers ($1,000)	5,918 (+0)

keep the computational complexity at an acceptable level. The proposed approach manages to tackle this problem by adopting the level-by-level design strategy with the help of thermodynamic analysis and a feedback mechanism for updating the superstructure with newly generated design options.

In this level-by-level design methodology, the design process starts with a small amount of information and options. The level of detail increases as the design process moves on when more data and options are available. Major aspects with a large impact on the overall economics are determined in the earliest stage, and are modified with design details that arise in the later stages. Although the design process is sequential, high-level major structural options and key variables are optimized simultaneously with lower level design details. With this new approach, the user can interact with the design process at any point. In the loop of the master model optimization, the designer is allowed to be involved in the selection of structural changes. Although change options are determined mainly based on the thermodynamic analysis, heuristics and the designer's own knowledge and experience can be employed as well. Therefore the designer is involved in updating the superstructure with the options he or she sees relevant.

Overall, this methodology reflects the nature of the design process and improves on the current design practice. Following this methodology, a complex problem can be solved efficiently without sacrificing the solution quality. Although this method is developed for thermal system design, the philosophy and basic ideas behind this method are applicable to the design of other systems.

Acknowledgments

The authors thank Dr. Chris Green and David Rowe from the Power Technology Centre of PowerGen plc for their support to this work.

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Appendix A: Nomenclature for Mathematical Models

Master model

General Indices. The following indices will be used:

I = {*i*/process units}
J = {*j*/process streams}
K = {*k*/parameter set}
T = {*k*/steam turbines}

Process Unit Sets. All these process unit sets, which are treated differently in modeling, belong to the general set I. However, there are some common equations used for every process unit, that can be dealt with conveniently with the definition of general set I.

BOIL = {*i*/simple boiler; BOIL ⊂ I}
BX = {*i*/boiler heat exchangers; BX ⊂ I}
COMB = {*i*/combustor; COMB ⊂ I}
COMP = {*i*/compressors; COMP ⊂ I}
COOL = {*i*/coolers; COOL ⊂ I}
EXP = {*i*/gas expanders; EXP ⊂ I}
FIXQ = {*i*/units with fixed heat duty; FIXQ ⊂ I}
FIXW = {*i*/units with fixed work; FIXW ⊂ I}
FWD = {*i*/direct-contact feedwater heaters; FWD ⊂ I}
FWI = {*i*/indirect-contact feedwater heaters; FWI ⊂ I}
GT = {*i*/gas turbines; GT ⊂ I}
HR = {*i*/boiler sections; HR ⊂ I}
MIX = {*i*/mixer; MIX ⊂ I}
OM = {*i*/optional process units; OM ⊂ I}
PUMP = {*i*/pumps; PUMP ⊂ I}
SPLI = {*i*/splitters; SPLI ⊂ I}
TUR = {*i*/steam turbine sections; TUR ⊂ I}

Stream Sets. Similarly to process units, sets are defined for each type of stream.

FP = {j/streams with fixed enthalpies and temperatures; FP $\subset$ J}

NP = {j/streams with variable enthalpies and temperatures; NP = $\overline{\text{FP}}$; NP $\subset$ J}

FUEL = {j/fuel streams; FUEL $\subset$ FP}

MF = {j/streams with minimum flow constraint; MF $\subset$ J}

TM = {j/streams with minimum temperature requirement; TM $\subset$ NP}

Connection Sets

IN = {($i \in$ I, $j \in$ J)/streams j to process unit i}

OUT = {($i \in$ I, $j \in$ J)/streams j from process unit i}

BLEED = {($i \in$ FWI, $j \in$ FP)/bleed-steam stream j to indirect heat exchanger i}

COUT = {($i \in$ FWI, $j \in$ FP)/condensate stream j from indirect heat exchanger i}

FWIN = {($i \in$ FWI, $j \in$ FP)/feedwater stream j to indirect heat exchanger i}

FWOUT = {($i \in$ FWI, $j \in$ FP)/feedwater stream j from indirect heat exchanger i}

AIN = {($i \in$ COMB, $j \in$ NP)/air stream j to combustor i}

FUIN = {($i \in$ COMB, $j \in$ FUEL)/fuel stream j to combustor i}

REC = {($i \in$ HR, $i \in$ BX)/boiler heat exchanger i assigned to heat recovery unit i}

TC = {($i \in$ TUR, $t \in T$)/turbine section i assigned to steam turbine t}

Parameters

a_j^h = correlation coefficient for enthalpy of stream $j \in NP$ (kJ/kg·K)

b_j^h = correlation constant for enthalpy of stream $j \in NP$ (kJ/kg)

c^e = cost of electricity (\$/kWh)

c_i^{gt} = capital cost of gas turbine $i \in GT$ (\$1,000)

c_i^f = fuel cost for process unit $i \in$ (BOIL, GT) (\$/kWh)

c_j^f = fuel cost for fuel stream $j \in$ FUEL (\$/kWh)

f^a = capital cost annualization factor (yr^{-1})

f_j^f = fuel/air ratio for fuel $j \in$ FUEL (kg fuel/kg air)

$\tilde{h}_{j,k}$ = specific enthalpy of stream $y \in$ FP for parameter set $k \in$ K (kJ/kg)

LHV$_j$ = lower heating value of fuel stream $j \in$ FUEL (kJ/kg)

m_i^{gt} = exhaust mass flow of gas turbine $i \in$ GT (kg/s)

$m_j^{\min}$ = minimum mass flow for stream $j \in$ MF (kg/s)

Q_i^{gt} = fuel consumption of gas turbine $i \in$ GT (kW)

Q_i^{fix} = fixed heat input or output of process unit $i \in$ FIXQ (kW)

t = annual operating time (1,000 h/yr)

T_i^{gt} = exhaust temperature of gas turbine $i \in$ GT (°C)

$T_{i,k}^{\text{maxin}}$ = maximum inlet temperature to boiler section $i \in$ HR for parameter set $k \in$ K (°C)

$T_{i,k}^{\text{maxout}}$ = maximum outlet temperature from boiler section $i \in$ HR for parameter set $k \in$ K (°C)

$T_j^{\min}$ = minimum temperature for a stream $j \in$ TM (°C)

$\tilde{T}_{j,k}$ = temperature of stream $j \in$ FP for parameter set $k \in$ K (°C)

U_i = overall heat transfer coefficient for feedwater heater $i \in$ FWI (kW/m^2°C)

W_i^{fix} = fixed work input or output of process unit $i \in$ FIXW (kW)

W_i^{gt} = work output of gas turbine $i \in$ GT (kW)

α_i = capital cost coefficient for process unit $i \in$ [I − (GT, TUR)] (\$1,000/kg/s, \$1,000/kW, \$1,000/m^2, or \$1,000/kW/°C)

β_i = capital cost constant for process unit $i \in$ [I − (GT, TUR)] (\$1,000)

$\alpha_{t,k}^{\text{tur}}$ = capital cost coefficient for steam turbine $t \in$ T for parameter set $k \in$ K (\$1,000/kW)

$\beta_{t,k}^{\text{tur}}$ = capital cost constant for steam turbine $t \in$ T for parameter set $k \in$ K (\$1,000)

$\Delta T_i^{\min}$ = minimum allowable temperature difference in boiler section $i \in$ HR (°C)

ϕ_i = capital cost coefficient for heat-recovery unit $i \in$ HR (\$1,000/kg/s)

$\gamma_j = c_p/c_v$ of stream $j \in$ NP

Γ = large value

η_i = thermal efficiency of process unit $i \in$ BOIL

η_i^∞ = polytropic efficiency of process unit $i \in$ (COMP, EXP)

ψ_i = capital cost coefficient for heat-recovery unit $i \in$ HR (\$1,000/kg/s)

Continuous Variables

$A_{i,k}$ = area of feedwater heater $i \in$ FWI for parameter set $k \in$ K (m^2)

$c_{i,k}^{\text{cap}}$ = capital cost of process unit $i \in$ I for parameter set $k \in$ K (\$1,000)

$c_{t,k}^{\text{tur}}$ = capital cost of steam turbine $t \in T$ for parameter set $k \in$ K (\$1,000)

$h_{j,k}$ = specific enthalpy of stream $j \in$ J for parameter set $k \in K$ (kJ/kg)

$m_{j,k}$ = mass flow of stream $j \in$ J for parameter set $k \in$ K (kg/s)

OBJ = annual profit (\$1,000/yr)

$Q_{i,k}$ = heat input to or output from process unit $i \in$ I for parameter set $k \in$ K (kW)

$Q_{i,k}^{fw}$ = heat duty of feedwater heater $i \in$ FWI for parameter set $k \in$ K (kW)

$Q_{i,k}^{\text{fuel}}$ = fuel consumption of process unit $i \in$ (GT, BOIL) for parameter set $k \in$ K (kW)

$r_{i,k}$ = pressure ratio of process unit $i \in$ (COMP, EXP) for parameter set $k \in$ K

$T_{j,k}$ = temperature of stream $j \in$ J for parameter set $k \in$ K (°C)

$T_{i,k}^{\text{bleed}}$ = temperature of bleed stream(s) to feedwater heater $i \in$ FWI for parameter set $k \in$ K (°C)

$T_{i,k}^{\text{cout}}$ = temperature of condensate stream(s) from feedwater heater $i \in$ FWI for parameter set $k \in$ K (°C)

$T_{i,k}^{fw\,\text{in}}$ = temperature of feedwater stream(s) to feedwater heater $i \in$ FWI for parameter set $k \in$ K (°C)

$T_{i,k}^{fw\,\text{out}}$ = temperature of feedwater stream(s) from feedwater heater $i \in$ FWI for parameter set $k \in$ K (°C)

$T^{in}_{i,k}$ = temperature of stream(s) to unit $i \in$ (COMP, EXP, HR, SPLI) for parameter set $k \in$ K (°C)

$T^{out}_{i,k}$ = temperature of stream(s) from unit $i \in$ (COMP, EXP, GT, HR, SPLI) for parameter set $k \in$ K (°C)

$W_{i,k}$ = work input to or output from process unit $i \in$ I for parameter set $k \in$ K (kW)

$\Delta T1_{i,k}$ = hot-end temperature difference in process unit $i \in$ (HR, FWI) for parameter set $k \in$ K (°C)

$\Delta T2_{i,k}$ = cold-end temperature difference in process unit $i \in$ (HR, FWI) for parameter set $k \in$ K (°C)

$\Delta T^{LM}_{i,k}$ = log mean temperature difference in process unit $i \in$ (HR, FWI) for parameter set $k \in$ K (°C)

Binary Variables

y^f_j = controlling minimum flow for streams $j \in$ MF

$y^{gt}_{i,k}$ = selection of gas turbine $i \in$ GT for parameter set $k \in$ K

$y^o_{i,k}$ = selection of process module $i \in$ OM for parameter set $k \in$ K

y^p_k = selection of steam parameter set $k \in$ K

Feedwater heater model

The nomenclature of this model closely follows the one of the master model. Only the differences are reported here.

Process Unit Sets

FWH = {i/all feedwater heaters; FWH $\subset$ I}

FWH1 = {i/direct or indirect contact feedwater heaters; FWH1 $\subset$ FWH}

FWH2 = {i/indirect contact condensate coolers; FWH2 $\subset$ FWH}

Connection Sets

$C1$ = {($i \in$ FWH1, $j \in$ NP)/condensate stream j forwards from heat exchanger i}

$C2$ = {($i \in$ FWH1, $j \in$ NP)/condensate stream j backwards from heat exchanger i}

CIN = {($i \in$ FWH, $j \in$ NP)/condensate stream j to heat exchanger i}

Parameters

dp_i = pressure rise in a pump $i \in$ PUMP (bar)

$\tilde{T}^{sat}_j$ = saturation temperature of stream $j \in$ FP (°C)

ΔT^{min1}_i = minimum allowable cold side temperature difference in feedwater heater $i \in$ FWH (°C)

ΔT^{min2}_i = minimum allowable hot side temperature difference in feedwater heater $i \in$ FWH (°C)

Continuous Variables

$c^1_i, c^2_i, c^3_i, c^4_i$ = auxiliary variables for capital cost of feedwater heater $i \in$ FWH (\$1,000)

F_j = auxiliary variable for mass flow of stream $j \in$ NP through a pump (kg/s)

T^{sat}_j = saturation temperature of stream $j \in$ FP (°C)

TS^{bleed}_i = saturation temperature of bleed stream(s) to feedwater heater $i \in$ FWH (°C)

Binary Variables

y^{fwh}_i = selection of type for feedwater heater $i \in$ FWH1

y^c_i = selection of condensate direction for feedwater heater $i \in$ FWH1

Appendix B: Constraints, Initial Assumptions and Notation for Case Study

Main constraints

Min. temp. difference for heat exchangers excluding condensate preheaters (°C)	25
Min. stack temperature (°C)	140
Max. steam temperature (°C)	538
Condenser pressure (bar)	0.05
Gas-turbine exhaust (°C)	554
Syngas from gasifier (°C)	965
Flue gas from CFBC (°C)	900

Assumptions

- One bleed heater (deaerator)
- Approach temperature for evaporators is 0°C
- CFBC cost is constant
- Superheat and reheat temperatures 450°C for the primary and 538°C for the secondary
- Overall heat transfer coefficient for feedwater heaters 3.7 kW/m$^2 \cdot$K
- TTD for feedwater heaters 3°C
- Pump work is neglected

Capital cost coefficients

Type	α_i	β_i	ϕ_i	ψ_i
Superheaters, reheaters, and economizers	3.481	0	21.276	1.184
Evaporators excl. syngas cooler	1.741	0	21.276	1.184
Syngas cooler	3.481	0	21.276	0
Other ash and air coolers	1.741	0	21.276	0
Feedwater heaters	1.02	0	0	0
Steam turbine (master model)	47.1–48.6	4276–4411		
Steam turbine (conceptual design)	322.567			

Notation

AC = air cooler

CASH = ash cooler for the CFBC

CECO = economizing section in CHRSG

CEVAP = evaporating section in CHRSG

CFECO = economizing section in CFBC

CFEVAP = evaporating section in CFBC

CFRH1 = primary reheating section of CFBC

CFRH2 = secondary reheating section of CFBC
CFSH1 = primary superheating section of CFBC
CFSH2 = secondary superheating section of CFBC
CRH1 = primary reheating section of CHRSG
CRH2 = secondary reheating section of CHRSG
CSH1 = primary superheating section of CHRSG
CSH2 = secondary superheating section of CHRSG
D/A = deaerator
GASH = ash cooler for the gasifier
H1–H3 = condensate preheaters
HECO = economizing section in HRSG
HEVAP = evaporating section in HRSG
HFWH = condensate preheating section in HRSG
HRH1 = primary reheating section of HRSG
HSH1 = primary superheating section of HRSG
IC = gas turbine intercooler
PG = pulse gas cooler
SGAS = syngas cooler
ST1–ST6 = steam turbine stages
TTD = terminal temperature difference

Manuscript received Apr. 12, 1999, and revision received May 22, 2000.