AN ABSTRACT OF THE THESIS OF

Jatila Ranasinghe for the degree of Doctor of Philosophy in Mechanical Engineering presented on April 10, 1989.

Title: Use of the Exergy Concept for Design Improvement of Heat Exchangers and Heat Exchanger Networks.

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The second law of thermodynamics, through the exergy concept, allows us to quantify and rationally cost the consumption of exergy (irreversibility) used to drive the heat exchange process and the effluent losses of exergy in a heat exchanger. For systems with a network of heat exchangers, the exergy concept recognizes that properly integrated heat pumps reduce the heat transfer irreversibility; this results in reduced utility consumption. Furthermore, heat engines properly integrated in heat exchanger networks recover a fraction of the thermodynamic potential destroyed during the heat transfer process and generate power at very high efficiencies.

Heat exchanger design conditions are initially characterized in this thesis and potential trade-off options are discussed. A modification to the irreversibility minimization method is proposed next, and the proposed method is shown to give more realistic guideposts for heat exchangers, compared to the corresponding guideposts

obtained from present methods. This thesis also proposes a method to obtain the irreversibility cost coefficients for heat exchangers residing in complex systems. The application of the modified irreversibility method proposed here, and the thermoeconomic method, are illustrated by optimizing an emerging technology ceramic heat exchanger residing in a complex power plant.

A method based on the exergy concept is developed to recognize the potential for improvement of processes with process integrated heat pumps and heat engines. Once potential processes have been identified, economically optimum load and level of integration have to be determined. The method of formulating the economic optimization problem is presented, and bounds for some design variables are finally developed.

Use of the Exergy Concept for Design Improvement of Heat Exchangers and Heat Exchanger Networks

by

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NOMENCLATURE

A	=	heat transfer surface area
c	=	unitary product cost
cp	=	specific heat
C	=	capacity rate
COP	=	coefficient of performance
C_t	=	total cost
d_	=	small change
EX	=	exergy
f(_)	=	function of
h	=	heat transfer coefficient and enthalpy
I	=	irreversibility
ID	=	inside diameter of heat exchanger tube
k	=	constant of propotionality
l	=	length of heat exchanger tube per single pass
LMTD	=	log mean temperature difference
npn	=	number of pipes normal to the gas flow direction
npp	=	number of pipes parallel to the gas flow direction
Nc	=	non dimensionalized cost objective function
Ns	=	entropy generation number
NTU	=	number of heat transfer units
Q	=	heat transfer, energy flow
t	=	inlet temperature ratio
t_{ap}	=	application life
T	=	temperature
U	=	overall heat transfer rate
W	=	work

Z = cost of capital expenditure

β = variable parameter

ϵ = heat exchanger effectiveness

ω = capacity rate ratio

ΔT = minimum approach temperature

ΔW = lost power output per unit of irreversibility

σ = entropy production

η = first law efficiency

φ = parameter used for process screening

ψ = exergetic efficiency

γ = material exergy parameter

<u>subscripts</u>

a = air-side

A = due to heat exchanger area

boil = evaporator of heat engine

c = cold stream, and cost

cc = cold composite

cf = counterflow

ch = chemical

cond = condenser of heat pump

cool = condenser of heat engine

CU = cold utility

e = electricity produced from power plant

evap = evaporator of heat pump

f = fuel

g = gas-side

h = hot stream

hc = hot composite

he = heat engine

hemhx = heat engine minus the heat exchangers

hp = heat pump

hpmhx = heat pump minus the heat exchangers

hx = heat exchanger

HEN = heat exchanger network

HU = hot utility

i = at the current state of technology

in = at inlet

m = material

max = maximum

mi = material, taking into account manufacturing process

min = minimum

out = at outlet

p = product

pi = process integrated

pf = parallel flow

r = reversible manufacturing process

t = total

TI = temperature interval

Δp = due to pressure drop

ΔT = due to heat transfer

0 = dead state

<u>superscripts</u>

$'$ = inlet

$''$ = outlet

.	=	rate
*	=	optimum
*L	=	local optimum

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CHAPTER 1

INTRODUCTION

The heat exchange process, whether performed in a single heat exchanger or a complex system of heat exchangers, represents an extremely important step in energy conversion and use. Analysis based on the first law of thermodynamics have shown that heat exchangers are quite often about 100% efficient. For this reason, they have been sometimes overlooked, to some extent, when trying to improve energy conversion processes. The second law of thermodynamics, through the exergy concept, allows us to recognize that heat exchangers can be far from 100% efficient and, in many areas, far from optimally designed.

The exergy of a system is a property which measures the potential of a system to cause change. Exergy is specifically defined as the maximum work which can be obtained from a system. Some exergy is destroyed in all real process: this is the essence of the second law of thermodynamics. The destruction of exergy is the irreversibility. As a process approaches ideality (reversibility) less irreversibility occurs, and evaluation of this irreversibility then shows how far from thermodynamic perfection a certain process is. Unlike exergy, energy is conserved. Thus the concept of design optimization based on exergy, rather than energy, is very plausible.[1]

One of the two primary ways in which exergy analysis assists in heat exchanger design is by pinpointing (including quantification) the consumption of exergy (irreversibility) used to drive the processes and the effluent losses of exergy. These are true losses which point the way toward improvement of a design. Another manner in which the exergy concept can be employed for design optimization of heat exchangers is with thermoeconomics. Exergy, being a true measure of the thermodynamic fuel value of any commodity, provides a common and rational basis for costing all flow streams, heat transports and work transfers in an energy conversion system. Hence, thermoeconomics can be used to optimize the trade off between the operating and capital costs in the system.

In systems with a network of heat exchangers, it is possible to further reduce the system irreversibility and the total cost of operation with properly integrated heat

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[1]It must be noted that exergy analysis is intended to complement, not to replace, energy analysis. All exergy analysis must incorporate the first law of thermodynamics.

pumps or heat engines. Process integrated heat pumps are capable of reducing the heat transfer irreversibility in the system, and the reduced irreversibility transforms into reduced utility cost. Process integrated heat engines recover a fraction of the thermodynamic potential destroyed during the heat transfer process, and generate shaft power at very high efficiencies. Hence, these two options also need to be considered when optimizing a system with a heat exchanger network.

The work presented in this thesis focuses on the use of the exergy concept for the design optimization of heat exchangers, and for the optimal synthesis of heat pumps and heat engines in processes.

The irreversibilities of a heat exchanger are initially delineated in this chapter, followed by a review of the current state of the technology of exergy concept based methods for heat exchanger optimization. Next, background information is presented on presently employed methods for the synthesis of heat pumps and heat engines in processes. Finally, the scope of the present thesis is presented.

THE IRREVERSIBILITIES OF A HEAT EXCHANGER

The total internal irreversibility of a heat exchanger can be written in terms of the entropy generation rate $\dot{\sigma}$ of the heat exchanger and the dead state temperature T_0 as[2]:

$$\dot{I}_{hx} = T_0 \dot{\sigma} \tag{1.1}$$

Another way of presenting the irreversibility of heat exchangers, proposed by Bejan (1977) and commonly used in the literature, is the non-dimensional entropy generation number Ns defined as:

$$Ns = \frac{\dot{I}_{hx}}{T_0 C_{max}} = \frac{\dot{\sigma}}{C_{max}} \tag{1.2}$$

The irreversibilities of a heat exchanger are due to:

(a) Heat transfer across a finite temperature difference.

(b) Pressure losses.

(c) Interaction with the environment.

(d) Streamwise conduction in the walls of the heat exchanger.

where (a), (b) and (d) are internal irreversibility terms, while (c) is an external irreversibility term. The various irreversibility terms are explained next.

Due to Heat Transfer

The irreversibility due to heat transfer is the principal form of exergy loss in a heat exchanger. This is due to the reduction in the quality of the thermal energy as

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[2] This equation does not account for external irreversibilities. Such irreversibility terms will have to be calculated separately, and added into equations (1.1) and (1.2), to obtain the total irreversibility of the heat exchanger.

it is transferred to a temperature closer to the dead state. In all real heat exchangers a finite temperature difference between the heat transfer media is required for a finite heat transfer rate, so heat transfer irreversibility is unavoidable. The structure of the heat transfer irreversibility has been investigated extensively in recent years by various authors (Bejan (1980), Seculic and Baclic (1984) and Bejan (1987)). The variation of the heat transfer irreversibility with NTU for various heat exchangers shows (see Fig. 1.1) an occurrence of a maximum in the irreversibility at a finite NTU, and a finite irreversibility when NTU approaches infinity. Designing heat exchangers with thermal duty less than the NTU value corresponding to the maximum irreversibility restricts the application of heat exchangers, and the minimum irreversibility on this side of the maximum corresponds to no heat exchange taking place. The finite heat transfer irreversibility in the infinite NTU limit is referred to as the flow imbalance irreversibility or the remanent irreversibility. The flow imbalance irreversibility has a finite value for all heat exchangers, and is equal to zero only for balanced (i.e: $\omega = C_{min}/C_{max} = 1$) counterflow heat exchangers. The remanent irreversibility for counterflow heat exchangers and parallel flow heat exchangers provide lower and upper bounds respectively for the remanent irreversibility of all heat exchangers. The remanent

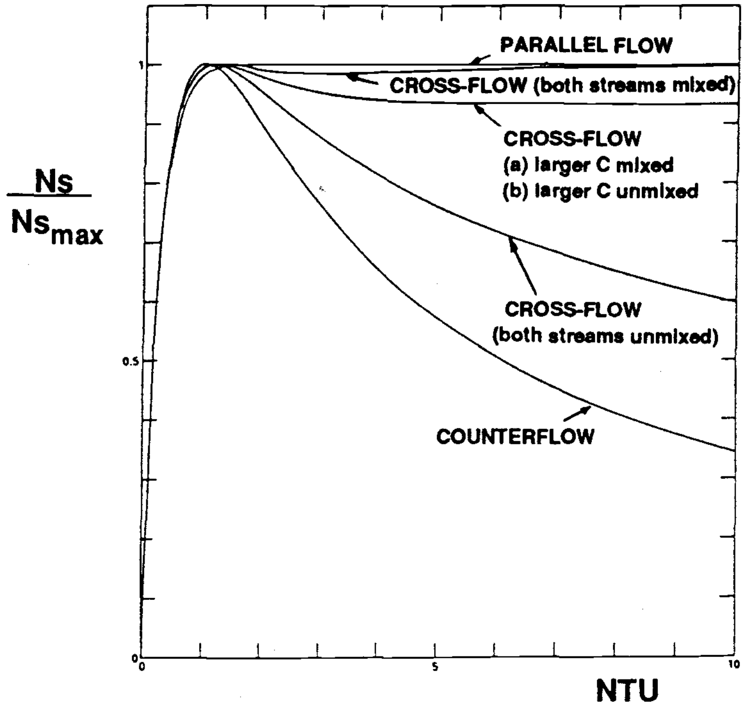

Figure 1.1 The structure of the heat transfer irreversibility of various heat exchanger configurations with τ = 0.5 and ω = 1 (after Sekulic and Baclic, 1984).

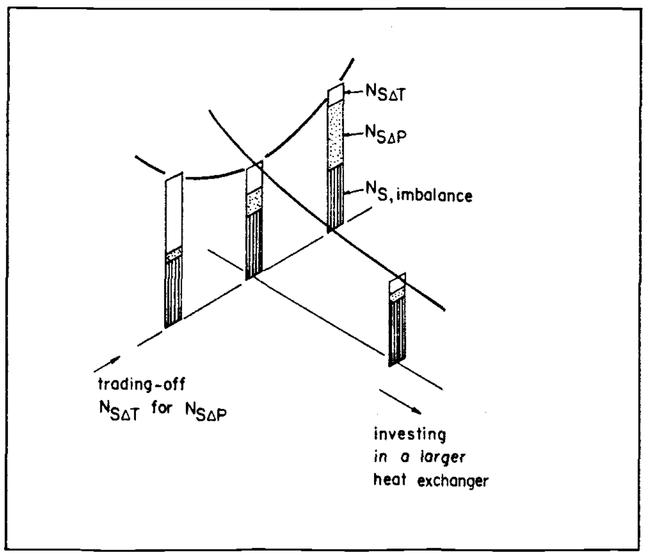

Figure 1.2. The structure of the total irreversibility generation in a heat exchanger (after Bejan, 1987)

irreversibility, for a given heat exchanger operating condition (i.e: for a fixed ω and τ, where $\tau = T_{c,in}/T_{h,in}$), also provides heat exchanger designers with a guidepost for the potential reduction of irreversibility possible with an improved design. A method to improve this guidepost is presented in a later chapter.

Due to Pressure Drop

In heat exchangers involving gaseous streams, the irreversibilities due to frictional losses can also be significant contributors to the total irreversibility of the heat exchanger. The pressure drop irreversibility of each side of the heat exchanger is a function of the flow velocity, surface geometry, heat transfer surface area etc. Hence, the pressure drop irreversibility and the heat transfer irreversibility are coupled to each other.

Due to Interactions with the Environment

There are three forms of irreversibility due to interactions with the environment. They are the irreversibilities generated due to heat transfer (stray losses) from the heat exchanger to the environment and, chemical and thermal diffusion of the exit streams into the environment. Only the diffusion terms are considered for the work presented in this thesis. For a heat exchanger with an outlet diffusing into the environment, the total exergy of the stream may be considered an irreversibility.

Due to Streamwise Conduction in the Walls

The irreversibility contribution due to axial conduction is generally more significant in short heat exchangers than in long ones, and at very low temperatures.

THE EXERGY CONCEPT BASED METHODS FOR HEAT EXCHANGER DESIGN

The two primary ways in which the exergy concept is used for heat exchanger design and optimization is with the irreversibility minimization method and thermoeconomic method. Finding the optimum balance between the various components of irreversibility, and determining the corresponding heat exchanger parameters, is the objective of the irreversibility minimization method for heat exchanger design. Thermoeconomic analysis recognizes that all irreversibility components are not of equal value, and, hence, assigns various costs for the different irreversibilities. Obtaining the heat exchanger parameters to minimize the total cost of operation is the objective of the thermoeconomic optimization of heat exchangers. The two methods are reviewed in the next two sub sections.

The Irreversibility Minimization Method

The irreversibility minimization method was first applied to heat exchangers by McClintock (1951). Various authors have since used this technique for heat exchanger analysis, particularly since the work of Bejan (1977,1980) where results regarding basic design variables were developed in a generalized, non-dimensional manner. The work in this area has been reviewed by Bejan (1987) and Ranasinghe et al. (1987).

The irreversibilities of a heat exchanger have been shown to consist mainly of those due to heat transfer and pressure drop. Thus, the traditional irreversibility rate based objective function for a heat exchanger Ns can be expressed as,

$$Ns = Ns_{\Delta p} + Ns_{\Delta T} \tag{1.3}$$

The heat transfer term has been further divided into those that can be reduced by additional surface area and those that result from the capacity mismatch ($Ns_{imbalance}$) of the heat exchanger streams (Bejan, 1987). These irreversibility terms are illustrated qualitatively in Fig. 1.2. This figure shows that $Ns_{\Delta T}$ and $Ns_{\Delta p}$ have a local optimum, but the global optimum corresponds to the infinite area heat exchanger limit. Hence, the use of an objective function with only the heat transfer and flow irreversibility terms gives large heat exchangers as the global optimum; an unacceptable result from an engineering design viewpoint.

The Thermoeconomic Method

Analyses based on irreversibility minimization provide valuable guides during the design process because reduced irreversibilities translate into reduced energy expenses. However, there are other costs associated with the operation of a heat exchanger, such as capital costs and maintenance costs, that have to be considered in a more detailed design process. Analyses that combine a second law analysis with a cost evaluation have been popularly titled thermoeconomic analysis.

Thermoeconomics has its origins in the work of Keenan (1932), who suggested that cost accounting should be based on exergy instead of energy. Later work includes both theoretical developments (Tribus et al., 1966; El-Sayed and Evans, 1970) and applications to systems (Obert and Gaggioli, 1963; Gaggioli, 1977; London and Shah, 1983).

The formulation of the objective function, optimization strategy and the calculation of the cost coefficients for the thermoeconomic optimization of heat exchangers are presented in the next two sub sections.

(a) Problem Formulation and Optimization:

In thermoeconomics, irreversibilities are expressed in terms of their costs. Here, the analysis focuses on expressing irreversibility and capital costs for the particular system or "isolated" zone to be optimized. In complex systems, the desire is to optimize the zone in a manner that coincides with optimum for the overall system. The objective function is expressed as the capital costs plus a penalty cost from the existing irreversibilities.

The objective function is (Evans et al., 1981; Tapia and Moran, 1986):

$$\dot{C}_t = \sum_i c_i \dot{I}_i + \sum_n \dot{Z}_n \quad (1.4)$$

where,

c_i - unit cost of irreversibility produced in process i

$\dot{I}_i$ - irreversibility produced in the process i

$\dot{Z}_n$ - zonal cost of capital expenditure and other associated costs.

The irreversibility cost coefficients represent the lost income of the overall system or the additional expense for the overall system, per unit of the corresponding irreversibility.

Optimization of an overall system, or the isolated zone, can be carried out provided that all the input exergy costs are known, and independent of the decision variables which are heat exchanger design parameters. The optimization scheme should be based on the following criteria. A decrease in the total cost caused by an increase in a variable means that the value of the variable should be increased to reach the optimum (or at least, a relative optimum, if the objective function has multiple critical points), or vice versa. The values of all the variables are changed accordingly until there is no change in the total cost for small changes in the decision variables, or until a restriction is reached. The magnitude of the change of a design variable during an iteration is dependent on the selected optimization scheme.

Overall systems are usually complex, involving large interactions among their different components. Evaluating the change in overall costs for each change in a decision variable may be a difficult, time consuming process. These difficulties have caused many authors to decompose the system into subsystems, which can be optimized individually with relative ease. The heat exchanger is treated as the subsystem to be optimized, when the objective is to determine the optimum heat exchanger design and operating parameters. The objective function for a subsystem is identical to that of Equation (1.4), where c_i represents the unit cost of the irreversibility generated due to the irreversibility generation mechanism i of the heat exchanger (i.e: due to heat transfer, pressure drop etc.). The optimization approach is also similar. A small increment in a decision variable produces a small change in the irreversibility generation $d\dot{I}_i$ and the capital cost $d\dot{Z}$, so that the total subsystem cost change is:

$$d\dot{C}_t = \sum_i c_i \, d\dot{I}_i + d\dot{Z} \qquad (1.5)$$

The independent variable is then changed according to the sign of $d\dot{C}_t$ as discussed previously. However, the decomposition method is complicated by two facts that do not appear in an overall system optimization. These are:

* The cost coefficients c_i appearing in Equation (1.5) represent marginal costs (i.e. the unit cost of a differential amount of irreversibility increase). These costs may be difficult to calculate in many systems.
* The interactions with other plant elements cause the marginal cost coefficients c_i to change as the independent variables are changed.

Optimization of the system requires then a double iteration process. First, the decision variables are changed until an optimum is reached for the given cost coefficients. Then, new cost coefficients have to be calculated. The iteration has to be repeated until the cost coefficients do not change from one iteration to the next.

Although marginal costs do change between iterations, Frangopoulos and Evans (1984) have reported that the changes are small. They report this cost invariance as a major advantage of optimization methods based on exergy, as opposed to conventional optimization methods. A discussion on cost coefficients is included next.

(b) Cost Coefficients:

Calculation of the marginal costs for optimization requires the use of correlations for capital cost of the system components as a function of the decision variables. These correlations do not exist in general, or are expressed in terms of parameters not convenient for the analysis. If the correlations are given in an adequate form, the cost coefficients can be calculated by differentiation. The expressions for a simplified case are shown by El-Sayed and Gaggioli (1987).

A procedure to evaluate the marginal costs is developed in a later chapter. The procedure could be used to calculate marginal costs if a simulation code for the total system exist. The procedure consists of introducing a small increase to the irreversibility components, one at a time, and observing how these changes affect the output. The change in plant income per unit of irreversibility increase yields an approximation to the marginal cost.

IRREVERSIBILITY REDUCTION THROUGH PROCESS INTEGRATION

Properly integrated heat pumps or heat engines have the potential to reduce the cost and energy expenditure in some processes. Heat pumps are integrated to reduce the utility consumption, while heat engines are integrated to exploit the thermal energy degradation occurring in the heat exchange processes, to develop power at a high efficiency. The utilization of heat pumps and heat engines in a process plant should be analyzed in relation to the total plant. The pinch technology method, detailed by Linnhoff and co-workers (Linnhoff and Hindmarsh, 1983; Linnhoff and Ahmad, 1986. The method is briefly explained in Appendix A), is useful for analyzing this type of problem. A network of heat exchangers can be optimally synthesized with this method, if the initial and target temperatures and the energy content of all the streams are specified. However, the optimum temperature levels of integration and the heat loads of the working fluid streams of the evaporator and condenser of the heat pumps or the heat engines are not known. Once these optimum loads and levels of integration have been determined, the pinch technology method can be used to optimally synthesize the total heat exchanger network.

The optimum level of integration and heat loads for heat engines and heat pumps corresponds to the minimum total cost of operation. Cost optimization requires the process integration problem to be evaluated together with the utility producing section of the power plant. This is because the utility cost is a function of the utility consumption for the process. Therefore, determining the economically optimum load and level of integration becomes a complicated task. Also, all processes are not necessarily improved with process integrated heat engines or heat pumps. Hence, it is useful to have a preliminary screening procedure to determine the potential of a process for improvement with process integration.

Some rules for the preliminary screening of processes, to determine the potential for improvement with heat pump integration, have been given by Townsend and Linnhoff (1983). These authors showed that only processes with a pinch are candidates for improvement with heat pump integration. The process grand composite curve has been used by Renade et al. (1986a, 1986b) to determine the potential improvement possible with a process integrated heat pump. The method is based on selecting an approximate level of integration from a visual inspection of the process grand composite curve. The coefficient of performance of a heat pump corresponding to the approximate levels of integration is then used as an indicator of the economic potential. The method is strongly dependent on the intuition of an expert, and the method is not very reliable in determining the potential of integrating a heat engine as an alternative to a heat pump. Also, to the authors knowledge, the procedure for the economically optimum synthesis of heat pumps and heat engines in process plants have not been publicized.

SCOPE OF THE PRESENT THESIS

Heat exchanger design conditions need to be characterized to gain a better insight to the various trade-off possibilities during the design process. Heat exchanger design conditions are characterized in Chapter 2, and potential trade-off options are identified. Additional design constraints resulting from the interactions with other system components are also illustrated with an example.

One of the objectives of the present work is to develop an approach to obtain realistic performance limits for heat exchangers. The inclusion of the material exergy in the irreversibility rate based objective function is proposed in Chapter 3. This circumvents the unrealistic result of an infinite area heat exchanger being the global optimum. The resulting designs are also shown to be conceptually beneficial guideposts in designing heat exchangers.

An extended application of the modified irreversibility rate based objective function and the thermoeconomic objective function, to optimize heat exchangers in complex systems, is presented in Chapter 4. A method to obtain the incremental costs of an isolated component in a complex power plant is also presented in this chapter.

A method for the preliminary screening of processes with a network of heat exchangers, to recognize potential improvement with a process integrated heat pump or heat engine, is developed in Chapter 5.

The procedure of formulating the economic optimization problem, to determine the optimum load and level of integration of heat pumps or heat engines in a process plants, is presented in Chapter 6. Constraints for the design variables are also developed in this chapter.

The conclusions drawn from the present work, and potential related areas for future research are given in Chapter 7.

CHAPTER 2

THE DESIGN PROBLEM IN FIXED-DUTY VS. NON FIXED-DUTY HEAT EXCHANGERS

The objective of this chapter is to gain an insight to the various irreversibility and capital cost trade-off possibilities in heat exchanger design. Recognizing the trade-off possibilities allows a designer to properly formulate the objective function for design optimization, accurately cost the irreversibilities if a cost objective function is selected and also clearly understand the physical process which drives the system towards an optimum.

THE DESIGN CONDITIONS

For the present investigation, heat exchanger design conditions are divided into two main categories based on the specified operating conditions[3]. The first class of heat exchanger design condition considered is when the total heat transfer rate $\dot{Q}$, capacity rates of the two streams C_{min} and C_{max}, and two of the four inlet or outlet temperatures of the streams are specified. These conditions become specified due to constraints of the system in which the heat exchanger resides. This corresponds to a heat exchanger

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[3] Although the philosophy of the material in this chapter is general, the specifics have been developed considering only two stream heat exchangers with both streams being single phase or with one stream being two phase but without any subcooling or superheating.

with the position of both the hot and cold streams fixed on a Temperature-Energy (T-E) diagram such as Fig. 2.1. Here, the stream being cooled is referred to as the hot stream and the stream being heated is referred to as the cold stream. This constrained heat exchanger design condition is referred to as "fixed duty". The second class of heat exchanger design condition considered is where one or more of the specified operating conditions of the fixed-duty conditions is treated as a decision variable in the design. Thus, one or more of $\dot{Q}$, C_{min}, C_{max} and two of the four inlet or outlet temperatures of the streams are design variables. For the present investigation, one of the two temperature design variables is replaced by UA, since this is a more directly controllable design variable for heat exchangers. (This is possible due to the equality relationship $\dot{Q} = UA \times LMTD$.) This second class of heat exchanger design condition is termed "non-fixed duty".

The design of a heat exchanger requires the stipulation of many variables. Some of these variables pertain exclusively to the heat exchanger (the heat exchanger geometry variables) while others pertain to the system as a whole (mass flow rates, temperatures, etc.). Possible design variables for the fixed-duty design condition pertain exclusively to the heat exchanger. The degrees of freedom in the design problem is then equal to the selected heat

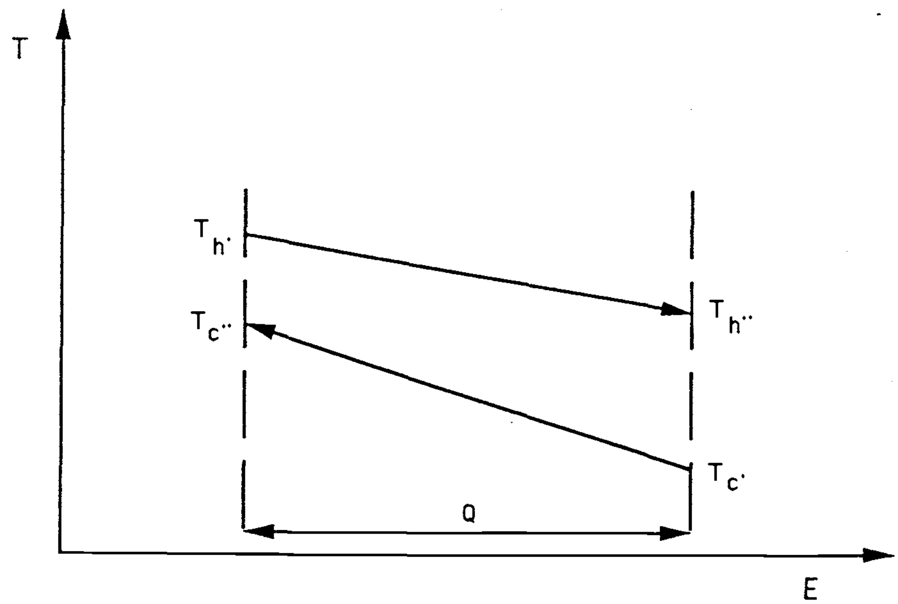

Figure 2.1. The T-E diagram for a heat exchanger with two single phase streams.

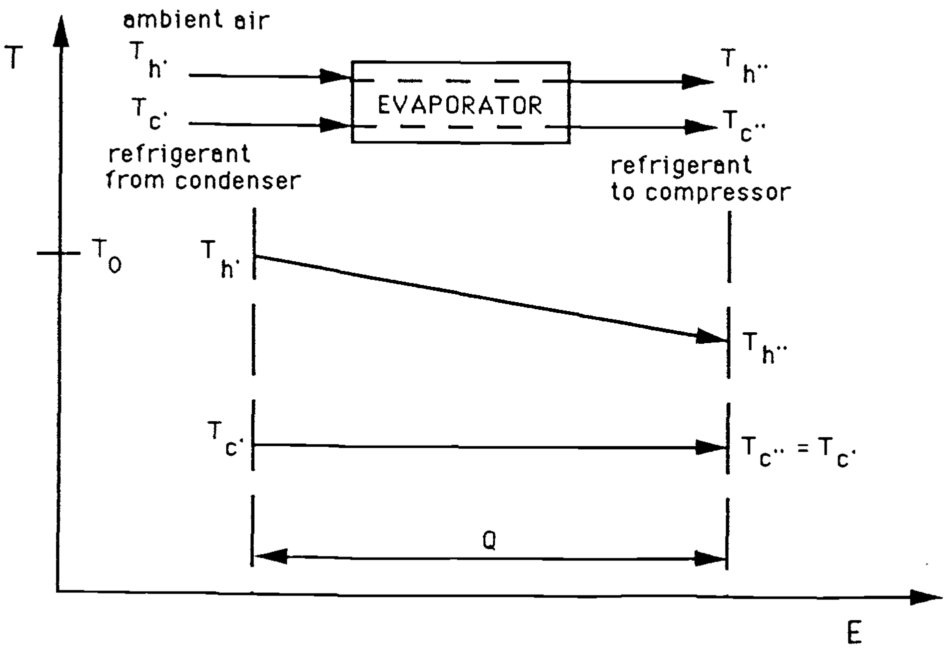

Figure 2.2. The evaporator of a heat pump, and the T-E variation of the hot and cold fluids.

exchanger geometry variables minus the number of equality constraints for the specific design problem. The maximum additional degrees of freedom of a non fixed-duty heat exchanger design condition compared to a fixed-duty condition is five. The five additional variables chosen for the present work are, capacity rates of the two streams C_{min} and C_{max}, the heat transfer rate $\dot{Q}$, one of the four inlet or outlet temperatures of the two streams, and the product UA. These additional variables pertain to the system as a whole. The additional degrees of freedom for most practical design conditions are less than five due to some of the variables being specified directly, or because additional equations relating the heat exchanger design variables are needed to satisfy constraints of other components in the system (an example of such an equality constraint is presented in the section "non fixed-duty heat exchangers").

To make the results more general and not specific to a particular heat exchanger or selected geometry variables, the trade-off options outlined in the next two sections are related to the corresponding change of common heat exchanger operating or design parameters. However, some of the variables considered in the next two sections (ex: heat transfer coefficient, heat transfer area) are controlled during the design process with specific heat exchanger geometry variables. A specific heat exchanger optimization is also presented in a later chapter titled "design optimization of a heat exchanger in a complex system."

FIXED-DUTY HEAT EXCHANGERS

As indicated above in Fig. 2.1 for two single phase streams, the fixed-duty conditions specify the position of the two streams on the T-E diagram. Hence, a specified amount of energy has to be transferred across a fixed temperature difference, and hence the heat transfer irreversibility, $\dot{I}_{\Delta T}$, is a constant for this fixed duty unit.

Since the energy transfer rate $\dot{Q}$ and LMTD are fixed for the present case, the product UA is also fixed. However, the overall heat transfer coefficient U and the total heat transfer surface area A can be adjusted while keeping the value of the product constant. Trade-off possibilities for a fixed-duty heat exchanger are as follows:

<u>Increase A and decrease U while keeping UA constant</u>

* Area increases.
* The total pressure drop irreversibility decreases.

The trade-off in this case is between the additional area and the pressure drop. The use of an objective function which accounts for only the flow and heat transfer irreversibility terms, such as the traditional irreversibility rate based objective function, would give a very large area heat exchanger as the global optimum. The same trade-off possibilities exist if U is increased and A is decreased while keeping UA constant.

Increase h_h and decrease h_c while keeping U and A constant

* Area remains constant.
* $\dot{I}_{\Delta ph}$ increases and $\dot{I}_{\Delta pc}$ decreases.

The trade-off in this case is between the frictional losses on the hot and cold sides respectively. Hence, the optimum pressure drops for the two sides of the heat exchanger can be obtained. The same trade-off possibilities exist if h_c is increased and h_h decreased while keeping U and A constant.

NON FIXED-DUTY HEAT EXCHANGERS (AN EXAMPLE)

The specific example of an evaporator of a heat pump is considered next to illustrate some of the trade-off possibilities in non-fixed duty heat exchangers. Only possible trade-off options useful to establish the optimum values of the additional degrees of freedom are discussed in this section. This then results in a fixed-duty heat exchanger. Thus when these optimum values have been established, further improvement of the now fixed-duty heat exchanger is possible as explained in the previous section.

For a heat pump, it can be assumed for purposes here that there is no superheating at the exit of the evaporator and the entire refrigerant in the evaporator is two phase. The condenser temperature and the subcooling, the ambient air temperature, and the displacement of the compressor are also assumed to be constant.

Since, $T_{h'}$, and C_c for the evaporator are fixed (C_c is approximately equal to infinity), the number of additional degrees of freedom for the present case compared to a fixed duty condition is three. The three additional degrees of freedom are $\dot{m}_h$, $\dot{Q}$ and UA (cp_h is assumed to be constant, and therefore $\dot{m}_h$ is treated as a variable instead of $(\dot{m}cp)_h$). The T-E diagram for a specified set of the three variables is illustrated in Fig. 2.2.

Since we restrict our attention to a heat pump with a fixed displacement compressor, and assume a constant condenser operating temperature and subcooling (for convenience only), a functional relationship between the evaporator heat transfer rate and the other two design variables can be established as follows: Due to the fixed compressor displacement,

$$\dot{m}_c = f(T_{c''}) \tag{2.1}$$

also,

$$\dot{Q} = \dot{m}_c(h_{c''} - h_{c'}) \tag{2.2}$$

where $h_{c'}$ and $h_{c''}$ are the enthalpies of the cold fluid at the inlet and outlet, respectively. Since $h_{c'}$ is fixed and $h_{c''}$ is a function of $T_{c''}$,

$$\dot{Q} = f(T_{c''}) \tag{2.3}$$

since $T_{c''}$ is also a function of UA and $\dot{m}_h$, this allows the functional relationship,

$$\dot{Q} = f(UA, \dot{m}_h) \tag{2.4}$$

The above equation is an equality constraint relating the design variables, thus reducing the degrees of freedom of the problem to two. This equality constraint was necessary to satisfy a constraint in the compressor.

The mass flow rate of the hot fluid $\dot{m}_h$ and UA are selected as the two independent variables. The trade-off possibilities for the two cases when $\dot{m}_h$ is a constant and UA is a variable, and when UA is a constant and $\dot{m}_h$ is a variable are considered next.

For increased UA at fixed $\dot{m}_h$

When UA is increased, LMTD decreases and as a result $T_{c'}$ increases. The new position of the hot and cold streams on the T-E diagram is indicated by the broken lines in Fig. 2.3. The slope of the hot stream does not change, but $T_{h''}$ decreases as a result of the increase in $\dot{Q}$. Although $\dot{Q}$ increases, $\dot{I}_{\Delta T}$ per unit heat transfer rate decreases[4] because energy transfer is now from a temperature closer to the dead state. The trade-off options for this case are as

[4] The increased heat transfer rate results in the decrease in the time of operation of the heat pump, for a fixed heat load.

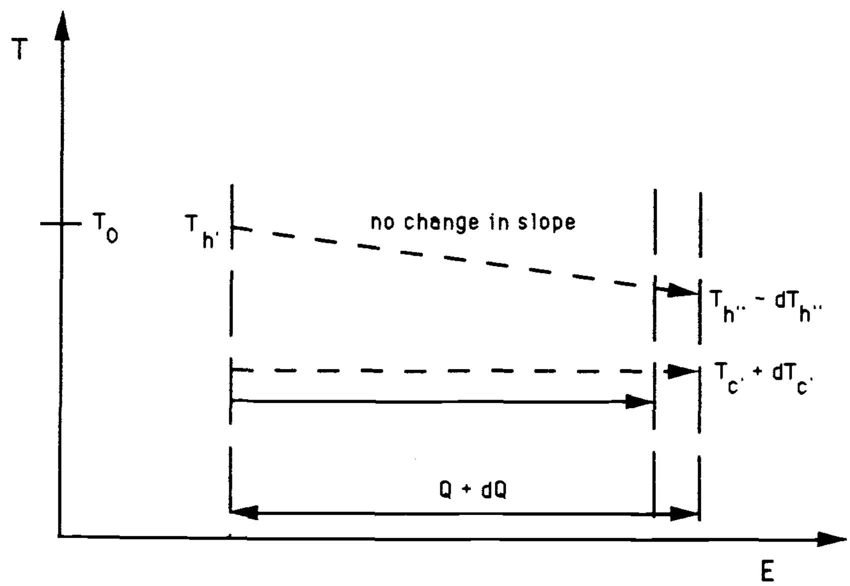

Figure 2.3. The change of the hot and cold streams of the evaporator for the case of fixed m_h and increased UA. The new positions are indicated by the broken lines and the old positions by the solid lines.

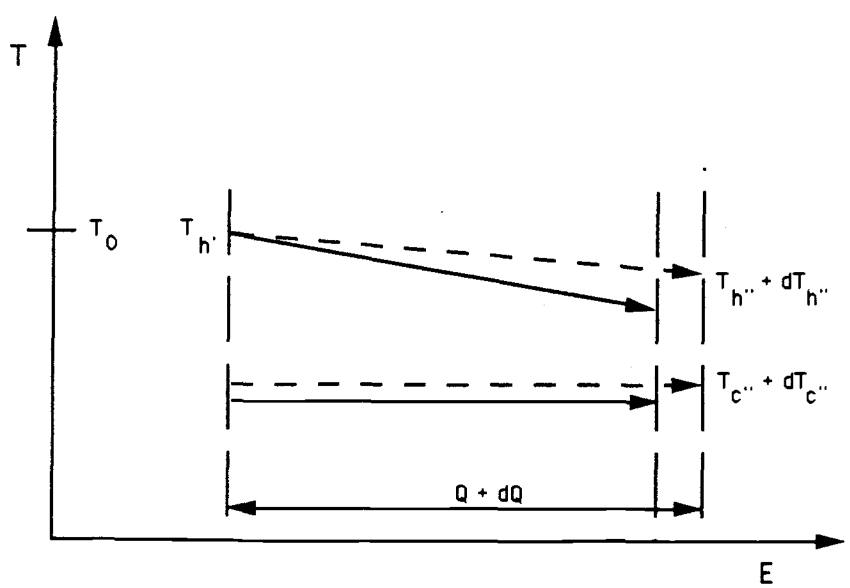

Figure 2.4. The change of the hot and cold streams of the evaporator for the case of fixed UA and increased m_h. The new positions are indicated by the broken lines and the old positions by the solid lines.

follows:

* $\dot{I}_{\Delta T}$ per unit heat transfer rate decreases.
* $\dot{I}_{\Delta ph}$ increases.
* No significant change in $\dot{I}_{\Delta pc}$.
* Area A may increase.
  - The other possibility is to increase U while keeping A constant.
* $\dot{I}_{effluent}$ to the atmosphere remains approximately constant.
* $\dot{I}_{sys,exterior}$ decreases.
  - The exergy of the cold stream at the outlet of the heat exchanger decreases.

Since the exergy of the cold stream decreases at the heat exchanger exit, the interactions with other system components have to be considered during the optimization.

For increased $\dot{m}_h$ at fixed UA

Increased $\dot{m}_h$ results in the increase in $T_{h''}$. Hence, for a constant UA, $T_{c'}$ should increase. Heat transfer rate $\dot{Q}$ increases in this case too. The new positions of the hot and cold streams are indicated in Fig. 2.4 as broken lines. Since energy transfer now is between two streams closer to the dead state temperature, $\dot{I}_{\Delta T}$ per unit heat transfer rate decreases. The resulting trade-off possibilities are as follows:

* $\dot{I}_{\Delta T}$ decreases as a result of energy transfer taking place

at temperatures closer to the dead state temperature (which is considered to be atmospheric for this case) and increases because of larger temperature differences.

* $\dot{I}_{\Delta ph}$ increases.
* $\dot{I}_{\Delta pc}$ remains approximately constant.
* Heat exchanger area decreases.
  - Increase in $\dot{m}_h$ would cause U to increase, and A should be reduced to keep UA constant.
* $\dot{I}_{effluent}$ to the atmosphere decreases.
* $\dot{I}_{sys,exterior}$ decreases.

Here too, the impact on the other plant components resulting from the decrease in the exergy of the cold fluid at the heat exchanger outlet needs to be considered during the heat exchanger design.

DISCUSSION

Characterization of heat exchanger design conditions allow potential trade-off options to be presented in a generalized form. Heat exchanger design conditions were characterized in this chapter, and potential trade-off options were discussed. Since, heat exchangers are always a part of a larger thermal system, their interaction with other plant components need to be considered during the design process. Additional design constraints resulting from such interactions were illustrated by considering the evaporator of a heat pump.

CHAPTER 3

THE CONCEPT OF INCLUDING THE MATERIAL IRREVERSIBILITY IN THE IRREVERSIBILITY MINIMIZATION METHOD

As discussed in chapter 1, the traditional irreversibility minimization method does not account for the exergy or economic cost of the heat exchanger material, thus resulting in unacceptably large area heat exchangers being the global optimum. This unrealistic result diminishes the value of the resulting guideposts obtained from this method.

This chapter presents an extension of the minimum irreversibility generation analysis wherein the irreversibility generation equation includes a term that takes into account the exergy value of the heat exchanger materials and the application life of the heat exchanger.[5] This method yields optimal designs which do not change with time or location and which represent the desired optimal design in the limit of exergy costs being dominant relative to labor and profits. The resulting optimal designs are physically more realistic than those obtained from the traditional irreversibility minimization method, and although the designs do not represent the optimal design from a thermoeconomic viewpoint, they represent limits that

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[5] The concept of charging a device for an energy (or exergy) term that considers the materials of the device is not new. It has been considered in various manners, principally as "net energy analyses." The focus of this chapter is to develop a general, thermodynamically rigorous evaluation that can be widely accepted.

may serve as guideposts for design. Such guideposts are quite useful conceptually and particularly important in times of rapidly changing and unpredictable economic parameters.

In addition to the basic method presented, this chapter also presents (i) the application of the method to obtain guideposts for heat exchangers, (ii) an evaluation of exergetic efficiencies of heat exchangers including the principal of the inclusion of the exergy of the material, (iii) specific application of the method, and comparison of the results with the corresponding results obtained from the traditional irreversibility method and the thermoeconomic method.

THE BASIC METHOD

From the base established by Boyd et al. (1981) and Le Goff and Giulietti (1982), an irreversibility rate based objective function that includes a term to account for the exergy of the materials of the heat exchanger is proposed here, and Equation (1.3) then becomes:

$$Ns_r = Ns_{\Delta T} + Ns_{\Delta p} + Ns_m \qquad (3.1)$$

where, Ns_m is a non-dimensional measure of the irreversibility associated with the heat exchanger material. The logic of this is that a minimum exergy expenditure equal to the exergy of the material is required to make the heat exchanger from the dead state. This is an exergy expenditure that is effective for the "application life" of the heat exchanger

Thus, the equivalent irreversibility rate assignable to the materials of the heat exchanger is the exergy of the material EX_m divided by the application life[6] of the heat exchanger:

$$\dot{I}_m = \frac{EX_m}{t_{ap}} \tag{3.2}$$

and in non-dimensionalized form:

$$Ns_m = \frac{\dot{I}_m}{T_0 \; C_{max}} \tag{3.3}$$

For simple heat exchangers, the exergy of heat exchanger materials EX_m is approximately equal to the chemical exergy of the material. Materials acquire chemical

---

[6] The application life is a crucial parameter in the overall analysis presented here. Some people may, in fact, argue that because of this term, the analysis here no longer is just thermodynamic in nature. The author respects this position, but also feel that as soon as it is indicated that an optimum engineering design is desired, we must pass the fence from pure science to the design process. The author feels that the use of an application life is realistic for analyzing designs. If the designer wants to be as close to pure science as possible, he/she can take the application life as the actual physical life of the heat exchanger operating with the given media (i.e., the time before the unit would be physically destroyed by corrosion, etc.). If, on the other hand, the designer wants the optima to reflect the economic application life, he/she should use a time period that reflects the minimum acceptable payback period as the application life.

exergies when they are transformed from the dead state into their final form. The chemical exergy is found in the usual manner by calculating the maximum work that can be obtained when the material under consideration is brought to chemical equilibrium with the components of the dead state.

Figure 3.1 is a modification of Fig. 1.2 which includes the material irreversibility contribution. The material irreversibility increases as the area is increased, yielding optimum designs with a finite area.

Current processes to produce materials have low efficiencies, and the actual exergy expenditure for a given heat exchanger is much greater than just the exergy of the material. The objective function for irreversibility minimization analysis can be formulated to take into account all the exergy expenses associated with the fabrication of the material by dividing the material irreversibility rate by an adequate efficiency for the overall construction process. The objective function expressing the global irreversibility generation at the current state of technology is:

$$Ns_i = Ns_{\Delta T} + Ns_{\Delta p} + Ns_{mi} \tag{3.4}$$

where,

$$Ns_{mi} = \frac{Ns_m}{\psi_m} \tag{3.5}$$

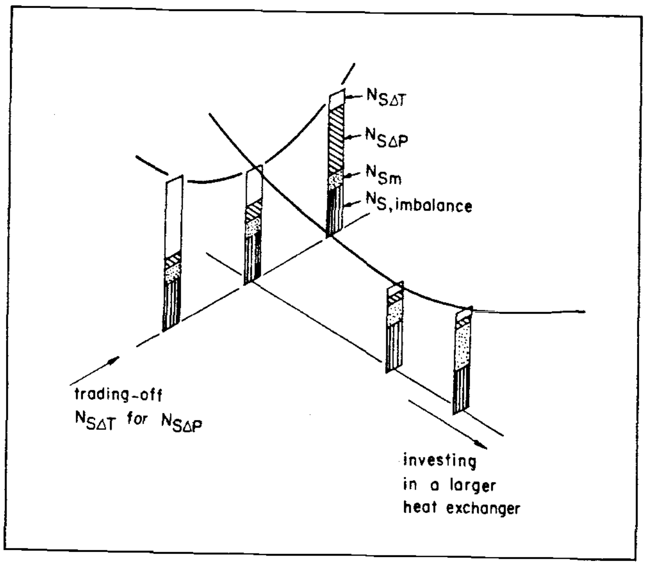

Figure 3.1. The structure of the total irreversibility generation in a heat exchanger including the material irreversibility.

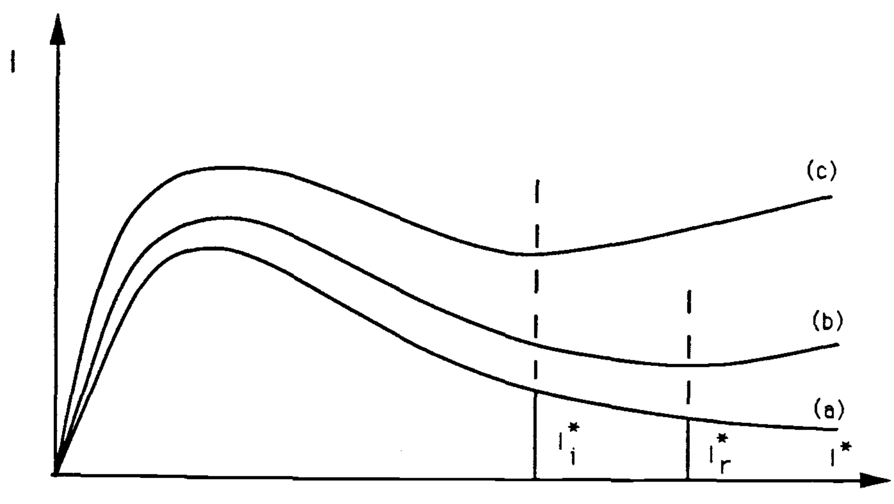

Figure 3.2. The guideposts for the minimum irreversibility and maximum NTU obtained from the three irreversibility rate based objective functions, Equation (1.3), Equation (3.1), and Equation (3.4). The plots for the three equations are indicated as (a), (b), and (c), respectively.

and ψ_m is the exergetic efficiency for the overall manufacturing process.

The objective function presented in Equation (3.1) corresponds to the case when $\psi_m = 1$, or, for the case of an ideal manufacturing process of the heat exchanger. Hence, the use of the optima obtained from Equation (3.1) is restricted; their primary value lies in their representing non-changing guideposts. The guideposts obtained from Equation (3.4) is a function of the efficiency of the manufacturing process. However, ψ_m is not expected to change significantly during the useful life of a heat exchanger. Hence, Equation (3.5) is the preferred irreversibility rate based objective function for heat exchanger design.

IMPROVED GUIDEPOSTS

The variation of the total irreversibility with NTU for the three irreversibility rate based objective functions are shown in Fig. 3.2. Plots (a), (b) and (c) correspond to the objective functions given by Equations (1.3), (3.1) and (3.4) respectively. The traditional irreversibility method gives the remanent irreversibility (irreversibility corresponding to the infinite NTU limit, and indicated as I^* on Fig. 3.2), as the guidepost for the potential reduction in the total irreversibility in a finite area heat exchanger. The maximum NTU is finite when Equation (3.1) is used as the objective function, and the corresponding irreversibility on (a), indicated as I_r^* in Fig. 3.2, is an improved guidepost for the potential reduction in the total irreversibility in a finite area heat exchanger. With Equation (3.4) as the objective function, the value of the optimum NTU further decreases, and the corresponding irreversibility on (a), indicated as I_i^* in Fig. 3.2, becomes a guidepost for the potential irreversibility reduction in a finite area heat exchanger. This is a more realistic guidepost for the potential irreversibility reduction and the maximum NTU of an "economically optimum heat exchanger", compared to the previous guideposts. The irreversibility of an "economically optimum heat exchanger" would always be higher than the above guidepost, and the NTU would be always lower. The guidepost obtained from the modified irreversibility method (Equation (3.4)) would approach the corresponding values in an "economically optimum heat exchanger" when the exergy cost is high relative to labor and profits.

EXERGETIC EFFICIENCY

The exergetic efficiency of a plant component is defined in a variety of ways in the literature. The following expression is selected for this work (Tsatsaronis and Winhold, 1985):

$$y_{hx} = \frac{\dot{EX}_p}{\dot{EX}_f} \tag{3.6}$$

where $\dot{EX}_p$ is the rate of exergy gained by the product, and $\dot{EX}_f$ is the rate of exergy loss by the fuel. When the component is a heat exchanger designed for heating, the exergy gained by the cold stream is equal to $\dot{EX}_p$, and the exergy loss by the hot stream plus the rate of exergy loss due to the heat exchanger material is equal to $\dot{EX}_{f,m}$. When the irreversibility contribution from the material is not included in the analysis, the rate of exergy loss by the fuel is the rate of exergy loss by the hot stream and is designated as $\dot{EX}_f$. The exergetic efficiencies for a heat exchanger with and without the irreversibility contributions from the material are developed next.

The exergy gained by the product is,

$$\dot{EX}_p = \dot{EX}_{c''} - \dot{EX}_{c'} \tag{3.7}$$

When the material irreversibility rate is not included, the exergy loss by the fuel is given by,

$$\dot{EX}_f = \dot{EX}_{h'} - \dot{EX}_{h''} \tag{3.8}$$

When the material irreversibility rate is included, the exergy loss by the fuel is given by,

$$\dot{EX}_{f,m} = \dot{EX}_f + \dot{I}_A A \tag{3.9}$$

For an ideal gas, when the pressure drop irreversibility and heat losses to the surroundings are negligible, the above equations can be written as,

$$\dot{EX}_p = T_0 C_{min} \{(T_c/T_0) \epsilon (\tau^{-1} - 1) - \ln [1 + \epsilon (\tau^{-1} - 1)]\} \quad (3.10)$$

$$\dot{EX}_f = T_0 C_{max} \{(T_c/T_0) \omega\epsilon (\tau^{-1} - 1) + \ln [1 - \omega\epsilon (1 - \tau)]\} \quad (3.11)$$

$$\dot{EX}_{f,m} = \dot{EX}_f + \gamma T_0 C_{min} NTU \quad (3.12)$$

where T_c is the inlet temperature of the cold stream, ϵ is the effectiveness of the heat exchanger and τ is the inlet temperature ratio. The value of γ is replaced by γ_i to take into account all the irreversibilities associated with the manufacturing process, at the current state of technology. The exergetic efficiencies with and without the material irreversibility rate, ψ_{hx} and $\psi_{hx,m}$ respectively, can now be written as,

$$\psi_{hx} = \frac{\dot{EX}_p}{\dot{EX}_f} \quad (3.13)$$

$$\psi_{hx,m} = \frac{\dot{EX}_p}{\dot{EX}_{f,m}} \quad (3.14)$$

The equations developed above for the exergetic efficiency are now applied to both counterflow and parallel flow heat exchangers.

The variation of the exergetic efficiencies ψ_{hx} and $\psi_{hx,m}$ effectiveness for counterflow and parallel flow heat exchangers with the same γ (γ = .01), are shown in Fig. 3.3. This figure is for a balanced capacity rate with the cold stream inlet temperature equal to the dead state temperature. The exergetic efficiency $\psi_{hx,m}$ is always less than ψ_{hx}. At the infinite area heat exchanger limit (ϵ = 0.5 for the parallel flow heat exchanger, ϵ = 1.0 for the counterflow heat exchanger) $\psi_{hx,m}$ tends to zero while ψ_{hx} is a maximum. The exergetic efficiency $\psi_{hx,m}$ for the parallel flow heat exchanger at a given ϵ is always lower than $\psi_{hx,m}$ for a counterflow heat exchanger, and this difference can be clearly seen at values of the effectiveness close to 0.5. The exergetic efficiencies ψ_{hx} for the parallel and counterflow heat exchangers are the same. However, for a given duty (a given effectiveness in Fig. 3.3) a counterflow heat exchanger is known to be more attractive than a parallel flow unit, because of the reduced area; this is properly shown in the exergetic efficiency if the material irreversibility is included as illustrated in Fig. 3.3. The irreversibility rate in building and operating an infinite area heat exchanger makes the exergetic efficiency in the large heat exchanger area limit tend to zero, when the application life is finite. Hence, including the material irreversibility results in physically realistic values for

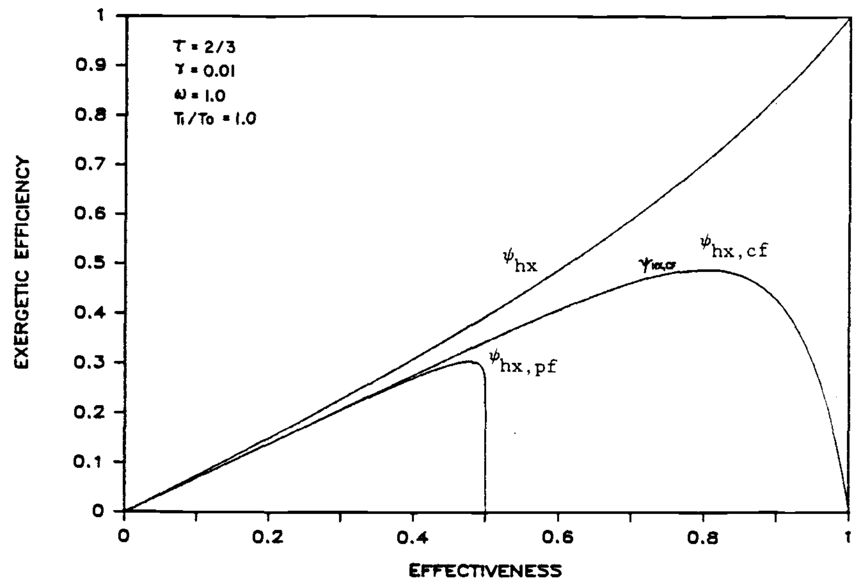

Figure 3.3. Exergetic efficiencies for counterflow and parallel flow heat exchangers without (ψ_{hx}) and with ($\psi_{hx,m}$) the effect of the material irreversibility, for a balanced capacity rate ratio ω of 1, an inlet temperature ratio τ of 2/3, and a material exergy parameter γ equal to 0.01.

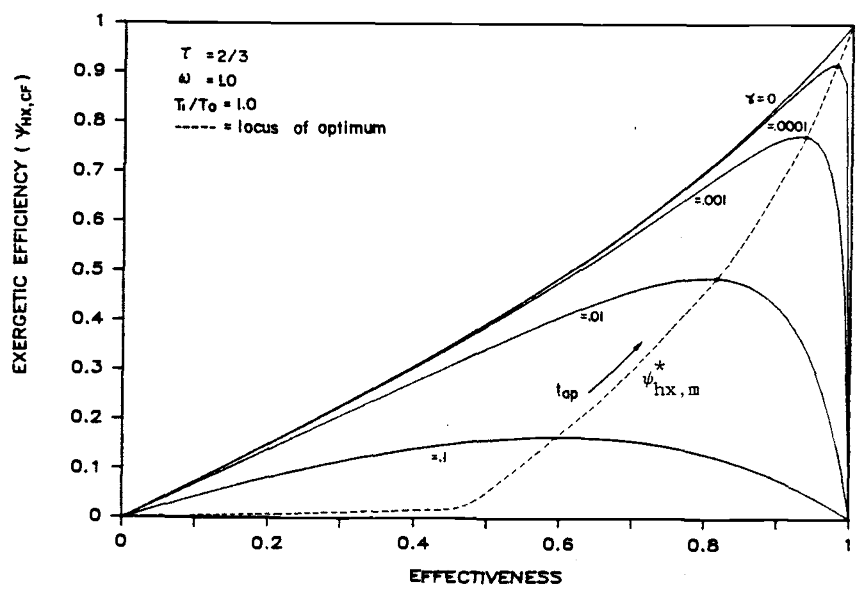

Figure 3.4. Exergetic efficiency of a counterflow heat exchanger as a function of heat exchanger effectiveness for different values of the material exergy parameter γ.

the exergetic efficiency.

The parameter γ is inversely proportional to the application life of the heat exchanger, t_{ap}. When the application life approaches zero, γ tends to infinity and the exergetic efficiency ψ_{hx} is equal to zero in the limit. However, if the heat exchanger has an infinite application life, then γ is equal to zero, and the exergetic efficiency $\psi_{hx,m}$ is equal to ψ_{hx}. The variation of the exergetic efficiency $\psi_{hx,m}$ with the effectiveness for a balanced counterflow heat exchanger is shown in Fig. 3.4, for different values of γ. This figure is for a heat exchanger inlet temperature ratio of 2/3 and the cold stream inlet temperature equal to the dead state temperature. Figure 3.4 shows that $\psi_{hx,m}$ increases with decreasing γ (increasing application life, t_{ap}), for a given value of ϵ, and has a maximum for any value of γ. The dotted line on Fig. 3.4 shows the locus of the maximum exergetic efficiency $\psi^{*}_{hx,m}$ with the arrow indicating the direction of increasing application life. When the heat exchanger is reversible ($\psi_{hx,m} = 1$), then $\gamma = 0$ and $\epsilon = 1$. This shows that an infinite area balanced counterflow heat exchanger with an infinite application life is reversible. Hence, including the material irreversibility term results in adding a time constraint in the reversible heat exchanger limit.

APPLICATION

This section presents the irreversibility rate based objective functions and the thermoeconomic objective function, for a case when $Ns_{\Delta p}$ is negligible and ideal gas behavior can be assumed for the heat exchanger streams. The analysis also neglects heat losses to the surroundings. The method developed is then applied to counterflow heat exchangers, and the guideposts obtained from the irreversibility methods are compared to the corresponding values of the thermoeconomically optimum designs.

Objective Functions

The exergy concept based objective functions for the case considered in this section, are developed next.

(a) Traditional Irreversibility Method:

When the pressure drop irreversibility is negligible, Equation (1.3) becomes:

$$Ns = Ns_{\Delta T} \tag{3.15}$$

When ideal gas behavior is assumed for the flowing fluid, and there are no heat losses to the surroundings, the heat transfer irreversibility generation number is:

$$Ns_{\Delta T} = \omega \ln [1 + (\epsilon/\tau)(1-\tau)] + \ln [1 + \omega\epsilon\ (\tau-1)] \tag{3.16}$$

The guideposts obtained from the traditional irreversibility method for the maximum NTU and minimum irreversibility are denoted here as NTU^* and Ns^*, respectively.

(b) Modified Irreversibility Method (with $\psi_m = 1$):

For a case with negligible pressure drop irreversibility, Equation (3.1) becomes:

$$Ns_r = Ns_{\Delta T} + Ns_m \tag{3.17}$$

As previously discussed, the heat exchanger material has an exergy value at the moment of installation. The exergy value of the heat exchanger can be viewed as being used up after a time equal to the application life t_{ap}. The decrease in the exergy of the material can be considered as an irreversibility generation rate given by:

$$Ns_m = \frac{\dot{I}_A A}{T_0 C_{max}} = \gamma \omega NTU \tag{3.18}$$

where $\dot{I}_A = \dot{I}_m/A$ is the irreversibility rate per unit area of the heat exchanger, ω is the capacity rate ratio C_{min}/C_{max}, and γ is the material exergy parameter defined as:

$$\gamma = \frac{\dot{I}_A A}{T_0 U} \tag{3.19}$$

Substitution of (3.16) and (3.18) into (3.17) yields the final form of the objective function:

$$Ns_r = \omega \ln [1 + (\epsilon/\tau)(1-\tau)] + \ln [1 + \omega\epsilon\ (\tau-1)] + \gamma\ \omega\ NTU \quad (3.20)$$

The variation of the heat transfer irreversibility (first term), the material irreversibility (second term), and the total irreversibility as a function of NTU is shown in Fig. 3.5.

The guideposts obtained from Equation (3.20), for the maximum NTU and minimum irreversibility are denoted here as NTU_r^* and Ns_r^*, respectively.

(c) Modified Irreversibility Method (with ψ_m fixed by the current state of the technology):

The objective function for this case is similar to Equation (3.20) with γ replaced by γ_i, resulting in the following form,

$$Ns_i = \omega \ln [1 + (\epsilon/\tau)(1-\tau)] + \ln [1 + \omega\epsilon\ (\tau-1)] + \gamma_i \omega\ NTU \quad (3.21)$$

with the non-dimensional parameter γ_i defined as:

$$\gamma_i = \frac{\gamma}{\psi_m} \quad (3.22)$$

The guideposts obtained from the traditional irreversibility method for the maximum NTU and minimum irreversibility are denoted here as NTU_i^* and Ns_i^*, respectively.

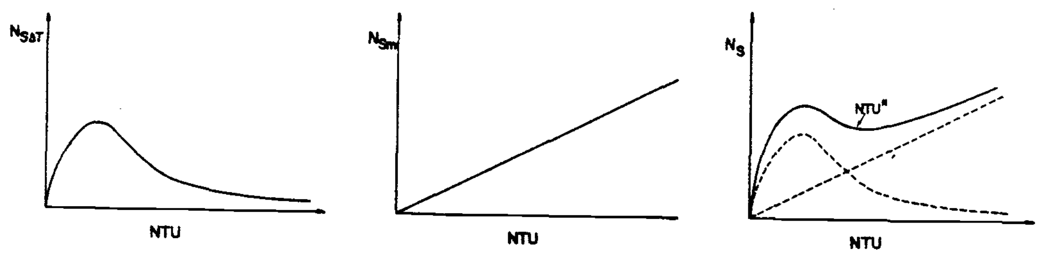

Figure 3.5. The non-dimensional irreversibility generation number as a function of NTU, showing: (a) contribution from $Ns_{\Delta T}$, (b) contribution from Ns_m, and (c) total ($Ns_{\Delta T}$ + Ns_m).

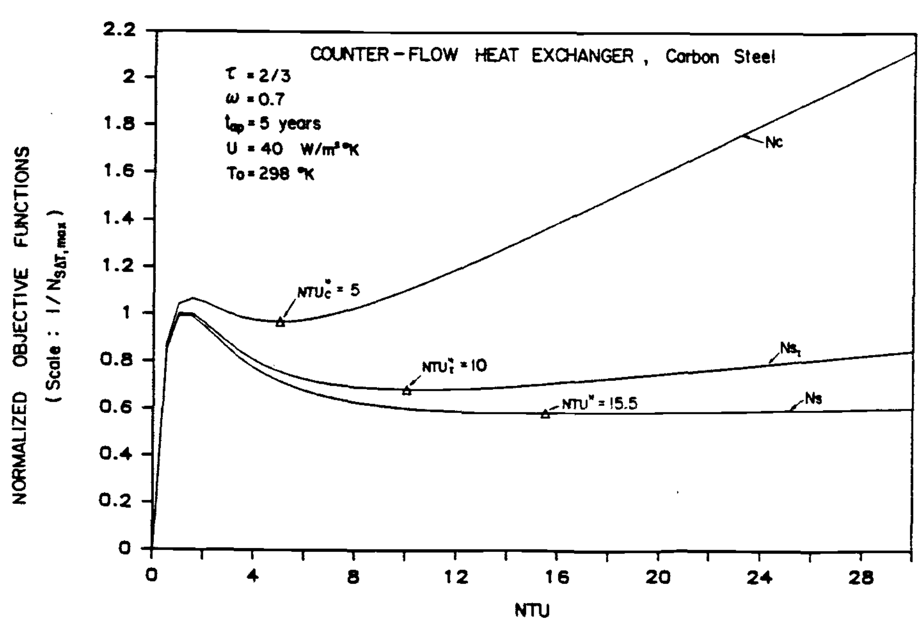

Figure 3.6. Values of NTU for the three objective functions Ns_r, Ns_i and Ns_c, for the example.

(d) Thermoeconomic Method:

Assuming that the capital costs are proportional to the area, and pressure drop irreversibility is negligible, the thermoeconomics based objective function (Equation (1.4))becomes:

$$\dot{C}_t = \dot{C}_A \, A + c_{\Delta T} \, \dot{I}_{\Delta T} \tag{3.23}$$

where C_A is the capital cost per unit area of the heat exchanger and $c_{\Delta T}$ is the cost associated with heat transfer irreversibility.

The objective function can be non-dimensionalized as follows:

$$Nc = \frac{C_t}{c_{\Delta T} \, T_0 \, C_{max} \, t_{ap}} \tag{3.24}$$

Under the assumptions of ideal gas behavior, the objective function becomes:

$$Nc = \omega \ln \left[1 + (\epsilon/\tau)(1-\tau)\right] + \ln \left[1 + \omega\epsilon(\tau-1)\right] + \gamma_c \, \omega \, NTU \tag{3.25}$$

where the non-dimensional parameter γ_c is defined as:

$$\gamma_c = \frac{C_A}{c_{\Delta T} \, T_0 \, U \, t_{ap}} \tag{3.26}$$

Note that the thermoeconomic objective function given by Equation (3.25) has the same functional form as the two previously derived irreversibility rate based objective functions given by Equation (3.20) and (3.21). The only difference is that the value of γ_c has to be used instead of γ or γ_i. The optimum NTU obtained from Equation (3.25) is denoted here as NTU_c^*. The irreversibility at this optimum is denoted by Ns_c^*.

<u>Application to Counterflow Heat Exchangers</u>

The objective functions developed previously are used next for the design optimization of a counterflow heat exchanger.

(a) Example:

Determine NTU^*, NTU_i^* and NTU_c^* for a counterflow heat exchanger for values of the application life of 5 and 10 years, and for U = 40 and 70 W/m^2-K. Also determine Ns^*, Ns_r^*, Ns_i^*, and compare with Ns_c^*. The capacity rate ratio and the inlet temperature ratio are $\omega = 0.7$ and $\tau = 2/3$, respectively. The shell and tubes of the heat exchanger are made of carbon steel, and some data for the steel tubes are given in Table 3.1. The cost of heat exchanger area (C_A) is approximately 72 $\$/m^2$ (from Matley, 1983), an estimate for the cost of heat transfer irreversibility is 0.056 \$/kW-hr and the gross energy requirement to produce steel tubes is 35.7×10^3 J/kg (from Chapman and Roberts, 1983). The temperatures T_0 = 298 K and T_1 = 298 K.

Table 3.1. Data for carbon steel tubes.

Density	**7770 kg/m^3**
Thickness	**3.2 mm**
Chemical Exergy	**6764 kJ/kg**

(b) Solution:

The chemical exergy of the heat exchanger material is:

$$EX_{m,ch} = (6764 \times 10^3)\ (7770)\ (3.2 \times 10^{-3})\ A \quad (J)$$
$$= 1.6818 \times 10^8\ A \quad (J)$$

The material exergy, EX_m, is approximately equal to the chemical exergy $EX_{m,ch}$.

The effectiveness of the manufacturing process for carbon steel tubes can be calculated as,

$$\psi_m = \frac{6764}{35.7 \times 10^3} = 0.19$$

The cost of heat transfer irreversibility is:

$$c_{\Delta T} = \frac{0.056}{3600 \times 1000} = 1.556 \times 10^{-8} \quad (\$/J)$$

The values of γ, γ_i and γ_c can now be calculated by using Equations (3.19), (3.22) and (3.26) respectively as,

$$\gamma = \frac{1.6818 \times 10^8}{2.98\ U\ t_{ap}} = \frac{564 \times 10^3}{U\ t_{ap}}$$

$$\gamma_i = \frac{\gamma}{0.19} = \frac{2970 \times 10^3}{U\ t_{ap}}$$

$$\gamma_c = \frac{72}{(1.556 \times 10^{-8})\ (298)\ U\ t_{ap}} = \frac{15527 \times 10^3}{U\ t_{ap}}$$

Finally, the NTU^* values can be obtained by solving the single degree of freedom optimization problems. (The variation of the two modified irreversibility rate objective functions and the thermoeconomic objective function with NTU, when the overall heat transfer coefficient is 40 W/m^2 K and the application life is 5 years, is shown in Fig. 3.6.) The results are summarized in Table 3.2.

Table 3.2. Values of NTU^*

U (W/m^2 K)	t_{ap} (years)					
	5			10		
	NTU_r^*	NTU_i^*	NTU_c^*	NTU_r^*	NTU_i^*	NTU_c^*
40	15.5	10	5	18	12	7
70	17.5	12	6.5	19.5	14	8.5
$NTU^* = \infty$, for all cases.						

The guideposts for the minimum irreversibility are obtained by substituting the optimum NTU values in Table 3.2 in Equation (3.15). The irreversibility of the thermoeconomically optimum heat exchanger is also obtained similarly. The guideposts for the minimum irreversibility and the irreversibility of the thermoeconomically optimum designs, for the examples, are presented in Table 3.3.

Table 3.3 Guideposts for minimum irreversibility

U (W/m^2 K)	t_{ap} (years) 5			10		
	Ns_r^*	Ns_i^*	Ns_c^*	Ns_r^*	Ns_i^*	Ns_c^*
40	0.01833	0.01920	0.02322	0.01822	0.01871	0.02084
70	0.01823	0.01871	0.02130	0.01818	0.01844	0.01983
Ns^* = 0.01812, for all cases.						

DISCUSSION

The method presented in this chapter, which adds an irreversibility term due to the material of construction of the heat exchanger in the overall irreversibility minimization equation for heat exchanger optimization, allows physically realistic optimization to be conducted. The resulting optimum designs provide conceptually beneficial guideposts which do not change with time or location. Such optima are in contrast to the optima obtained by presently advocated methods which on one hand indicate unrealistic infinite area heat exchangers and on the other hand point to optima which may change dramatically with location and time.

The optima obtained by the method indicated here are conceptually similar to the economic optimization, and, in fact both objective functions have the same functional form, in some instances. The use of the optima obtained from the basic method delineated here is restricted; their primary value lies in their representing non-changing guideposts. With modifications to represent the current state of technology, the method delineates optima that are substantially closer to those specified by current thermoeconomic optimization.

Exergetic efficiency expressions that similarly include an irreversibility term due to the material of construction of the heat exchanger show physically more realistic values than the usual expressions that don't include such a term. Such exergetic efficiencies clearly show the thermodynamic advantage of counterflow as compared to parallel flow arrangements, and they show a value of zero for all infinite area heat exchangers except those with infinite application lives. This is in contrast to some other usual exergetic efficiency expressions that show counterflow and parallel flow units having equal efficiencies and which can yield values of 100% for infinite area heat exchangers.

Although the analysis presented here does not represent the "cure-all" for heat exchanger analysis (economic optimization is still recommended), it does provide conceptually valuable analysis methods and non-changing guideposts for optimal design.

CHAPTER 4

DESIGN OPTIMIZATION OF A HEAT EXCHANGER IN A COMPLEX SYSTEM

The present chapter illustrates the application of the modified irreversibility rate based objective function and the thermoeconomic objective function to optimize an emerging technology ceramic heat exchanger which is a part of a complex power plant. It is included here, to extend the work of the previous chapters through to a completed detailed design.

BACKGROUND

The ceramic heat exchanger considered in this chapter is an air-to-gas unit with multiple passes in the gas side. A schematic of the heat exchanger is presented in Fig. 4.1. The heat exchanger resides in a combined-cycle, wood-fueled power plant. A diagram indicating the position of the heat exchanger in the power plant is shown in Fig. 4.2. The heat exchanger is used to pre-heat the compressed air for the indirectly fired gas turbine. The exergy of the flue gas exiting the heat exchanger and the exergy of the turbine exhaust are partially recovered by a Rankine steam cycle. A detailed description of the base[7] case heat exchanger, as

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[7] The ceramic heat exchanger initially recommended for the subject power plant will be here referred to as the base case design.

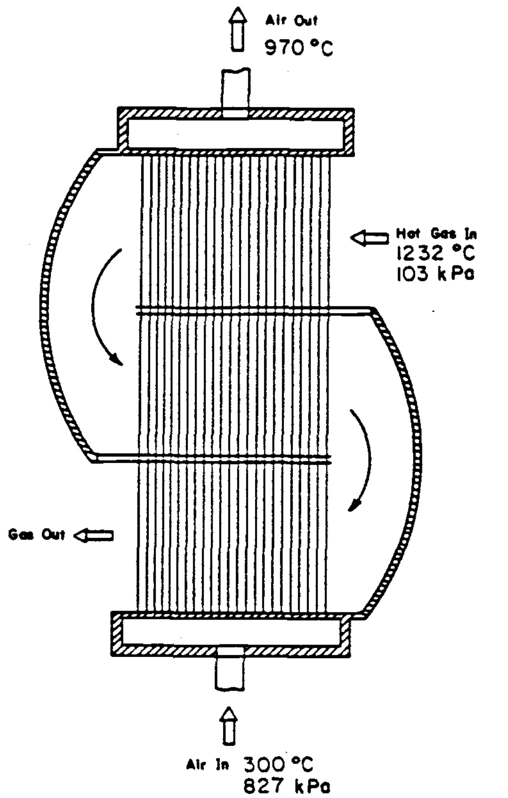

Figure 4.1. A diagram of the ceramic heat exchanger.

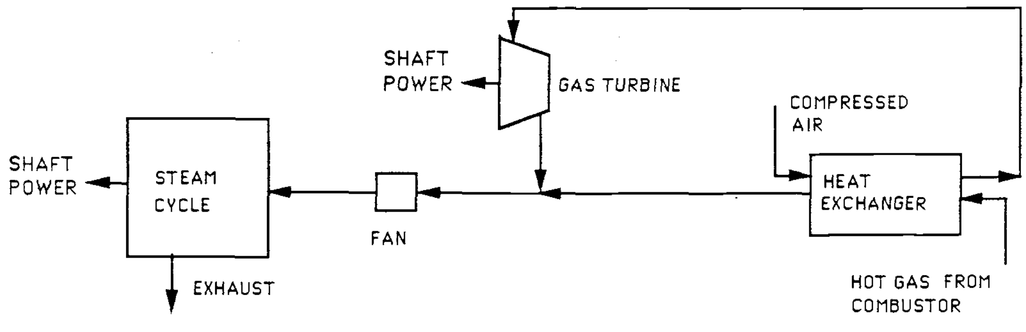

Figure 4.2. A schematic indicating the position of the ceramic heat exchanger in the complex thermal system.

well as the power plant in which it resides is given by Ranasinghe and Reistad (1987).

FORMULATION OF THE PROBLEM

The objective functions, design variables, design constraints, and constant parameters used for the present optimization are presented next.

Objective Functions

A previous analysis of the power plant indicated that (as expected) the maximum plant efficiency is obtained when the air outlet temperature from the heat exchanger (and inlet to the gas turbine) reaches its maximum. Reducing this temperature below its maximum does not result in any improvement, and thus the heat exchanger is optimized with this temperature fixed at its maximum allowable value. Consequently, the analysis here is limited to a fixed-duty heat exchanger (the inlet temperature and the capacity rates of both streams are also fixed), with the irreversibility due to heat transfer a constant. The thermoeconomic objective function and the modified irreversibility rate based objective function for the fixed-duty heat exchanger are presented in Equations (4.1) and (4.2) respectively as,

$$\dot{C}_t = \dot{C}_A + c_{\Delta pa} \dot{I}_{\Delta pa} + c_{\Delta pg} \dot{I}_{\Delta pg} \tag{4.1}$$

$$\dot{I}_{t,mi} = \dot{I}_{\Delta pa} + \dot{I}_{\Delta pg} + \dot{I}_{mi} \tag{4.2}$$

Where $c_{\Delta pa}$ and $c_{\Delta pg}$ are the cost coefficients associated with the air-side and gas-side pressure drop irreversibilities, $\dot{I}_{\Delta pa}$ and $\dot{I}_{\Delta pg}$, respectively.

The cost of a heat exchanger as a function of its heat transfer surface area was assumed to be represented by the relationship:

$$C_A = k\ A^{0.6} \tag{4.3}$$

where k is a constant and A is the heat exchanger area. The constant k is evaluated from the base-case cost estimate for the biomass power plant ($3,000,000 for a heat exchanger with surface area of 740 m^2).

Decision Variables

The independent decision variables used here are the length and diameter of the ceramic tubes, and the number of pipes normal and parallel to the flow direction. The tube wall thickness is assumed to be equal to 6.35 mm, which is the minimum value due to strength considerations for high pressure and temperature applications.

Constraints

The inequality constraints and bounds used for the present analysis are as follows:

* The maximum flue gas velocity is constrained at 17.5 m/s (to prevent tube erosion).
* Maximum length and diameter of the ceramic tubes are

constrained to 2.7m and 65 mm respectively, due to manufacturing limitations.

* The minimum tube gap is constrained to 25 mm due to maintenance requirements. Optimum designs for minimum tube gap constraint greater than 25 mm are also presented.

Constants

Parameters which were kept constant during the optimization are presented in Table 4.1.

SOLUTION METHOD

The optimization procedure for the heat exchanger starts by using the simulation model for the biomass power plant to obtain the constant thermodynamic property data, flow rates and composition of the working fluids (the source code for the power plant is given in Dadkhah-Nikoo et al., 1987). The ceramic heat exchanger subroutine is used as a separate module for the heat exchanger calculations. The source code of the heat exchanger subroutine is presented in Appendix B (the subroutines used in the heat exchanger model are given in Ranasinghe, 1986). Initially a set of values for the decision variables is chosen. The heat exchanger subroutine calculates the thermal and flow properties of the fluids, including irreversibility generation rates. Next, the output from the heat exchanger calculations, including the value of the objective function and constraints, is fed

Table 4.1. Constant parameters.

Parameter	Air-Side	Gas-Side
Temperature in (C)	300.0	1232.0
Temperature out (C)	971.0	--
Mass flow rate (kg/s)	21.5	22.5
Pressure at inlet (kPa)	827.0	103.0
Dead State Condition: 298 K, 1 atm, and the chemical composition as specified by Kotas, 1985.		
Chemical Exergy of Ceramics: 27538.7 kJ/kg (Kotas, 1985)		
Exergetic Efficiency of Manufacturing Process: 0.1[1]		
Application Life: 2 years.[2]		
Cost of Electricity: $.09/kW-hr, Huque and Reistad, 1987		

1 The effectiveness of the manufacturing process for carbon steel tubes is approximately 19% (Chapman and Roberts, 1983). The effectiveness of the manufacturing process for the ceramic tubes was not available; an effectiveness was selected at a nominal value about half that for the carbon steel tubes.

2 The first few rows of the ceramic heat exchanger have to be replaced as soon as after 1 year, while some tubes can be used as long as 4-5 years. Based on these figures, an application life of 2 years is assumed, to simplify the present analysis. A penalty for a larger frontal area, due to the reduction in application life, could be introduced in a more complete analysis.

to an optimization computer program package. New values for the design variables are subsequently generated in the optimizer, and heat exchanger calculations are repeated. The iterations are continued until the objective function is optimized within the accuracy specified and all the constraints are satisfied.

For the thermoeconomic optimization, the cost coefficients are calculated by using the power plant simulation model. A small additional pressure drop is introduced in the side of the heat exchanger for which the cost coefficient is required. This pressure drop causes an increase in irreversibility generation and a loss of output power. The losses in output power per unit of pressure drop irreversibility in the air and gas sides are denoted here as $\Delta W_{\Delta pa}$ and $\Delta W_{\Delta pg}$, respectively. The value of the output power is the cost of electricity produced from the power plant, c_e.[8] The cost coefficients are calculated by multiplying the power loss per unit of irreversibility by the cost of electricity.[9]

The initial cost coefficients are calculated using the base case ceramic heat exchanger configuration. Once the optimum design is calculated, new cost coefficients are calculated by using the simulation program for the biomass power plant. This procedure is continued until the designs

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[8] The cost of producing electricity from the power plant varies as the heat exchanger design is changed. However, these changes are small and therefore the electricity cost is taken as a constant.

[9] Cost coefficients calculated by using this procedure are not average costs but marginal costs at the design conditions, and they are expected to vary from one design to another. However, the results obtained from the computer simulation indicate that these costs have a variation of only about 1% over a wide range of design conditions (Table 4.4). This results in the marginal costs being approximately equal to the average cost, over the domain of decision variables.

converge to the specified accuracy.

RESULTS

Results have been developed to indicate the specific optimal designs as well as to illustrate a comparison of the optimization with the two objective functions, $\dot{I}_{t,mi}$ and $\dot{C}_t$. The comparison, presented first, takes a restricted -- single-degree-of-freedom -- case of the subject heat exchanger based on the base case design parameters previously indicated. The overall optimization presents the optimal design for the decision variables delineated above.

Comparison of optimization for minimum $\dot{I}_{t,mi}$ and $\dot{C}_t$

Comparison of the optimal designs obtained with the objective functions of minimum $\dot{C}_t$ and minimum $\dot{I}_{t,mi}$ are presented for a single-degree-of-freedom optimization done around the base case design (i.e. all the design variables, other than the decision variable chosen for the analysis, are fixed at the base design values). The independent decision variable chosen for this comparison is the number of pipes normal to the flue gas flow direction (npn).

The optimum for the objective function $\dot{I}_{t,mi}$ was evaluated and found to occur at an npn of 56. As discussed previously, this optimum value will not change unless the efficiency of the manufacturing process changes.

The optimum for the $\dot{C}_t$ objective function is expected to change with time and location and thus should be evaluated over a range of costs. The variations of costs were found to be conveniently expressed in terms of a single parameter, β, which comes out of the objective function equations as follows:

$$\dot{C}_t = (k\ A^{0.6})/t_{ap} + (c_e\ \Delta W_{\Delta pa})\ \dot{I}_{\Delta pa} + (c_e\ \Delta W_{\Delta pg})\ \dot{I}_{\Delta pg} \tag{4.4}$$

or, rewritten

$$\dot{C}_t = c_e\{(\beta\ A^{0.6})/t_{ap} + \Delta W_{\Delta pa}\ \dot{I}_{\Delta pa} + \Delta W_{\Delta pg}\ \dot{I}_{\Delta pg}\} \tag{4.5}$$

where β is defined as k/c_e.

In the above equation c_e is the cost of electricity in \$/kJ and $\Delta W_{\Delta pa}$ and $\Delta W_{\Delta pg}$ are the lost power output from the power plant per unit of irreversibility on the air and gas sides respectively. The values of $\Delta W_{\Delta pa}$ and $\Delta W_{\Delta pg}$ for the base case design were calculated as described earlier,and are equal to 1.463 kJ/kJ and 4.762 kJ/kJ respectively.[10] The parameter β is a function of the capital cost of the heat exchanger and the cost of electricity produced from the power plant.

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[10] The results obtained for $\Delta W_{\Delta pa}$ and $\Delta W_{\Delta pg}$ indicate that the gas side pressure drop irreversibility is roughly 3 times as valuable as the air side pressure drop irreversibility. The reason is that gas side pressure drop has to be overcome by feeding high quality energy to a fan, while the pressure drop in the air side only reduces the exergy of the air stream which has to be converted to work in the turbine (see Fig. 4.2).

If the minimum possible cost of a heat exchanger of area 740 m^2 is assumed to be \$1,000,000, and the maximum possible cost is \$5,000,000, for all locations and times, the maximum and minimum values of k are 22757 and 4551, respectively. The parameter β has its maximum value when k is a maximum and c_e is a minimum, and β is minimum when k is minimum and c_e is maximum. Hence, the maximum and minimum values of β are equal to 1.365×10^9 and 1.638×10^8, respectively. The values of β between the maximum and minimum correspond to the various combinations of the heat exchanger cost and the cost of electricity.

Figure 4.3 shows the optimum npn values for a single degree of freedom optimization of the ceramic heat exchanger with the $\dot{C}_t$ objective function. The optimum designs are presented for the total range of values of β, between its maximum and minimum. Hence, these optima represent the optimum designs for any combination of heat exchanger cost and cost of electricity. The optimum obtained from the objective function $\dot{I}_{t,mi}$ (npn = 56), is also indicated in this figure.

The results from the above parametric study, with a single degree of freedom, illustrate that the optimum design obtained from the thermoeconomic objective function $\dot{C}_t$ is a strong function of the cost parameters. The results also show that the objective function $\dot{I}_{t,mi}$, which is not as sensitive to time and location as $\dot{C}_t$, yields optimum designs

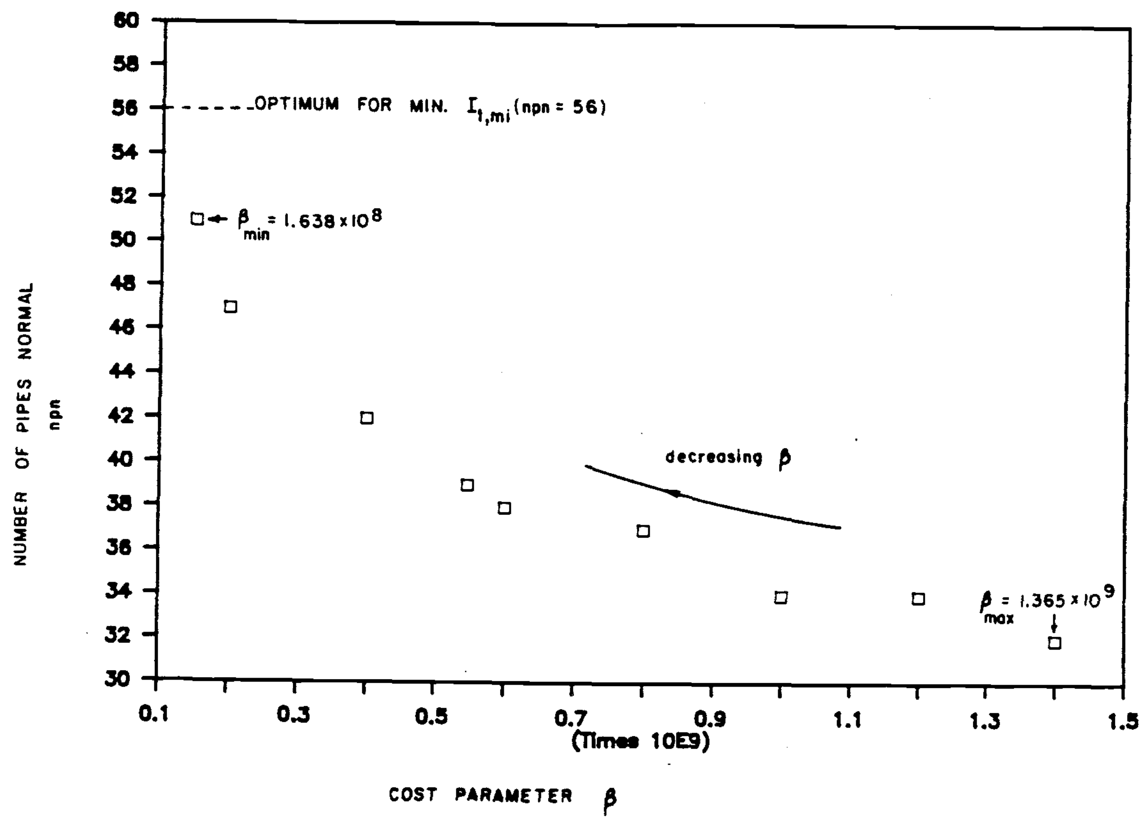

Figure 4.3. Variation of the optimum number of pipes normal to the flow direction, based on the objective function C_t, as a function of the cost parameter β, for a single degree of freedom optimization around the base-case design. The optimum npn for minimum $I_{t,mi}$ is also indicated on the figure.

in the limit of exergy cost being dominant relative to labor and profits. These may readily serve as guideposts for design.

Optimum Ceramic Heat Exchanger Design

The independent decision variables for the optimization as delineated above are npn,npp,l and ID. The optimum designs obtained from the modified irreversibility rate criterion are initially presented, followed by the optimum designs for the minimum thermoeconomic cost criterion. Finally, the optimum designs obtained from the two criteria are compared for cost, exergy savings, and pressure drops relative to each other, and the base case design.

a) Optimum designs for minimum $\dot{I}_{t,mi}$:

The results from the optimization based on the criterion of minimum irreversibility, $\dot{I}_{t,mi}$, are presented in Table 4.2 for a range of npn values. The base case design is also shown for comparison. The results are presented for a range of npn values because the npn value at each optimum determines the tube spacings which can be a major design constraint. The influence of npn and tube spacing is discussed below.

The values of length, l, and internal diameter, ID, are constant at the optima for all npn values; the l value is optimum at its maximum while ID was determined to be optimum at an unconstrained value. The optimum npp decreases as npn

Table 4.2. Optimum designs for minimum $\dot{I}_{t,mi}$. (Base case design included for comparison).

npn	npp	l (m)	ID (mm)	Tube Gap (mm)	Δpa (%)[1]	Δpg (%)[1]	$\dot{I}_{t,mi}$ (kW)
Base Case 40	24	2.74	17.8	39.7	2.26	2.28	153.77
30	42	2.74	20.3	87.4	0.71	1.45	137.52
40	30	2.74	20.3	60.3	0.78	1.23	128.88
50	24	2.74	20.3	48.7	0.78	0.99	124.00
60	20	2.74	20.3	41.2	0.78	0.83	120.77
70	16	2.74	20.3	30.9	0.89	0.91	119.00
80	14	2.74	20.3	27.3	0.89	0.85	117.45

Note: $\dot{I}_{\Delta T}$ is constant at 1583 kW.

1Pressure drops expressed as a percentage of inlet pressure. The inlet pressures are 827 kPa and 103 kPa for the air and gas-sides, respectively.

increases. The objective function $\dot{I}_{t,mi}$ decreases with the increase in npn, although the rate of decrease in the objective function reduces at higher npn. However, the increase in npn corresponds to a decrease in the tube gap (as presented in Table 4.2), and hence becomes a constraint to the maximum possible improvement. The npn value of 80 corresponds to a tube gap of approximately 25 mm. Since a minimum tube gap of 25 mm was assumed for the present analysis, the overall optimum design for minimum $\dot{I}_{t,mi}$ corresponds to a npn value of 80.

If slag formation and cleaning requirements warrant the use of a larger tube gap, an optimum design corresponding to such a tube gap has to be chosen. For example, if a minimum tube gap of approximately 50 mm is needed, the optimum design corresponding to npn of 50 would satisfy this requirement (see Table 4.2).

The results shown in this section are designs that minimize the irreversibility of the heat exchanger taken individually, and these designs do not correspond, in general, with the heat exchanger that minimizes the irreversibility generation in the whole plant. However, the optimum designs presented here were observed to have the effect of increasing the overall plant net efficiency relative to the base case design.

b) Optimum designs for minimum $\dot{C}_t$:

The results from a thermoeconomic optimization of the ceramic heat exchanger are presented in Table 4.3. Optimum designs for different npn values are presented in this table. The minimum tube gap constrains the value of npn as in the previous case when using the objective function $\dot{I}_{t,mi}$. The optimum design corresponding to npn of 80 results in a minimum tube gap of approximately 25 mm, and is the overall optimum for the present thermoeconomic analysis. However, if a larger tube gap is needed, the optimum design has to be determined as described earlier.

Table 4.3. Optimum designs for minimum $\dot{C}_t$. (Base case design included for comparison.

npn	npp	l (m)	ID (mm)	Tube Gap (mm)	Δpa (%)[1]	Δpg (%)[1]	$\dot{C}_t$ ($/s)
Base 40 Case	24	2.74	17.8	39.7	2.26	2.28	0.0551
30	34	2.74	15.2	59.3	4.26	2.50	0.0563
40	25	2.74	15.2	43.3	4.43	1.98	0.0546
50	20	2.74	15.2	35.2	4.43	1.59	0.0537
60	18	2.59	14.0	31.4	5.55	1.44	0.0534
70	17	2.44	14.0	29.7	4.35	1.29	0.0532
80	15	2.44	14.0	26.7	4.28	1.13	0.0529

[1]Pressure drops expressed as a percentage of inlet pressure. The inlet pressures are 827 kPa and 103 kPa for the air and gas-sides, respectively.

Table 4.4. Lost power output per unit of pressure drop irreversibility on the air and gas-sides, at the optimum design conditions.

npn	Base Case	30	40	50	60	70	80
$\Delta W_{\Delta pa}$	1.463	1.465	1.479	1.475	1.484	1.466	1.472
$\Delta W_{\Delta pg}$	4.762	4.761	4.772	4.763	4.778	4.764	4.769

Comparison of $\Delta W_{\Delta pa}$ and $\Delta W_{\Delta pg}$ in Table 4.4 shows that the pressure drop irreversibility on the gas side is approximately 3 times as valuable as that for the air side. It is noted that the variations of the incremental costs over the heat exchanger design points are not very significant (since c_e is constant, incremental cost is proportional to ΔW).

The thermoeconomic optimum design results in a net saving of $139,000, relative to the base case design, over the useful life of the heat exchanger.

c) Comparison of the optimum designs:

The irreversibility and cost of the optimum designs presented in Tables 4.2 and 4.3 are shown in Figs. 4.4 and 4.5, respectively, as a function of the optimum npn with the other decision variables fixed at their optimum value. The base case design is also indicated on these figures. As expected, designs obtained from irreversibility minimization generate less irreversibility but cost more than those obtained from thermoeconomics. The irreversibility and cost for the base case falls between both optima corresponding to the base case npn (npn=40), but the base case cost is substantially higher than that at the highest npn (npn=80) which corresponds to the optimum obtained from the present thermoeconomic analysis.

The pressure drops for the optimum designs differ substantially from those used in the base case. Gas-side pressure drops in the minimum irreversibility analysis fall below those of the base case. Air-side pressure drops in the minimum cost analysis are greater than the pressure drops of the base case. The base case values were selected as representative of values for metallic heat exchangers.

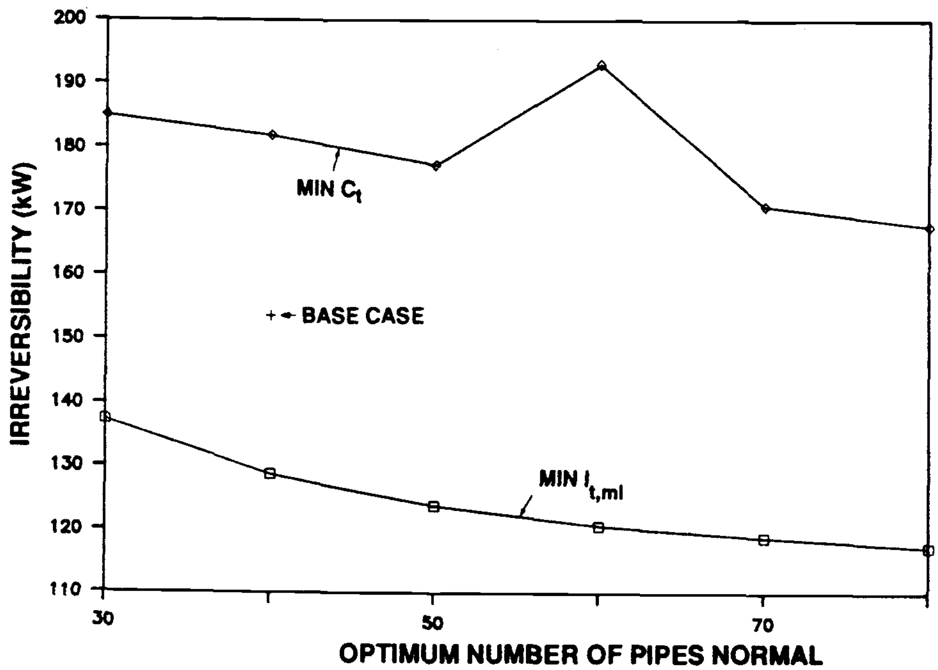

Figure 4.4. Irreversibility (not including $\dot{I}_{\Delta T}$) of the base case and optimum designs, as given in Tables 4.2 and 4.3.

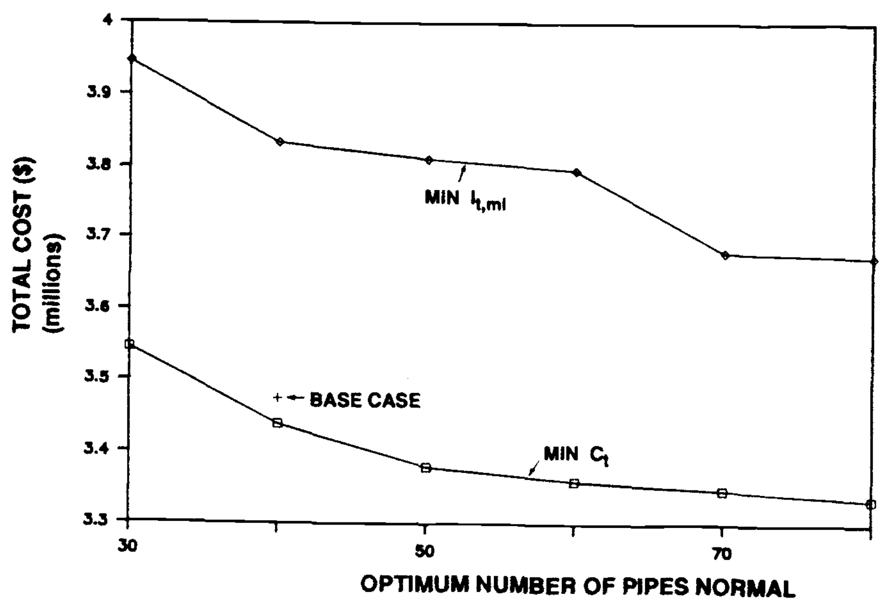

Figure 4.5. Total cost (area and pressure drop irreversibility) of the base case and optimum heat exchanger designs, as given in Tables 4.2 and 4.3, during the application life of the heat exchanger.

These results show that pressure drop values arbitrarily selected by a designer from past experience with metallic heat exchangers might be far from optimum for a given design condition.

As expected, the pressure drops obtained from the minimum irreversibility analysis are quite low. The thermoeconomic optimum shows that the high costs of heat exchanger area require substantially greater pressure drops than those prescribed for the minimum irreversibility optimum. Because losses on the gas-side are more costly than losses on the air-side, the gas-side pressure drops do not increase above the minimum values of the irreversibility minimization analysis nearly as much as do those of the air-side.

DISCUSSION

The work presented in this chapter illustrates the optimization procedure for a heat exchanger in a complex power plant. The specific case of a ceramic heat exchanger in a biomass fueled power plant is shown. The major conclusions drawn from the work can be categorized as those of a general nature and those specific to the system evaluated. They are:

General

* The presence of an overall system simulation code allows the irreversibility costs to be readily evaluated such

that thermoeconomic optimization of the heat exchanger as an isolated unit will result in optimization of the overall system with respect to the heat exchanger design variables.

* Optimization with the procedure of minimizing $\dot{I}_{t,mi}$ yields reasonable guideposts for design.
* Optimum pressure drops for ceramic heat exchangers need to be evaluated from an optimization rather than depend on approximate values obtained from past experience with metallic heat exchangers.

Specific

* Both optimization methods show that the optimum ceramic heat exchanger for the specific application favors a high npn (large width) and the maximum width is constrained by the minimum allowable tube gap.

CHAPTER 5

A PROCEDURE TO RECOGNIZE POTENTIAL PROCESS IMPROVEMENT WITH PROCESS INTEGRATION

As discussed previously in Chapter 1, the present preliminary screening procedures are based on the subjective judgment of an expert from a visual inspection of the process grand composite curve. The methods are also not very reliable in determining the potential of integrating a heat engine as an alternative to a heat pump.

The work presented in this chapter recognizes that the information gathered from a visual inspection of the grand composite curve is in fact the loss of thermodynamic potential (generation of irreversibility) in the process. A method to quantify the irreversibility generation, with and without process integration, is presented in this chapter. It is also proposed to use the information from a thermodynamically optimally integrated heat pump or heat engine (i.e: integrated so as to minimize the loss of thermodynamic potential), to screen different process plants or sectors of the same process plant for potential improvement with process integration. Candidate processes selected from the preliminary screening procedure must always then be evaluated using economics, to determine the economically optimum load and level of integration. The method of formulating the economic optimization problem is presented in chapter 6.

The present work is limited to the consideration of a single heat pump or heat engine integrated into the process. The work is directed at developing the method of determining the levels of integration and the heat loads of the heat engine or heat pump, since the actual network synthesis procedure for process streams with known load and temperature has been detailed elsewhere (Linnhoff and Hindmarsh, 1983).

TEMPERATURE INTERVAL IRREVERSIBILITY

The procedure to quantify the temperature interval irreversibility without and with process integration is presented in this section.[11]

Without Process Integration

The imaginary process composite streams are used to calculate the irreversibility of the heat exchange processes. A temperature interval in the process energy cascade is shown in Fig. 5.1. The temperatures T_{in} and T_{out} are the boundary temperatures of the specific temperature interval, Q_{in} and Q_{out} are the energy flows in and out of

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[11] Some of the terminology used here are those normally used in the pinch technology method. More details are given in Appendix A, where background material on the pinch technology method is presented.

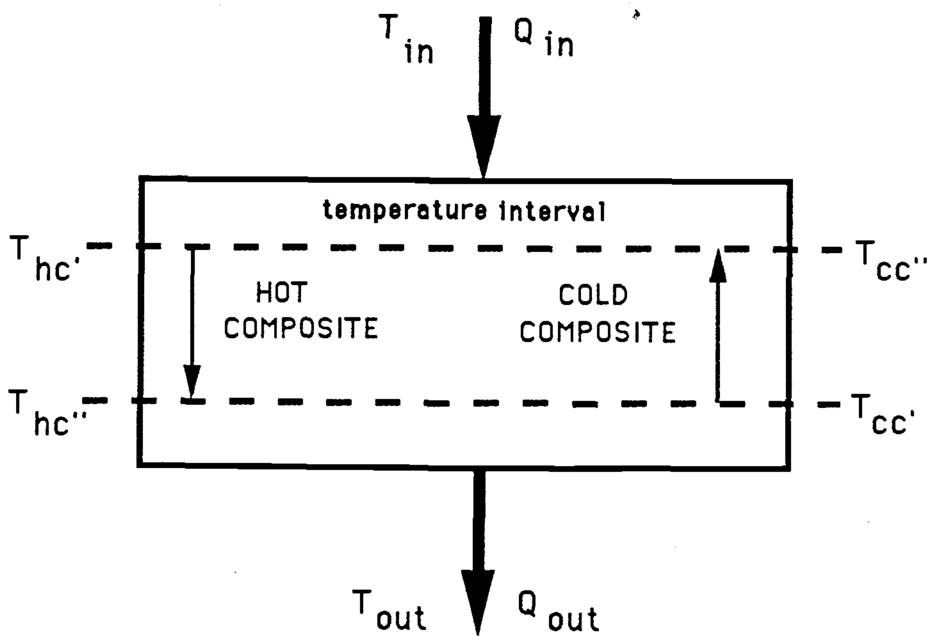

Figure 5.1. Energy flow in a temperature interval.

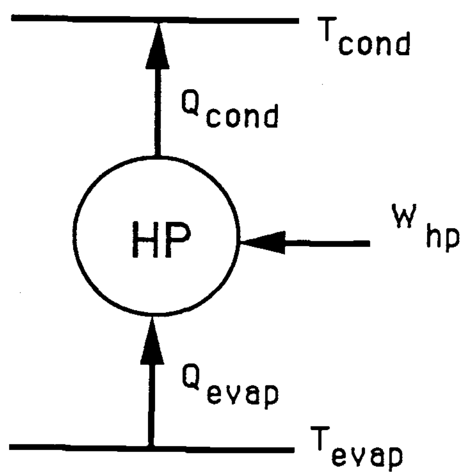

Figure 5.2. Energy flow through a heat pump.

the temperature interval respectively.[12] The terminal temperatures of the hot and cold composite streams are $T_{hc''}$, $T_{hc'}$, and $T_{cc''}$, $T_{cc'}$ respectively (the boundary temperatures can be selected so that these terminal temperatures correspond to the boundary temperatures). The capacity flow rates of the cold and hot composites are C_{cc} and C_{hc} respectively. An exergy balance for the temperature interval gives the following equation:

$$\dot{I}_{TI} = \dot{EX}_{in} - \dot{EX}_{out} + \dot{EX}_{hc} - \dot{EX}_{cc} \tag{5.1}$$

where,

$$\dot{EX}_{in} = \dot{Q}_{in} (1 - T_0/T_{in}) \tag{5.2}$$

$$\dot{EX}_{out} = \dot{Q}_{out} (1 - T_0/T_{out}) \tag{5.3}$$

$$\dot{EX}_{hc} = C_{hc} [(T_{hc'}-T_{hc''}) - T_0 \{\ln(T_{hc'}/T_{hc''})\}] \tag{5.4}$$

$$\dot{EX}_{cc} = C_{cc} [(T_{cc''}-T_{cc'}) - T_0 \{ \ln(T_{cc''}/T_{cc'})\}] \tag{5.5}$$

---

[12]The physical interpretation of these energy flows are as follows: For a temperature interval boundary above the process pinch, the energy crossing the temperature interval boundary is the surplus heating load available at that temperature, from the process streams and utilities at higher temperatures than the boundary temperature. This energy is used to heat cold process streams below this temperature. The energy flow at a boundary above the pinch is also a measure of the maximum heat load required from a condenser of a heat pump or heat engine operating at that boundary temperature. For a temperature interval boundary below the process pinch, the energy crossing the temperature interval boundary is the deficit in the cooling load at that temperature. It is also a measure of the maximum heating load available for the evaporator of a heat pump or heat engine operating at this temperature.

From the first law of thermodynamics,

$$\dot{Q}_{out} = \dot{Q}_{in} + C_{hc}(T_{hc'} - T_{hc''}) - C_{cc}(T_{cc''} - T_{cc'}) \tag{5.6}$$

Combining Equations (5.1)-(5.6), the resulting equation for the TI irreversibility is:

$$\dot{I}_{TI} = T_0 \left[\dot{Q}_{out}/T_{out} + C_{cc} \ln(T_{cc''}/T_{cc'}) - \dot{Q}_{in}/T_{in} - C_{hc} \ln(T_{hc'}/T_{hc''}) \right] \tag{5.7}$$

where T_0 is the dead state temperature. The total irreversibility for the heat exchange process is the sum of the temperature interval irreversibilities.

$$\dot{I}_{HEN} = \sum_{1}^{n} \dot{I}_{TI} \tag{5.8}$$

where n is number of temperature intervals. The above equation was formulated under the assumption that the temperature interval boundary temperatures were higher than the dead state temperature. If the interval boundary temperatures are lower than the dead state temperature, the temperature interval irreversibility has to be formulated by noting that the exergy increases with decreasing temperature.

<u>With Process Integration</u>

The integration of a heat engine or a heat pump results in the change of some energy flows in the process energy cascade. This is due to the evaporator and condenser heat exchangers. Once the working fluid streams of the evaporator or condenser have been included in the composite streams,[13] Equation (5.8) will give the total irreversibility in the process heat exchangers including the heat pump or heat engine exchangers. However, the irreversibility generated in the compressor, valves etc., of the heat pump or the irreversibility generated in the turbine, pumps etc., of the heat engine has to be calculated separately. The procedure to calculate these irreversibilities will be presented next.

The energy flow diagram for a heat pump is shown in Fig. 5.2. The temperatures of the evaporator and condenser are T_{evap} and T_{cond}, and, Q_{evap} and Q_{cond} are the energy flows respectively. The irreversibility in the heat pump minus the irreversibilities in the evaporator and condenser heat exchangers is given by,

$$\dot{I}_{hpmhx} = T_0 \, (\dot{Q}_{cond}/T_{cond} - \dot{Q}_{evap}/T_{evap}) \tag{5.9}$$

Application of the first law of thermodynamics, and the definition of the coefficient of performance for heat pumps (COP) gives the following relation between $\dot{Q}_{cond}$ and $\dot{Q}_{evap}$:

$$\dot{Q}_{cond} = \dot{Q}_{evap} \, (1 + 1/COP) \tag{5.10}$$

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[13]It is assumed for purposes here, that the temperature of these streams are constant (i.e: no superheating or subcooling).

Equation (5.9) can also be written in terms of the evaporator load and the temperature levels of integration as,

$$\dot{I}_{hpmhx} = T_0\, \dot{Q}_{evap} [\, 1/T_{cond} - 1/T_{evap} + 1/(T_{cond}\, COP)\,] \tag{5.11}$$

(Experimental correlations for COP as a function of T_{cond} and T_{evap} are available. However, the applicability of these correlations are restricted to specific conditions.)

The total irreversibility for the heat exchange process with a process integrated heat pump is,

$$\dot{I}_{HEN,hp} = \dot{I}_{HEN} + \dot{I}_{hpmhx} \tag{5.12}$$

The irreversibility of the heat engine minus the irreversibilities in the evaporator and condenser heat exchangers ($\dot{I}_{hemhx}$) can be calculated from a similar procedure to the one described above. The final expression for $\dot{I}_{hemhx}$ as a function of the levels of integration and the boiler load $\dot{Q}_{boil}$ is,

$$\dot{I}_{hemhx} = T_0\, \dot{Q}_{boil} [1/T_{cool} - 1/T_{boil} - \eta_{he}/T_{cool}] \tag{5.13}$$

where, η_{he} is the efficiency of the heat engine. The total irreversibility for the heat exchange process with a process integrated heat engine is,

$$\dot{I}_{HEN,he} = \dot{I}_{HEN} + \dot{I}_{hemhx} \tag{5.14}$$

THERMODYNAMICALLY OPTIMUM PLACEMENT

The formulation of the optimization problem and the solution procedure, for thermodynamically optimum placement of heat pumps and heat engines in processes, are presented in the next two sub sections.

Problem Formulation

The proposed objective function for the preliminary screening procedure is the total irreversibility generation in the heat exchange processes. For a process being screened for heat pump integration the objective function is given by Equation (5.12), and a process being screened for heat engine integration the objective function is given by Equation (5.14).

For a process integrated heat pump, the design variables are the condenser and evaporator temperatures T_{cond}, T_{evap}, and the heat load of the evaporator $\dot{Q}_{evap}$ (note that $\dot{Q}_{cond}$ or $\dot{W}_{hp}$ can be used as a design variable instead of $\dot{Q}_{evap}$, since, when the level of integration is fixed all three are dependent on each other). The design variables for a process integrated heat engine are the levels of integration T_{boil}, T_{cool}, and the boiler heat load $\dot{Q}_{boil}$. Again, $\dot{Q}_{cool}$ or $\dot{W}_{he}$ can be used as an alternative design variable to $\dot{Q}_{boil}$.

Upper and lower bounds for the temperatures are obtained from the rules given by Townsend and Linnhoff

(1983), for appropriate placement of heat pumps and heat engines in processes. The rule states that a heat pump should always be placed across the pinch and a heat engine should always be placed entirely above or below the pinch. An upper bound for the heat load is indirectly specified by preventing the shifting of the process pinch (This constraint will be discussed in detail in chapter 6). The ability to specify bounds for all three decision variables simplifies the optimization problem considerably.

Solution Method

A computer code was developed to calculate the value of the objective function and the constraints, for a given set of the decision variables. A relationship for the COP of a heat pump or the efficiency of the heat engine have to be written in the code.[14] The program listing is given in Appendix C. In addition to the design variables, the stream data, dead state temperature, and the hot utility temperature are also needed as inputs to the code. The cold utility is fixed at the dead state temperature, and all process streams are assumed to be above the dead state temperature. The program can be used for either a heat pump or heat engine optimization.

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[14] Note that these correlations are not general, and hence, have to be coded for each specific application. The correlations coded in the listing presented in Appendix C, for the COP of a heat pump and the efficiency of the heat engine, are those used for the example presented later in this chapter.

The above code is then coupled to an optimization code where the optimum values of the condenser temperature and heat load are determined, for a fixed evaporator temperature. This is a local optimum for the fixed evaporator temperature, and the optimization problem has two degrees of freedom. Since the upper and lower bounds for the evaporator temperature are specified, the global optimum corresponds to the minimum irreversibility generation from all the local optima within the specified bounds. The local optima are also saved since they are needed for the preliminary screening procedure proposed in the next section.

SCREENING PROCEDURE

A non-dimensional parameter, φ, is defined as,

$$\varphi = \frac{\dot{I}_{HEN,pi}}{\dot{I}_{HEN}} \tag{5.15}$$

where $\dot{I}_{HEN,pi}$ is the irreversibility of the heat exchange process with a process integrated heat engine or heat pump, and $\dot{I}_{HEN}$ is the irreversibility before integration. For a heat engine or heat pump optimally integrated in the process,

$$\varphi^{*} = \frac{\dot{I}^{*}_{HEN,pi}}{\dot{I}_{HEN}} \tag{5.16}$$

where $\dot{I}^*_{HEN,pi}$ is the heat exchange process irreversibility with an optimally integrated heat engine or heat pump. The non-dimensional parameter φ^* is a measure of the improvement to the system, from a thermodynamic point of view, due to process integration. The bounds of φ^* are, $0 < \varphi^* < 1$. The upper limit of φ^* corresponds to the case of no improvement due to process integration, and such processes can be eliminated from further consideration. The potential of a process for improvement with process integration increases as φ^* decreases. The relative magnitudes of φ^* are also indications of the thermodynamic potential of a heat engine over a heat pump or vice-versa, for integration in the process under consideration.

The non-dimensional parameter φ is based on the second law of thermodynamics. The coefficient of performance of a heat pump and the efficiency of a heat engine are similar performance evaluation parameters based on the first law of thermodynamics. Experience with the use of these first law based parameters allows them to be used to gain some insight to the potential economic feasibility for improvement with process integration.

A high COP^* or a high η^*_{he} corresponding to a low φ^* is an indication that the process is a good potential candidate for improvement with process integration. Such processes need to be evaluated in detail for economic feasibility. A process with a high φ^* can be eliminated from further consideration, irrespective of the values of the first law parameters. A process with a low φ^* and a low COP^* or a low η_{he}^* should not be rejected from further consideration immediately. In this case, the COP^{*L} or η_{he}^{*L} at the local optima φ^{*L} near the global optimum φ^* need to be investigated. This is because the φ^{*L} curve could be flat near φ^* resulting in there being a local optimum with a high COP^{*L} or η_{he}^{*L} with very little increase in φ^{*L} from φ^*. An example of this condition is presented in the next section.

Past experience of using the first law parameters (COP for a heat pump or efficiency for a heat engine) allows an expert to approximately specify the ranges where they are considered to be low or high. However, ranges of φ, where it is considered to be high or low, can not be specified with such accuracy at the present time. Hence, the main use of φ^* at this time would be as a tool for comparison, since more experience with the use of such a parameter is needed before the absolute value of φ could be interpreted more accurately.

Example

The initial screening procedure proposed in this work is illustrated in this section with an example.

a) Problem Statement:

Process stream data from two sections of a process plant (the two sections will be referred to as section A and section B) are given in Tables 5.1 and 5.2 respectively. Heat exchange between process streams in section A with process streams in section B are not allowed due to practical reasons. Minimum approach temperature for all process streams including the evaporator and condenser streams of a process integrated heat pump or heat engine is 10 K. The hot utility produced from the utility producing section of the process plant is available at a temperature of 443 K. The cold utility is at dead state temperature of 293 K. It has been decided to investigate the energy saving potential for the process plant with a process integrated heat pump or heat engine. Assume that the COP for a heat pump is given by $1 - COP^{-1} = 0.9277 - 0.004609\,(T_{cond} - T_{evap})$,[15] and the efficiency of a heat engine is approximately 0.5 of the Carnot efficiency. Use the proposed preliminary screening procedure to determine, which section of the process plant has the greatest potential for improvement with a process integrated heat pump or heat engine.

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[15] This is an experimental correlation (El-Meniawy et al. (1981)) valid for temperature lift in the range of 20 to 50 C, for a water-to-water heat pump system using R22. Although this expression is used as an approximate expression for the COP of the heat pump for the present example, it should be noted that experimentally determined correlations of the COP of heat pump systems cannot be generalized for different heat pump systems or different conditions of operation.

Table 5.1. Process stream data for section A

stream name	initial temp. (K)	target temp. (K)	capacity flow rate (MW/K)
AH1	383	333	0.1
AH2	333	313	1.0
AC1	313	383	0.5

Table 5.2. Process stream data for section B

stream name	initial temp. (K)	target temp. (K)	capacity flow rate (MW/K)
BH1	383	333	0.1
BH2	333	323	4.0
BC1	313	343	2.0

Compare the above results with results obtained by using the present screening procedures.

b) Results:

The results presented in this section were obtained from an optimization with the objective function and constraints computed by using the computer code presented in Appendix C.

(i) The variation of the local optimum φ^{*L} with the evaporator temperature, for process integrated heat pumps in sections A and B, are shown in Fig. 5.3 and Fig. 5.4 respectively. The optimum temperature lift corresponding to each evaporator temperature for sections A and B are shown

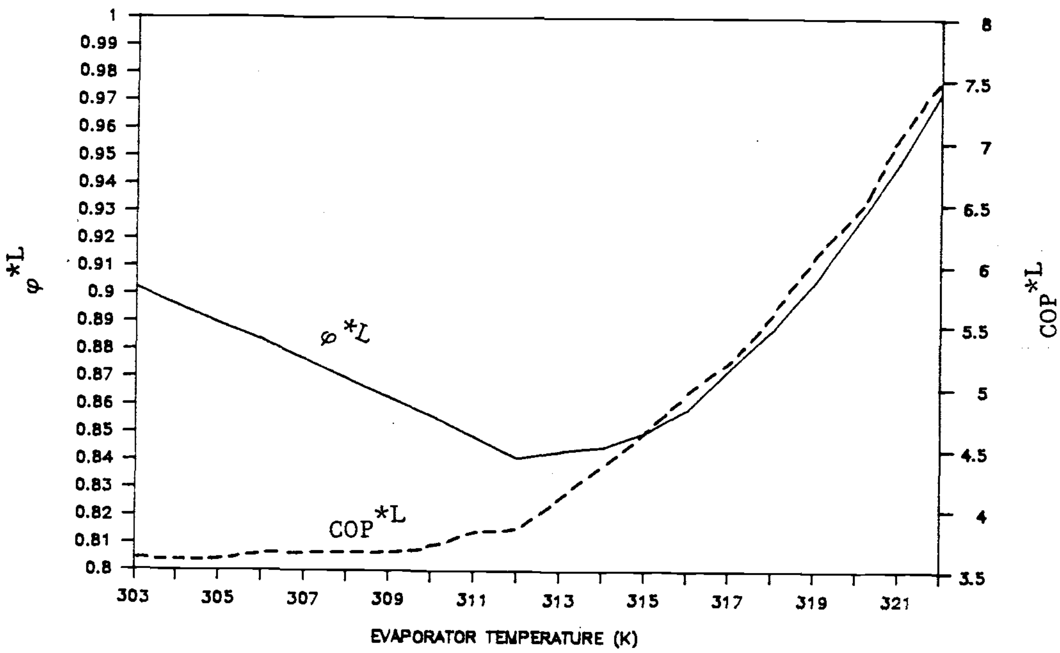

Figure 5.3. The variation of the φ^{*L} and COP^{*L} with the evaporator temperature, for a thermodynamically optimally placed heat pump in section A.

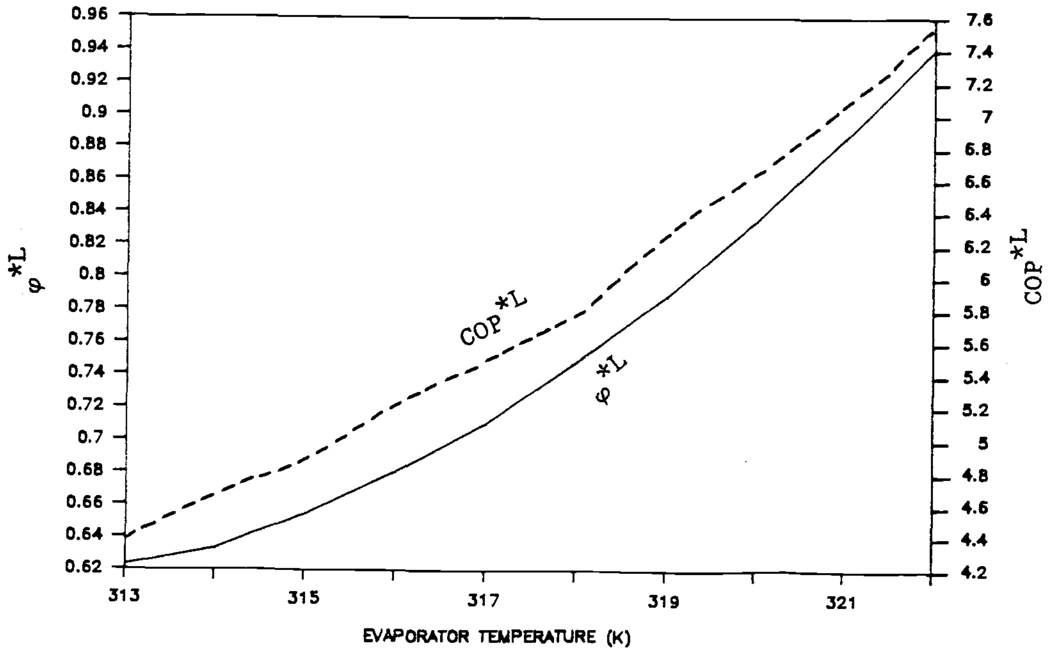

Figure 5.4. The variation of φ^{*L} and COP^{*L} with the evaporator temperature, for a thermodynamically optimally placed heat pump in section B.

in Fig. 5.5 and Fig. 5.6, respectively. The global optimum φ^* corresponds to the minimum value of φ^{*L}. Hence, for section A (Fig. 5.3) φ^* is approximately equal to 0.84, and for section B (Fig. 5.4) φ^* is approximately equal to 0.625. Therefore, section B has more potential from a thermodynamic point of view for improvement with a process integrated heat pump. Although the COP^* of the optimum heat pump in section A (approximately 4.5) is slightly higher than the COP^* for the optimum heat pump in section B (approximately 4.3), a heat pump with COP^{*L} of 4.5 in section B would still have a φ^{*L} approximately equal to 0.65. The high COP^{*L} also suggest that a process integrated heat pump is likely to be economically viable.[16] Hence, section B is thermodynamically and economically more suitable for improvement with a process integrated heat pump.

The variation of φ^{*L} with the evaporator temperature, for a process integrated heat engines in section A and B, are shown in Fig. 5.7 and Fig. 5.8 respectively. φ^* is approximately equal to 0.81 and 0.71 for sections A and B

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[16] The variation of the local optimum temperature lift for the heat pumps at various evaporator temperatures, for sections A and B of the process plant, are shown in Figs. 5.7 and 5.8, respectively. These two figures show that the temperature lift is less than 20 C above an evaporator temperature of 319 K for section A (Fig. 5.5) and above an evaporator temperature of 318 K for section B (Fig. 5.6). Hence, the COP values presented in Figs. 5.5 and 5.6 for evaporator temperatures above 319 K for section A and 318 K for section B are not very reliable. The very high COP for process integrated heat pumps in this region is partly due to the inaccuracy of the correlation in this region.

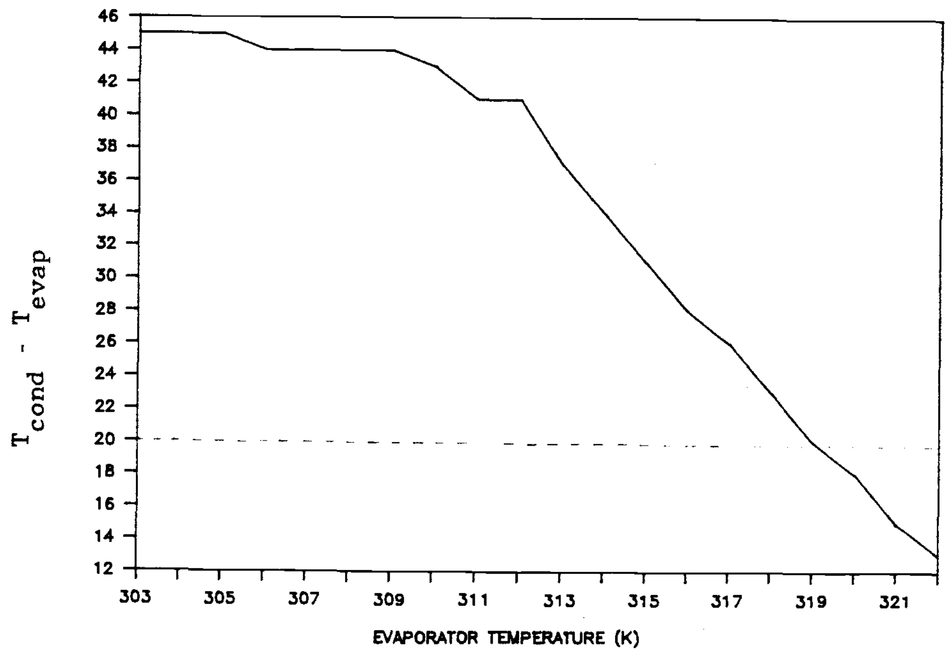

Figure 5.5. The variation of the local optimum temperature lift with the evaporator temperature, for section A.

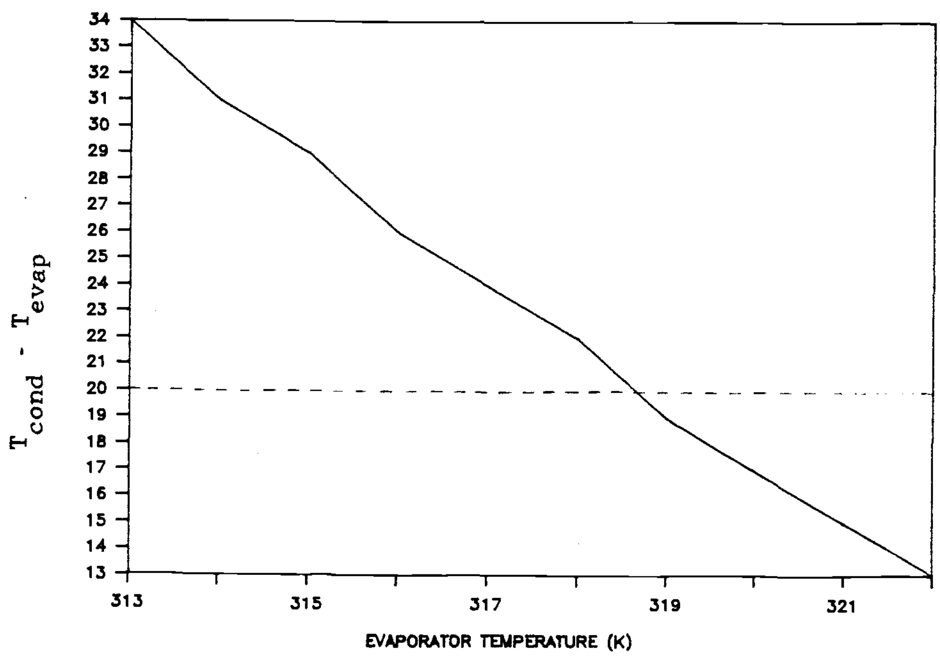

Figure 5.6. The variation of the local optimum temperature lift with the evaporator temperature, for section B.

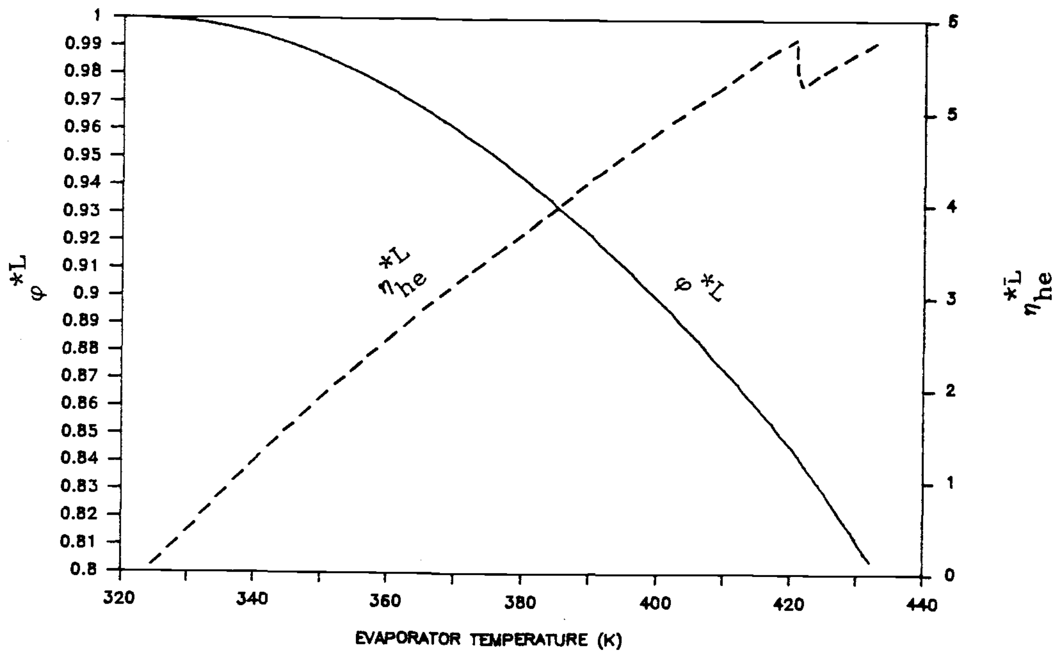

Figure 5.7. The variation of the local optimum φ^{*L} and η_{he}^{*L} with the evaporator temperature, for a thermodynamically optimally placed heat engine in section A.

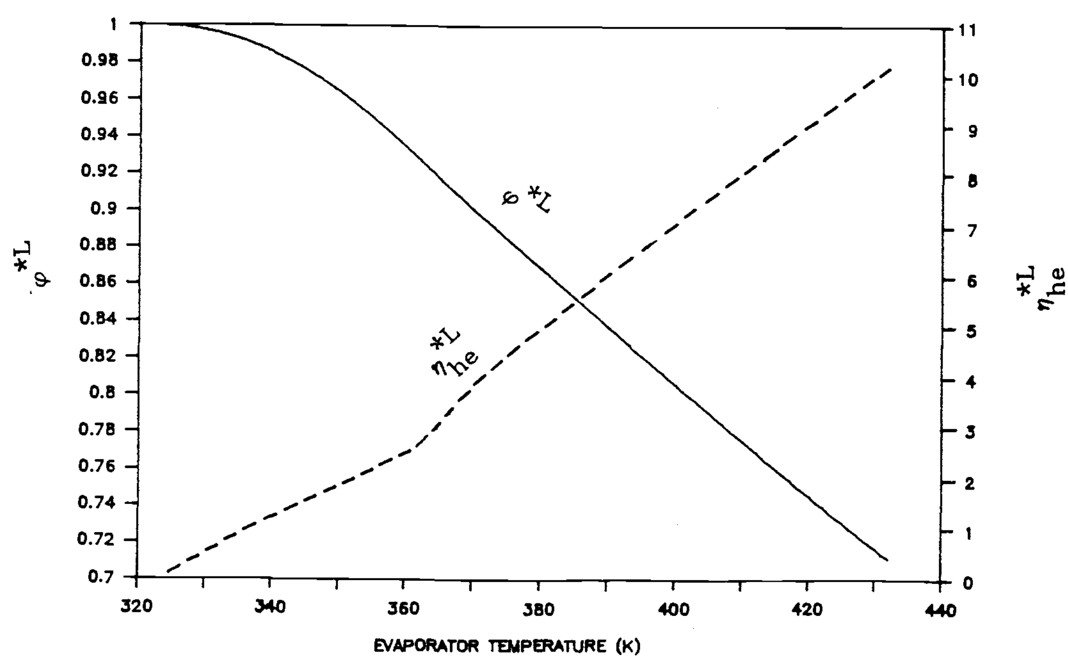

Figure 5.8. The variation of the local optimum φ^{*L} and η_{he}^{*L} with the evaporator temperature, for a thermodynamically optimally placed heat engine in section B.

respectively. The corresponding efficiencies are 5.75% and 10% respectively, and these efficiencies are the maximum in the domain of the decision variables. Hence, section B is also more suitable for heat engine integration than section A. However, the low efficiency suggest that even section B is not a candidate for improvement, since such a low efficiency heat engine is not expected to be economically feasible.

The initial screening procedure has thus recognized that section B has the greatest potential for improvement with process integration, and a process integrated heat pump has the greatest energy saving potential for this section of the process plant. Thermodynamically optimum load and level of integration are also obtained from this method. Once the initial screening procedure recognizes the potential for improvement, it is necessary to carry out a detailed economic evaluation to decide on the optimum heat loads and levels of integration.

(ii) The screening procedures employed at the present time are based on the process grand composite curve. The grand composite curves for section A and section B of the process plant are presented in Fig. 5.9 and Fig. 5.10 respectively. Both processes have a well defined pinch. However, the grand composite curve for section B has a narrower pinch region, thus indicating that a process integrated heat pump has more potential over a process integrated heat pump in

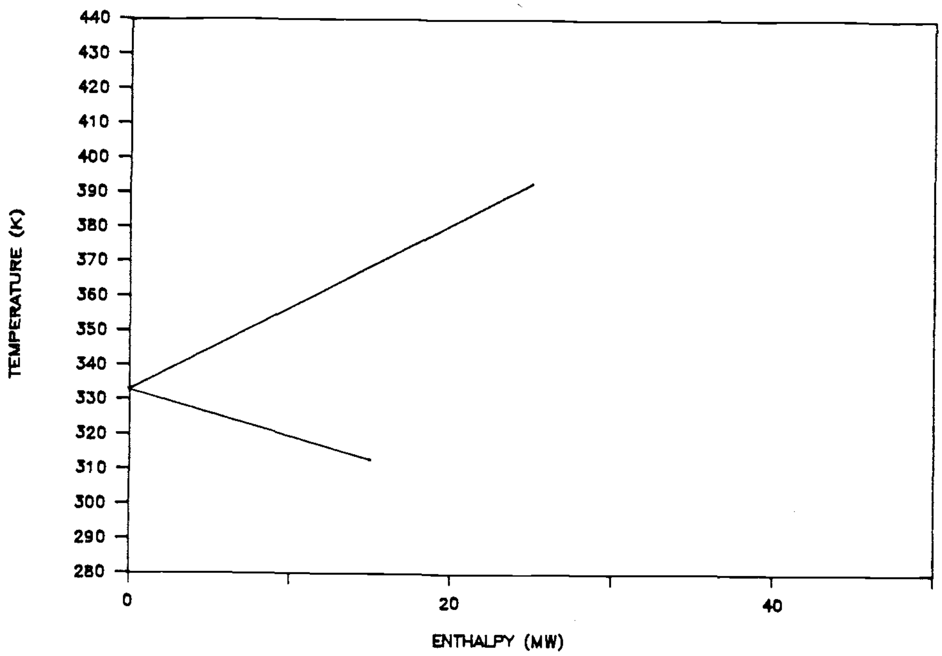

Figure 5.9. The grand composite curve for section A.

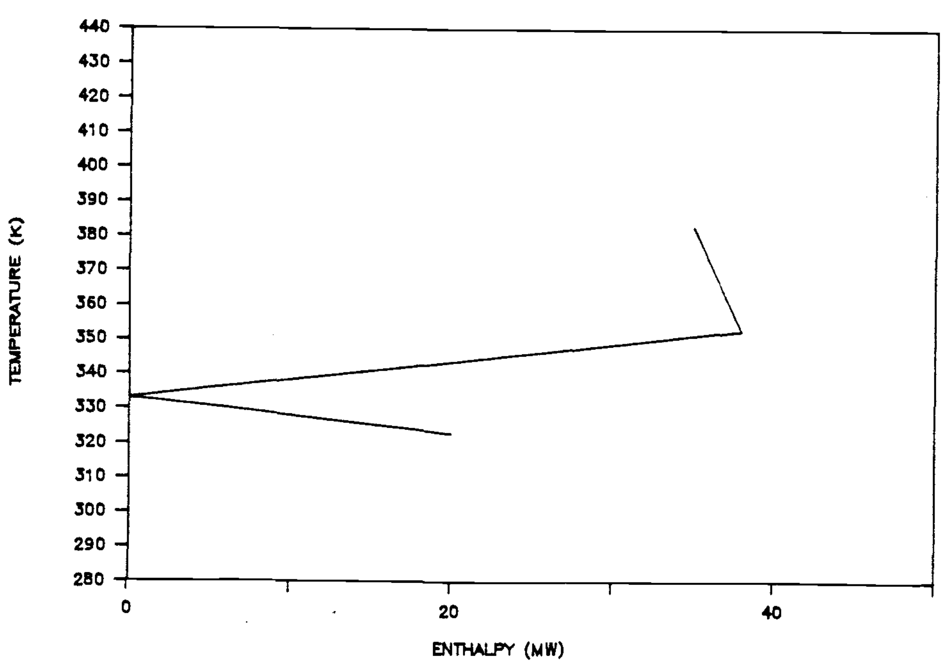

Figure 5.10. The grand composite curve for section B.

section B. The approximate COP for a process integrated heat pump has to be obtained by picking two temperatures from a visual inspection. The maximum heat engine efficiency would suggest that a heat engine might not be feasible for either case.

Hence, an expert might be in a position to interpret, from the grand composite curve, the same conclusions drawn from the proposed preliminary screening procedure. However, the proposed procedure gives quantitative indicators on which the decision could be based, rather than depend on one's intuition. Quantitative indicators also could be more beneficial in designing an automated screening procedure.

DISCUSSION

A method for the preliminary screening of processes to recognize the potential for improvement with process integration of heat pumps and heat engines have been developed. The method is based on the exergy concept. The method presented in this work requires much less subjective judgement from an expert, compared to the present preliminary screening procedures. The application of the method to a specific example was illustrated.

CHAPTER 6

OPTIMUM PLACEMENT OF HEAT PUMPS AND HEAT ENGINES IN PROCESSES

Candidate processes selected from the preliminary screening procedure has to be evaluated for economic feasibility. The objective of the economic optimization is to determine the set of design variables which would correspond to the minimum operational cost, and also satisfy the design constraints. The formulation of the economic optimization problem for the two cases: (i) process integrated heat pump and, (ii) process integrated heat engine are presented in this chapter. Once the problem is formulated as explained below, commercially available optimization codes (for example MINOS 5.1 developed by Murtagh and Saunders, 1987) are capable of performing the optimization.

PROCESS INTEGRATED HEAT PUMP

The objective of the optimization is to determine the values of T_{evap}, T_{cond} and $\dot{Q}_{evap}$ (once these three variables are known, $\dot{W}_{hp}$ and $\dot{Q}_{cond}$ can be determined), corresponding to the minimum total operational cost. The economic objective function, and some constraints for the design variables are developed in the next two sub sections.

Objective Function

The total variable operational cost of a process with a process integrated heat pump consists of the following costs:

(i) cost of hot and cold utilities

(ii) cost of heat exchanger area including the evaporator and condenser heat exchangers of the heat pump.

(iii) cost of shaft work required by the compressor of the heat pump.

(iv) cost of heat pump compressor, valves etc.

A process integrated heat pump adds a capital cost component to the total operational cost. However, a properly integrated heat pump reduces the hot and cold utility consumption which results in a cost saving. There is also a change in the cost of the total heat exchanger area. Hence, the objective of an economic evaluation is to determine the best combination of the design variables so that the sum of the above costs are a minimum. Hence, the objective function for a process integrated heat pump is,

$$\dot{C}_{HEN,hp} = \sum_{1}^{n_{HU}} c_{HU}\, \dot{Q}_{HU} + \sum_{1}^{n_{CU}} c_{CU}\, \dot{Q}_{CU} + \dot{C}_{A,HEN} + c_e\, \dot{W}_{hp} + \dot{C}_{hpmhx} \qquad (6.1)$$

where c_{HU} and c_{CU} are the incremental cost of the hot and cold utilities respectively. The first two summations are over the total number of hot utilities and number of cold utilities respectively. The hot and cold utility consumption is denoted by $\dot{Q}_{HU}$ and $\dot{Q}_{CU}$ respectively. $\dot{C}_{A,HEN}$ is the cost of the total heat exchanger area, and $\dot{C}_{hpmhx}$ is the cost of the heat pump compressor, valves etc.

The cost of hot and cold utilities is most often a function of the amount of utility consumed. This is specially true if the utility is generated in the same process plant in which the heat pump is to be integrated. Using incremental cost for the utilities would ensure that the optimally placed heat pump would correspond to an optimum operational cost for the total process plant and not just for the heat exchange section of the plant (the concept of using incremental cost for thermal system component analysis was first presented by El-Sayed and Evans, 1970). Assuming constant cost for the utilities can give designs which are far from optimum with respect to the total process plant. These incremental cost coefficients should be calculated by considering the exergy content of the utility rather than the energy content. This is because most often utilities are co-generated in process plants, and energy costing methods have been shown to be unreliable for costing co-generated utilities (Gaggioli, 1977). Details on the exergy method of costing utilities -- commonly referred to as thermoeconomics -- can be found in Gaggioli (1977) and Reistad and Gaggioli (1980).

For a given set of decision variables, the other terms in the objective function are obtained as follows: The utility consumption can be obtained from the pinch technology method. A method to obtain the approximate heat exchanger area is given by Townsend and Linnhoff (1984). The COP for the heat pump is fixed for a given level of integration. The COP has to be obtained from empirical correlations (which are restricted for specific applications) or experimental data. Once the COP is known, $\dot{W}_{hp}$ can be calculated. The cost of the heat pump compressor can be obtained from empirical correlations or from the manufacturers.

Constraints

Bounds for T_{evap} and T_{cond} can be obtained from the rules given by Townsend and Linnhoff (1983). These authors showed that a heat pump should always be integrated across the pinch to save energy. Hence, the lower limit of T_{cond} is the pinch temperature, and the upper limit is ΔT_{min} (i.e: the minimum approach temperature between the hot and cold streams) above the maximum cold stream temperature. The maximum limit of T_{evap} is ΔT_{min} below the pinch temperature, and the minimum is ΔT_{min} below the lowest hot stream temperature.

The lower limit of $\dot{Q}_{evap}$ corresponds to the condition of no integration. A method to specify the upper limit of $\dot{Q}_{evap}$ is explained next.

Consider a process with a pinch, and a single hot utility and a single cold utility. The process energy cascade of such a process is shown in Fig. 6.1. The hot and cold utility requirement for this process is $\dot{Q}_{HU}$ and $\dot{Q}_{CU}$ respectively. If a heat pump is integrated across the pinch at temperatures T_{evap} and T_{cond}, and $\dot{Q}_{evap}$ is increased from zero, an additional pinch could either occur at or below the evaporator temperature,or , at or above the condenser temperature. First consider the case of a heat pump integrated across the pinch, as shown in Fig. 6.2(a). Increasing $\dot{Q}_{evap}$ from zero to its present value has resulted in an additional pinch point at a temperature above the condenser temperature. The energy flow in the process energy cascade with an additional incremental energy flow $d\dot{Q}_{evap}$ from the evaporator, is shown in Fig. 6.2(b). An energy balance of the process energy cascade shows that the result of this incremental energy flow in the evaporator is an increase in the cold utility consumption by $d\dot{W}_{hp}$. Hence, increasing $\dot{Q}_{evap}$ above the value indicated in Fig. 6.2(a) is not desirable, and this becomes an upper limit for $\dot{Q}_{evap}$ for this level of integration. It should be noted that this upper limit of $\dot{Q}_{evap}$ is always a function of the other two design variables T_{evap} and T_{cond}. Fig. 6.3(a) shows a process where the additional pinch occurs at a temperature below the evaporator temperature, when $\dot{Q}_{evap}$ is increased from zero. Fig. 6.3(b) shows that an incremental evaporator load $d\dot{Q}_{evap}$ results in the saving of hot utility equal to

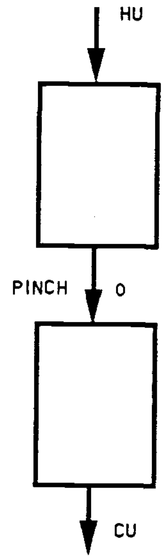

Figure 6.1. Process energy cascade of a process with a pinch and a single hot utility and a single cold utility.

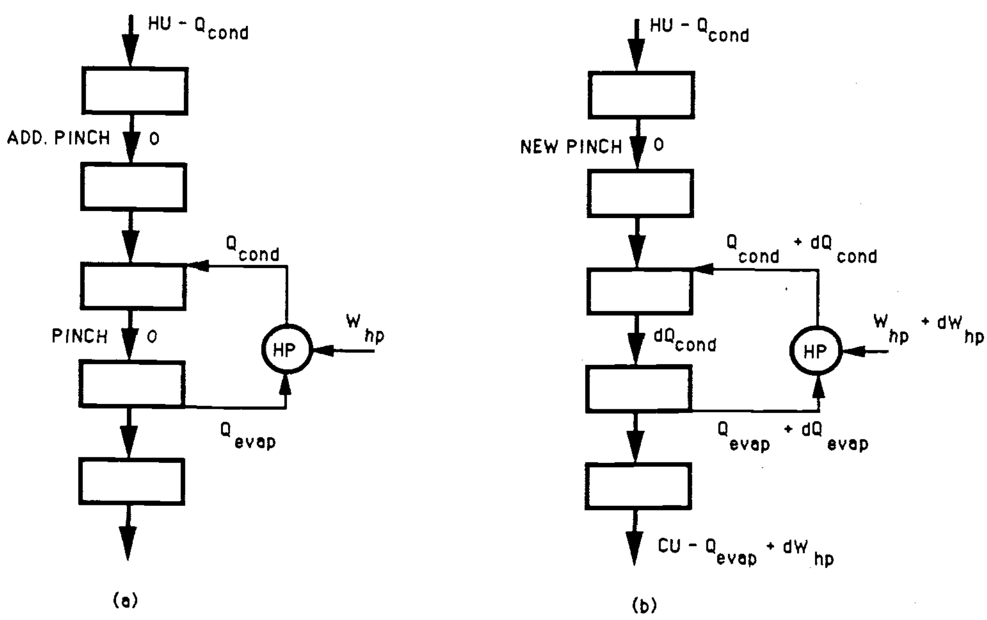

Figure 6.2. The process energy cascade of a process with a heat pump integrated across the pinch. (a) The condenser load has been increased to a point where an additional pinch first appears above the condenser temperature. (b) The effect of increasing the evaporator load by dQ_{evap}, above the evaporator load indicated in (a).

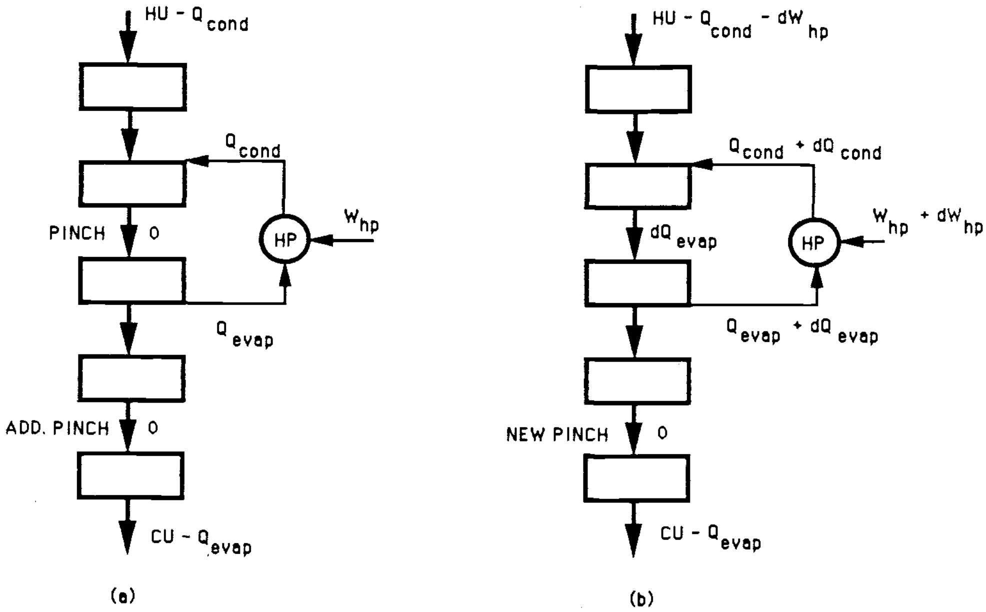

Figure 6.3. The process energy cascade of a process with a heat pump integrated across the pinch. (a) The evaporator load has been increased to a point where an additional pinch first appears below the evaporator temperature. (b) The effect of increasing the evaporator load by dQ_{evap}, above the evaporator load indicated in (a).

the incremental compressor work $d\dot{W}_{hp}$. However, shaft work is always more valuable than an utility with the same energy content. Hence, $\dot{Q}_{evap}$ should not be increased above the value in Fig. 6.3(a), and this value of the evaporator load is an upper limit for a specified level of integration.

Hence, the above results can be summarized as follows: The upper limit $\dot{Q}_{evap}$ is a function of the level of integration (i.e: T_{evap} and T_{cond}). The upper limit of $\dot{Q}_{evap}$ should be specified so as to prevent the occurrence of an additional pinch in the process.

PROCESS INTEGRATED HEAT ENGINE

The objective of the optimization is to select the level of integration, T_{boil} and T_{cool}, and the loads, $\dot{Q}_{boil}$ (once these three variables are known $\dot{Q}_{cool}$ and $\dot{W}_{he}$ can be determined), so that the total operational cost is a minimum. The economic objective function and the constraints in the optimization problem for a process integrated heat engine is presented in the next two sub sections.

Objective Function

The total variable cost of a process with a process integrated heat engine consists of the following costs:

(i) cost of hot and cold utilities

(ii) cost of the heat exchanger area including the cost of the evaporator and condenser of the heat engine.

(iii) income from the shaft work generated in the heat engine.

(iv) cost of the heat engine turbine, pumps etc.

The economic objective function for a process integrated heat engine is,

$$\dot{C}_{HEN,he} = \sum_{1}^{n_{HU}} c_{HU}\,\dot{Q}_{HU} + \sum_{1}^{n_{CU}} c_{CU}\,\dot{Q}_{CU} + \dot{C}_{A,HEN} + \dot{C}_{hemhx} - c_e\,\dot{W}_{he} \qquad (6.2)$$

where $\dot{C}_{hemhx}$ is the cost of the heat engine turbine, pumps etc., and $\dot{W}_{he}$ is the shaft power generated in the heat engine.

All the terms in the objective function, which were also present in Equation (6.1), are calculated similar to the case of a process integrated heat pump. The efficiency of the heat engine and a cost of the turbine have to be obtained from empirical correlations or from the manufacturers.

Constraints

Upper and lower bounds for the levels of integration can be specified for this case too. However, an upper bound for the load has to be specified indirectly with the aid of an additional design constraint.

A rule for the process integration of heat engines was also given by Townsend and Linnhoff (1983). The rule is, a heat engine should always be integrated completely above the process pinch or completely below the process pinch, and a heat engine should never be integrated across the process pinch. This rule provides upper and lower bounds for the evaporator and condenser temperatures of the heat engine.

A lower limit for $\dot{Q}_{boil}$ corresponds to the case of no integration. A heat engine integrated completely above the pinch can have an additional pinch only at or above the condenser temperature and below the evaporator temperature, when $\dot{Q}_{boil}$ is increased from zero (see Fig. 6.4a). The power generated after the occurrence of the pinch would cost more than before (see Fig. 6.4b, which shows that an incremental amount of power is now generated by consuming more utilities than before), but this could still be economically feasible (i.e: power generated would still be cheaper than the power generated from a stand-alone heat engine operating between two utilities). The prevention of an additional pinch point occurring can not be used as a constraint in this case. Hence, for a process integrated heat engine optimization problem, an additional constraint has to be always specified. This constraint could be the maximum capacity of the turbine, or the maximum hot or cold utility available, and this constraint would provide the upper bound for $\dot{Q}_{boil}$.

One more condition has to be always considered in determining the optimum process integrated heat engine.

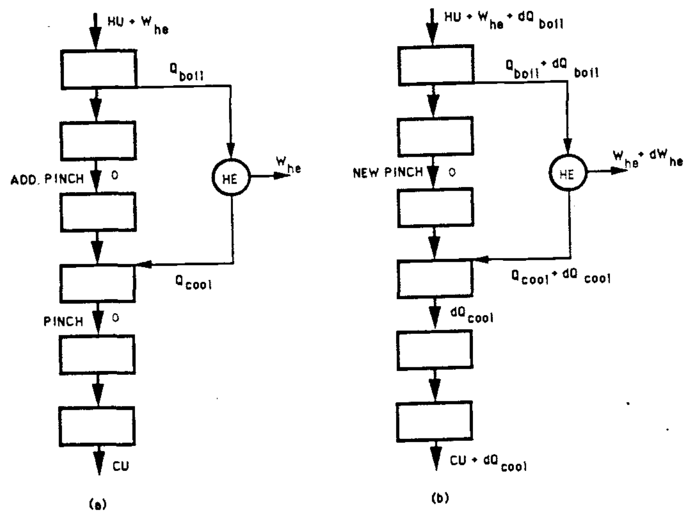

Figure 6.4. The process energy cascade of a process with a heat engine integrated completely above the pinch. (a) The condenser load has been increased to a point where an additional pinch first appears between the evaporator and condenser temperature. (b) The effect of increasing the evaporator load by dQ_{evap}, above the evaporator load indicated in (a).

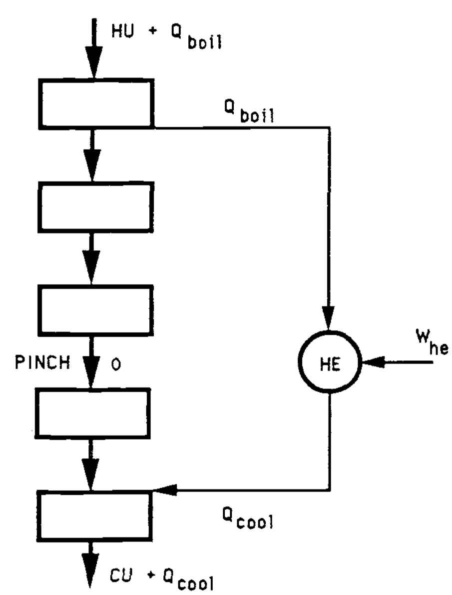

Figure 6.5. The process energy cascade of a process with a heat engine integrated across the pinch.

Consider a heat engine integrated across the pinch (integrating completely above or completely below the pinch is prevented), as shown in Fig. 6.5. The increase in hot and cold utility for this case is equal to the evaporator and condenser loads respectively. Hence, the optimum T_{boil} and T_{cool} has to obviously correspond to the hot and cold utility temperatures respectively. The optimum heat engine integrated across the pinch is one operating between the hot and cold utilities. This condition, although violating the rule given by Townsend and Linnhoff (1983), have higher efficiencies than a heat engine integrated completely above or completely below the pinch, and can be economically more attractive than the optimum heat engine subjected to the design rules for proper integration.

DISCUSSION

The procedure of formulating the economic optimization problem and specification of the bounds for the design variables were presented. Although presently used design rules are reliable for heat pump integration, it was shown that present design rules can be too restrictive for the case of heat engine integration. It was also shown that bounds for the maximum heat pump load can be obtained by considering the pinch points of the process, but an additional constraint had to be specified in the case of a process integrated heat engine.

CHAPTER 7

CLOSURE

The major conclusions drawn from the work presented in this thesis, and potential related areas for future research are presented in this chapter.

CONCLUSIONS

Characterization of heat exchanger design conditions allow potential trade-off options to be presented in a generalized form. Heat exchanger design conditions were characterized in this thesis, and potential trade-off options were discussed. Since, heat exchangers are always a part of a larger thermal system, their interaction with other plant components need to be considered during the design process. Additional design constraints resulting from such interactions were illustrated by considering the evaporator of a heat pump.

The irreversibility method for heat exchanger design allows conceptually beneficial guideposts to be established for heat exchangers. The inclusion of the material irreversibility in the irreversibility rate based objective function was shown to make these guideposts more realistic for heat exchangers. Exergetic efficiency expressions that similarly include a material irreversibility term show physically more realistic values than the usual expressions that don't include such a term.

The thermoeconomic method allows heat exchangers to be isolated and optimized, when information and time lines are appropriate, so that the performance of the total system also improves. This is possible due to the incremental cost coefficients which account for the interactions with other plant components. However, calculation of these cost coefficients can be difficult for heat exchangers residing in complex systems. The method presented in this thesis can be used to calculate these cost coefficients for heat exchangers residing in complex systems, if a system simulation model exists.

The application, and the resulting designs obtained from the irreversibility minimization and thermoeconomic methods were illustrated by considering an emerging technology ceramic heat exchanger residing in a complex system. As expected, the irreversibility method gives designs which provide a lower bound for the irreversibility and upper bound for the cost in an "economically optimum heat exchanger."

A method based on the exergy concept was developed in Chapter 5, to determine the potential of a process for improvement with a process integrated heat pump or heat engine. The method requires much less subjective judgment than the presently advocated methods. The application of the method to a specific example was illustrated.

The procedure of formulating the optimization problem to determine the optimum loads and levels of integration for heat pumps and heat engines were presented in Chapter 6. Additional bounds for design variables were also developed in this chapter. Such bounds simplify the optimization procedure.

POTENTIAL AREAS FOR FUTURE RESEARCH

The following are some of the potential areas of future study, to extend the work of this thesis.

* The characterization of heat exchanger design conditions allows various trade-off options to be identified. It would be useful to develop an expert system which is capable of identifying the trade-off options, given the information regarding the heat exchanger design conditions. This would assist those with limited expertise in the exergy method with the formulation of the optimization problem.
* Exergetic efficiency of the manufacturing process of common heat exchanger configurations need to be evaluated. This would allow the modified irreversibility method to be applied more widely.
* Thermal system or component optimization requires much subjective judgement of the designer. Some of these decisions are based on the designer's expertise in a particular field, and such expertise can not be written most often in terms of mathematical equations as design

constraints. However, such expertise could be incorporated into an expert system. If the methodology is developed to couple such expert systems to the optimization code, with the expert system module acting as an advisor to the optimization code, the design process could be automated. Hence, developing the methodology to couple expert systems and optimization codes, and application of the method initially for the design optimization of heat exchangers in simple systems is recognized as an area of potential research in the future.

* The preliminary screening procedure developed in this thesis uses quantitative indicators based on the exergy concept to recognize potential process improvement with process integrated heat pumps and heat engines. Similar quantitative indicators should be developed to assist in the synthesis of total thermal systems.

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APPENDICES

APPENDIX A

BACKGROUND ON THE PINCH TECHNOLOGY METHOD

The pinch technology method is a strong tool for the optimal synthesis of heat exchanger networks. In this method, all hot process streams (i.e: process streams requiring cooling) are combined to form a single imaginary hot composite stream, and all cold process streams (i.e: process streams requiring heating) are combined to form an imaginary cold composite stream. In a temperature-enthalpy plot of the composite streams, the point at which the hot and cold composite curves have the minimum allowable temperature difference (ΔT_{min}) corresponds to the process pinch. The minimum allowable temperature difference of the composite streams is determined so as to minimize the total operational cost of the heat exchanger network. A process with a pinch point has a net energy deficit above the pinch and a net energy surplus below the pinch. The net energy deficit of the composite curves is the hot utility requirement, and the net energy surplus of the composite curves is the cold utility requirement. The locus of the horizontal distance between the hot and cold composite curves is referred to as the grand composite curve. Hence, the grand composite curve is a plot of the energy deficit above the pinch and energy surplus below the pinch, as a function of the temperature.

The composite curves, process pinch and the utility targets for an example are presented next. The stream data for the example are shown in Table A.1. This process has two hot streams (H1 and H2) and one cold stream (C1). The ΔT_{min} for all process streams is assumed to be 10 C. A plot of the composite curves on a temperature-

Table A.1. Stream data for example.

Streame Name	Supply Temp. (C)	Target Temp. (C)	Heat Load (MW)
H1	60	10	5
H2	10	-10	20
C1	-10	60	35

enthalpy diagram is shown in Fig. A.1. The process pinch and the hot and cold utility requirement for the process are also indicated in this figure. The grand composite curve for this process is shown in Fig. A.2.

A region on the composite curve bounded by two temperatures is referred to as a temperature interval (TI). The composite curve is divided into many of these temperature intervals for the analysis leading to the calculation of the utility targets, and determination of the surplus or deficit of energy at any temperature in the process. The minimum number of heat exchanger units needed for the synthesis can also be obtained ahead of design. Having this information, Linnhoff and Hindmarsh (1983) give

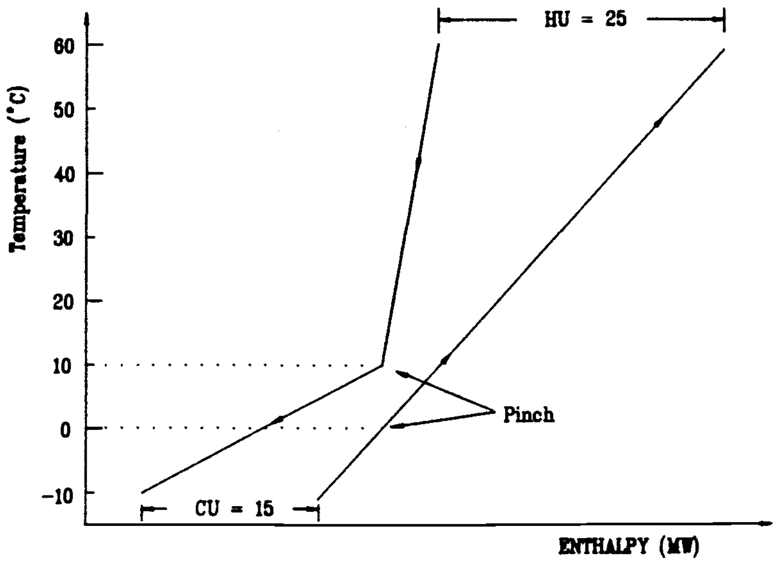

Figure A.1. Composite curves for example.

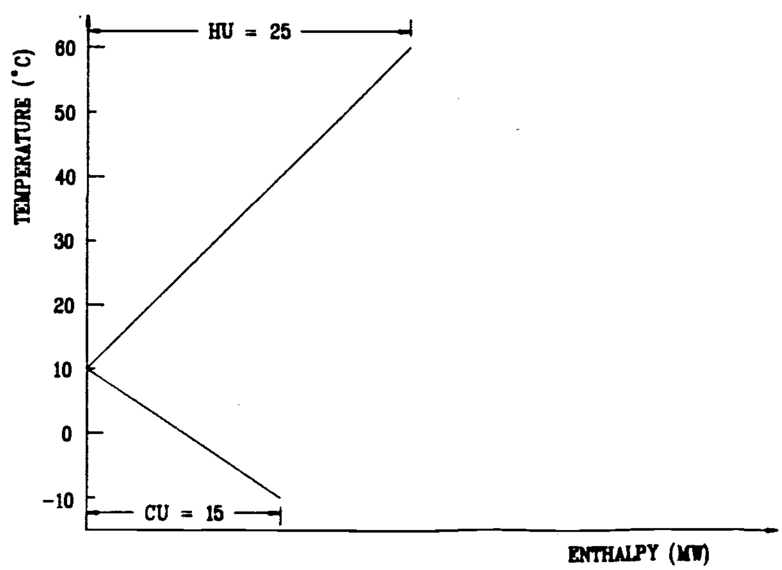

Figure A.2. Grand composite curve for the example.

rules to synthesize the heat exchanger network to meet the utility and cost targets predicted ahead of design. The above reference also gives a detailed description of the pinch technology method.

When using the pinch technology method for the synthesis of heat pumps or heat engines in processes, the working fluid in the heat pump or the heat engine is also considered as a process stream. For a heat pump, the condenser and evaporator streams are the hot and cold streams respectively. For a heat engine, the condenser and boiler streams are the hot and cold streams respectively. The hot and cold streams of the heat pump and heat engine are combined with process streams to obtain the imaginary composite streams. The change in the targeted utility consumption and cost can now be calculated from methods given by Linnhoff and co-workers. Finally, the heat exchanger network, including the heat pump or heat engine exchangers, can be synthesized to meet the predicted utility and cost targets by following the procedure given by Linnhoff and Hindmarsh (1983).

APPENDIX B

CERAMIC HEAT EXCHANGER SUBROUTINE

```
***********   CERAMIC CROSS FLOW HEAT EXCHANGER  *********
C......................Fixed Size Model.................
C
C  This subroutine simulates a multiple gas side pass,
C  ceramic cross-flow heat exchanger.  The size of the heat
C  exchanger is specified by the user by giving the values
C  of the number of pipes parallel and normal to the flow
C  direction, the number of passes on the gas side,
C  normalized pipe spacing and the pipe length in a single
C  gas pass.  The inlet thermodynamic states of the gas and
C  air are inputs to the subroutine.  The outlet
C  thermodynamic states of the air and flue gas is
C  calculated in the model.
C
C                            Jatila Ranasinghe, December,1985
C                                     revised, summer , 1987
C
C     Variable names:
C     ---------------
C
C     area   - Heat transfer area. (sq.ft)
C     Cp_    - Specific heat. (Btu/lbm F)
C     Derosn - Diameter of the dust particles. (in)
C     dP_    - Pressure drop. (psia,W.C)
C     Ex_    - Exergy. (KW)
C     Fcfhx  - Correction factor F.
C     foul   - Fouling in the heat exchanger.
C     h_     - Heat transfer coefficient. (Btu/sq.ft F)
C     H_     - Enthalpy. (Btu/lbm)
C     ID     - Inside tube diameter. (in)
C     IRR    - Irreversibility. (KW)
C     K_     - Thermal conductivity. (Btu/h ft R)
C     leakge - Percentage leakage.
C     length - Effective pipe length. (ft)
C     lmax   - Maximum allowable pipe length. (ft)
C     LMTD   - Log mean temperature differance.
C     MR_    - Mass flow rate. (lbm/s)
C     mu_    - Dynamic viscosity. (lbm/s ft)
C     noPg   - Number of passes on the gas side.
C     npn    - Number of pipes normal to the flow direction.
C     npp    - Number of pipes parallel to the flow
C              direction.
C     OD     - Outer tube diameter. (in)
C     P_     - Pressure. (psia,W.C)
```

```
C     Q       - Heat transfer. (Btu/s)
C     Ro_     - Density. (lbm/cu.ft)
C     rough   - Pipe roughness.
C     SnD     - Normalised pipe spacing normal to the flow
C               direction.
C     SpD     - Normalised pipe spacing parallel to the flow
C               direction.
C     T_      - Temperature. (F)
C     totalL  - Total pipe length per single air side pass.
C               (ft)
C     U       - Overall heat transfer coefficient. (Btu/sqft
C               F)
C     UA      - U*area
C     V_      - Velocity. (ft/s)
C
C
C     Subscripts:
C     -----------
C
C     a       - air
C     ab      - absolute value.
C     ave     - average value.
C     cnv     - due to convection.
C     ersn    - erosion.
C     fric    - due to friction.
C     g       - gas.
C     in      - at the inlet.
C     lk      - due to leakage.
C     loss    - total loss.
C     o       - at the outlet.
C     out     - outside.
C     pip     - inside the pipe.
C
C***********
C
$TITLE: 'Air-Gas heat exchanger'
      SUBROUTINE CERHXFD(Tain,Pain,MRain,yaCO2,yaH2O,yaO2,
     >yaN2,yaAr,yaCO,Tao,dPa,Pao,MRao,Taoreq,Tgin,Pgin,
     >MRgin,ygCO2,ygH2O,ygO2,ygN2,ygAr,ygCO,yoCO2,yoH2O,
     >yoO2,yoN2,yoAr,yoCO,dPg,Tgo,Pgo,MRgo,Pgoab,EXmat,
     >ROhx,psycer,tap,thick,ID,SnD,noU,npp,npn,type,noPg,
     >length,K,Derosn,rough,foul,ROash,leakge,Vmax,Versn,
     >Ca,Cdpa,Cdpg,IRRdpa,IRRdpg,IRRdt,IRRmi,IRRmr,
     >OBJIRR,OBJMI,OBJMR,OBJC)
C
C
      real MRa,MRg,ID,lmax,K,LMTD,length,MRpip,l2,l1,
     >     mua,mug,MUGAST,KGAST,ka,kg,IRR,noU,npp,npn,npnl,
     >     leakge,MRgin,MRain,MRalk,MRo,MRao,MRgo,noPg
      real IRRdpa,IRRdpg,IRRdt,IRRmi,IRRmr,Masshx

common/DS/Tds,Pds,ydsCO2,ydsH2O,ydsO2,ydsN2,ydsAr,ydsCO
```

```
      character*10 txt1,txt2
C
C
      PI=acos(-1.)
      OD=ID+2.*thick
C
C......Calculate the average mass flow rates.
C
      MRalk=MRain*leakge/100.
      MRao =MRain-MRalk
      MRgo =MRgin+MRalk
C
      MRa  =.5*(MRain+MRao)
      MRg  =.5*(MRgin+MRgo)
C
C......Calculate the total pipe length in one pass.
C
      totalL=length*noPg
C
C......Calculate the mass flow rate inside the pipe,
C......and total heat transfer area.
C
      MRpip=MRa/(npn*npp)
C
      if(noU.eq.0.)then
         area=OD/12.*PI*totalL*npn*npp
      else
         area=OD/12.*PI*totalL*npn*npp*noU*2.
      end if
C
C......Calculate enthalpy at inlets.
C
      Hain=HGAST(Tain,yaCO2,yaH2O,yaO2,yaN2,yaAr,yaCO)
      Hgin=HGAST(Tgin,ygCO2,ygH2O,ygO2,ygN2,ygAr,ygCO)
      Q1 =0.
C
C.......Iterate and find the exit air temperature.
C
  5   SpD  =SnD
      do 10 i=1,20
      Hao =HGAST(Tao,yaCO2,yaH2O,yaO2,yaN2,yaAr,yaCO)
      Qa  =MRa*(Hao-Hain)
      Hgo =Hgin - Qa/MRg
      Tgo =TGASH(Hgo,ygCO2,ygH2O,ygO2,ygN2,ygAr,ygCO)
C
C......Calculate the fluid properties.
C
      Tavea=(Tain+Tao)*.5
      Taveg=(Tgin+Tgo)*.5
      Twall=0.25*Taveg+0.75*Tavea
      Pavea=(2.*Pain-dPa)*.5
      Paveg=(2.*Pgin-dPg)*.5
```

```
      Tfilmg=(Twall+Taveg)*.5
C
C.....Calculate the pressure loss on the gas side.
C
      dPgl=PHXCF(MRg,ygCO2,ygH2O,ygO2,ygN2,ygAr,ygCO,Taveg,
      >     Tfilmg,Paveg,length,OD,type,SnD,SpD,npn,npp)
C
C......Pressure loss in bends.
C
      dPbg=0.
      dPg=dPgl*noPg+dPbg*(noPg-1.)
C
C.....Calculate the pressure loss inside the pipe.
C
      call PINPIP(MRpip,yaCO2,yaH2O,yaO2,yaN2,yaAr,yaCO,
     >     Tavea,Pavea,ID,rough,dPfric,fric)
C
      if(noU.eq.0.)then
         dPfric=dPfric*totalL
      else
        dPfric=dPfric*totalL*2.*noU
      end if
C
C......Pressure loss at entrance and exit.
C
      Roavea=ROGAS(Tavea,Pavea,yaCO2,yaH2O,yaO2,yaN2,
                   yaAr,yaCO)
      Vpip=MRpip/ROavea/PI/((ID/12./2.)**2)
      dPloss=(Vpip**2)/2.*ROavea*(0.78+1.)/32.174/144.
C
C......Total pressure loss in pipe.
C
      dPa=dPfric+dPloss
C
C......Calculate inside heat transfer coefficient.
C
      hincnv=HINCON(MRpip,yaCO2,yaH2O,yaO2,yaN2,yaAr,yaCO,
     >              Tavea,Pavea,ID,totalL)
      hin   =hincnv
C
C......Calculate the outside heat transfer coefficient.
C
      hout=HHXCF(MRg,ygCO2,ygH2O,ygO2,ygN2,ygAr,ygCO,Taveg,
     >     Twall,Paveg,length,OD,type,SnD,SpD,npn,npp,noU)
C
C......Calculate U and UA
C
      U=1./(1./((ID/OD)*hin)+1./hout+OD/(2.*K*12.)*
     >   alog(OD/ID)+foul)
      UA=U*area
```

```
C
C......Find the correction factor F.
C
      if(noPg.gt.1.)then
        Fcfhx=1.
      else
        Fcfhx=CORRHX(Tain,Tao,Tgin,Tgo,noU)
      end if
C
C......Calculate the LMTD,Cp and UAreq.
C
      dt1=Tgin-Tao
      dt2=Tgo-Tain
      LMTD=(dt1-dt2)/alog(dt1/dt2)
C
C.....Calculate the total heat transfer and enthalpy of air
C.....at outlet.
C
      Qua=UA*Fcfhx*LMTD/3600.
      Q=(Qua+Qa)/2.
      if(i.gt.1.)then
       Q2=(Q+Q1)/2.
      else
       Q2=Q
      end if
      Hao=Q2/MRa+Hain
C
C......Calculate the outlet temperature of air .
C
      Tao1=Tao
      Tao =TGASH(Hao,yaCO2,yaH2O,yaO2,yaN2,yaAr,yaCO)
C
C......Check for convergence.
C
      if(abs(Q1/Q-1.).LT.0.001)goto 40
      Q1=Q
   10 continue
C
  40  Q1=0.
      chktemp=abs(Tao-Taoreq)
      if(chktemp.lt.1.)goto 50
      if(chktemp.gt.10.)then
         if(Tao.gt.Taoreq)then
           SnD=SnD*1.02
         else
           SnD=SnD*.98
         end if
      else
         if(Tao.gt.Taoreq)then
           SnD=SnD*1.005
         else
           SnD=SnD*.995
```

```
            end if
         end if
C
         goto 5
C
C......Check the velocity for erosion.
C
   50    Pao=Pain-dPa
         Pgo=Pgin-dPg
         call VEROSN(Paveg,Taveg,MRg,length,OD,SnD,SpD,npn,
     >        Derosn,ROash,ygCO2,ygH2O,ygO2,ygN2,ygAr,ygCO,
     >        Vmax,Versn)
          if(Vmax.gt.Versn)write(*,*)'max velocity exceeded'
C
C......Make adjustments for leakage .
C
         call HXMIX(MRalk,yaCO2,yaH2O,yaO2,yaN2,yaAr,yaCO,
     >        Tao,Pao,MRgin,ygCO2,ygH2O,ygO2,ygN2,ygAr,ygCO,
     >        Tgo,MRgo,yoCO2,yoH2O,yoO2,yoN2,yoAr,yoCO,To)
C
C......Calculate the absolute temperature on gas side
C
         Pginab= 14.696 + .03613*Pgin
         Pgoab = 14.696 + .03613*Pgo
C
C.....calculate the mass of the heat exchanger
C
         Masshx=area*(thick/12.)*ROhx/35.314
C
C......calculate the irreversibility components
C
         if(Pgoab.lt.1.)then
          write(9,*)'gas side pres too low'
          OBJMI=1.e23
          OBJMR=1.e23
          OBJIRR=1.e23
          goto 111
         end if
         if(Pao.lt.1.)then
          write(9,*)'air side pres too low'
          OBJMI=1.E23
          OBJMR=1.E23
          OBJIRR=1.E23
          goto 111
         END IF
         Sain=SGASTP(Tain,Pain,yaCO2,yaH2O,yaO2,yaN2,yaAr,yaCO)
         Sao =SGASTP(Tao,Pao,yaCO2,yaH2O,yaO2,yaN2,yaAr,yaCO)
         Sgin=SGASTP(Tgin,Pginab,ygCO2,ygH2O,ygO2,ygN2,
     >             ygAr,ygCO)
         Sgo =SGASTP(To,Pgoab,yoCO2,yoH2O,yoO2,yoN2,yoAr,yoCO)
         Saodpa=SGASTP(Tain,Pao,yaCO2,yaH2O,yaO2,yaN2,
     >               yaAr,yaCO)
```

```
      Sgodpg=SGASTP(Tgin,Pgoab,ygCO2,ygH2O,ygO2,ygN2,
     >              ygAr,ygCO)
C
      Exain=(Hain-(Tds+459.67)*Sain)*MRain*1.0552
      Exao =(Hao-(Tds+459.67)*Sao)*MRao*1.0552
      Exgin=(Hgin-(Tds+459.67)*Sgin)*MRgin*1.0552
      Exgo =(Hgo-(Tds+459.67)*Sgo)*MRgo*1.0552
C
      IRR  = Exain+Exgin-Exao-Exgo
      IRRdpa=(Tds+459.67)*(Saodpa*MRao-Sain*MRain)*1.0552
      IRRdpg=(Tds+459.67)*(Sgodpg*MRgo-Sgin*MRgin)*1.0552
      IRRdt=IRR-IRRdpa-IRRdpg
      IRRmi=EXmat*Masshx/psycer/tap
      IRRmr=EXmat*Masshx/tap
C
C.....Calculate the objective functions
C
      OBJIRR=IRRdpa+IRRdpg
      OBJMI =IRRdpa+IRRdpg+IRRmi
      OBJMR =IRRdpa+IRRdpg+IRRmr
      OBJC  =Ca*(area**0.6)/tap + Cdpa*IRRdpa + Cdpg*IRRdpg
C
 111  RETURN
      END
```

APPENDIX C

PROCESS INTEGRATION SUBROUTINE

```
      SUBROUTINE PIHPHE(TE,TC,QE,ITER,TPINCH,REDIRR)
*************************
*  This subroutine calculates the objective function
*  'REDIRR' and the pinch temperature 'TPINCH', which is
*  sometimes used to specify constraints, for process
*  integrated heat pumps and heat engines.  The input data
*  has to be provided by the user.  The irreversibility with
*  no integration 'FIRR',which is needed as an input data,
*  is obtained by using this subroutine.  This is done by
*  setting QE=0 and FIRR=1, and the value of REDIRR is then
*  equal to the actual FIRR.  The pinch temperature before
*  integration is also obtained in the same run.  The other
*  inputs to the subroutine are as follows:
*  TE     = evaporator temperature
*  TC     = condenser temperature
*  QE     = evaporator load
*  ITER   = iteration number (set in the optimization code)
*  NS     = number of hot and cold streams
*  CHKHP  = Equal to 1. if hp is integrated.  Else = 0.
*  CHKHE  = Equal to 1. if he is integrated.  Else = 0.
*  DTMIN  = Minimum approach temperature of process streams
*  TO     = Dead state temperature
*  THU    = Temperature of hot utility
*  NUMS   = stream number
*  HORC   = For cold streams =1., for hot streams = -1.
*  TU     = High temperature of a stream
*  TL     = Low temperature of a stream
*  C      = Capacity rate of a stream (mcp)
*************************

      INTEGER NUMS(50),NS
      REAL HORC(50),TU(50),TL(50),C(50),TI(100),QTI(100),
     >     QF(100),TIIRR(50)
      ITER=ITER+1
C
C.....OPEN INPUT FILES, AND READ INPUT DATA
C
      IF(ITER.EQ.1)THEN
      WRITE(*,*)'FILE NAME FOR STREAM DATA'
      WRITE(*,*)
      OPEN(6,FILE=' ',STATUS='OLD')
      READ(6,*)NS
      READ(6,*)CHKHP,CHKHE
      READ(6,*)FIRR
      READ(6,*)DTMIN,TO,THU
```

```
      DO 10 I=1,NS
      READ(6,*)NUMS(I),HORC(I),TU(I),TL(I),C(I)
      IF(HORC(I).EQ.1.)THEN
        TU(I)=TU(I)+DTMIN
        TL(I)=TL(I)+DTMIN
      END IF
 10   CONTINUE
      END IF
C
      IF(CHKHP.EQ.1.)THEN
        TEVAP=TE
        TCOND=TC
        QEVAP=QE
      END IF
      IF(CHKHE.EQ.1.)THEN
        TBOIL=TE
        TCOOL=TC
        QBOIL=QE
      END IF
C
C.....ADJUST THE TEMPERATURES OF THE COLD STREAMS AND
C.....DETERMINE THE MAXIMUM AND MINIMUM INTERVAL TEMPS.
C
      TMAX=-1.E20
      TMIN=1.E20
      DO 50 I=1,NS
       IF(TU(I).GT.TMAX)TMAX=TU(I)
       IF(TL(I).GT.TMAX)TMAX=TL(I)
       IF(TU(I).LT.TMIN)TMIN=TU(I)
       IF(TL(I).LT.TMIN)TMIN=TL(I)
 50   CONTINUE
C
      TCU=TO
      IF((TMAX.GT.THU).OR.(TMIN.LT.TCU)THEN
          WRITE(*,*)'UTILITY TEMP NOT SUFFICIENT'
      END IF
      TMIN=TCU
      TMAX=THU
C
C.....FIX THE INTERVAL TEMPEARTURES
C
      J=0
      CHKOLD=1.E20
 60   CHK=-1.E20
      J=J+1
      DO 70 I=1,NS
       IF((TU(I).GT.CHK).AND.(TU(I).LT.CHKOLD))CHK=TU(I)
       IF((TL(I).GT.CHK).AND.(TL(I).LT.CHKOLD))CHK=TL(I)
 70   CONTINUE
      IF(CHKHP.EQ.1.)THEN
       IF((TEVAP.GT.CHK).AND.(TEVAP.LT.CHKOLD))CHK=TEVAP
       IF((TCOND.GT.CHK).AND.(TCOND.LT.CHKOLD))CHK=TCOND
```

```
      END IF
      IF(CHKHE.EQ.1.)THEN
       IF((TBOIL.GT.CHK).AND.(TBOIL.LT.CHKOLD))CHK=TBOIL
       IF((TCOOL.GT.CHK).AND.(TCOOL.LT.CHKOLD))CHK=TCOOL
      END IF
       IF((THU.GT.CHK).AND.(THU.LT.CHKOLD))CHK=THU
       IF((TCU.GT.CHK).AND.(TCU.LT.CHKOLD))CHK=TCU
      TI(J)=CHK
      CHKOLD=CHK
      IF(CHK.EQ.TMIN)THEN
       INTVLS=J-1
       GOTO 80
      ELSE
       GOTO 60
      END IF
 80   CONTINUE
C
C.....CALCULATE COPHP AND CONDENSER LOAD OF HEAT PUMP.
C.....THIS BLOCK IS SPECIFIC FOR THE EXAMPLE IN THE THESIS.
C
      IF(CHKHP.EQ.1.)THEN
       COPHP=1./(.0723+.004609*(TCOND-(TEVAP-DTMIN)))
       QCOND=QEVAP/(1.-1./COPHP)
      END IF
      IF(CHKHE.EQ.1.)THEN
       EFFHE=0.5*(1.-TCOOL/TBOIL)
       QCOOL=QBOIL(1.-EFFHE)
      END IF
C
C.....ENERGY BALANCE IN TEMPERATURE INTERVALS
C
      DO 90 J=1,INTVLS
      SUM=0.
      DO 100 I=1,NS
       IF((TU(I).GE.TI(J)).AND.(TL(I).LT.TI(J)))THEN
       SUM=SUM+HORC(I)*C(I)*(TI(J)-TI(J+1))
       END IF
       IF((TU(I).EQ.TI(J)).AND.(TL(I).EQ.TI(J)))THEN
       SUM=SUM+HORC(I)*C(I)
       END IF
 100  CONTINUE
C
      IF(CHKHP.EQ.1.)THEN
      IF(TEVAP.EQ.TI(J+1))SUM=SUM+QEVAP
      IF(TCOND.EQ.TI(J))SUM=SUM-QCOND
      END IF
      IF(CHKHE.EQ.1.)THEN
      IF(TCOOL.EQ.TI(J))SUM=SUM-QCOOL
      IF(TBOIL.EQ.TI(J+1))SUM=SUM+QBOIL
      END IF
C
      QTI(J)=SUM
```

```
 90   CONTINUE
C
C.....ENERGY FLOW WITH OUTILITY
C
      HU=1.E20
      QF(1)=0.
      DO 110 I=2,INTVLS+1
      QF(I)=QF(I-1)-QTI(I-1)
      IF(QF(I).LT.HU)THEN
       HU=QF(I)
       TPINCH=TI(I)
       BPINCH=I
      END IF
 110  CONTINUE
      HU=ABS(HU)
C
C.....ENERGY FLOW WITH UTILITY ADDED
C
      QF(1)=HU
      DO 120 I=2,INTVLS+1
      QF(I)=QF(I-1)-QTI(I-1)
 120  CONTINUE
      CU=QF(INTVLS+1)
C
C.....CALCULATE TEMPERATURE INTERVAL IRREVERSIBILITIES
C
      TOTIRR=0.
      DO 130 J=1,INTVLS
      SUM=0.
      DO 140 I=1,NS
       IF((TU(I).GE.TI(J)).AND.(TL(I).LT.TI(J)))THEN
       SUM=SUM+HORC(I)*C(I)
       END IF
 140  CONTINUE
       TIIRR(J)=ABS(T0*(QF(J+1)/TI(J+1)-QF(J)/TI(J)+SUM*
     >          ALOG(TI(J)/TI(J+1))))
C
      IF(CHKHP.EQ.1.)THEN
      IF(TCOND.EQ.TI(J))THEN
        TIIRR(J)=ABS(T0*(QF(J+1)/TI(J+1)-QF(J)/TI(J)
     >           -QCOND/TCOND+SUM*ALOG(TI(J)/TI(J+1))))
      END IF
      IF(TEVAP.EQ.TI(J+1))THEN
        TIIRR(J)=ABS(T0*(QF(J+1)/TI(J+1)+QEVAP/TEVAP
     >           -QF(J)/TI(J)+SUM*ALOG(TI(J)/TI(J+1))))
      END IF
      END IF
C
      IF(CHKHE.EQ.1.)THEN
      IF(TCOOL.EQ.TI(J))THEN
       TIIRR(J)=ABS(T0*(QF(J+1)/TI(J+1)-QF(J)/TI(J)
     >              -QCOOL/TCOOL+SUM*ALOG(TI(J)/TI(J+1))))
```

```
      END IF
      IF(TBOIL.EQ.TI(J+1))THEN
         TIIRR(J)=ABS(T0*(QF(J+1)/TI(J+1)+QBOIL/TBOIL
     >               -QF(J)/TI(J)+SUM*ALOG(TI(J)/TI(J+1))))
      END IF
      END IF
      TOTIRR=TOTIRR+TIIRR(J)
 130  CONTINUE
C
C.....CALCULATE THE HEAT ENGINE AND HEAT PUMP
C.....IRREVERSIBILITIES
C
      IF(CHKHP.EQ.1.)THEN
       WHP=QCOND-QEVAP
       PSYCMP=1.-T0*((1.-COPHP)/(TEVAP-DTMIN)+COPHP/TCOND)
       CMPIRR=WHP*(1.-PSYCMP)
       TOTIRR=TOTIRR+CMPIRR
      END IF
      IF(CHKHE.EQ.1.)THEN
       WHE=QBOIL-QCOOL
       TURBIRR=T0*QBOIL*(1./TCOOL-1./TBOIL-EFFHE/TCOOL)
       TOTIRR=TOTIRR+TURBIRR
      END IF
C
      REDIRR=TOTIRR/FIRR
C
      RETURN
      END
```