# ENERGY INTEGRATION OF INDUSTRIAL PROCESSES BASED ON A GRAPHIC REPRESENTATION OF EXERGY FACTORS.

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The energy integration of industrial processes is becoming increasingly more effective thanks to new methodological developments such as pinch technology. The original method centers primarily on a consideration of the heat transfer exergy losses represented by the area between the composite curves on a Carnot factor versus heat rate diagram. The present paper proposes an extension of this approach to account for pressure drop exergy losses as well as for the gray exergy associated with the fabrication of the heat exchangers. It is proposed to graphically represent on the same Carnot factor versus heat rate diagram the above mentioned losses as well as other high exergy inputs and outputs linked, for example, to the introduction of heat pumps and cogeneration units. Such an extended exergy synthesis results in an improved and more coherent exergy balance for comparing energy recovery schemes. The proposed approach is suitable for future extension to include pollution and resource scarcity factors.

NOMENCLATURE

| | |
|---|---|
| $A$ | heat exchanger area |
| $Ex_g$ | gray exergy |
| $\dot{E}$ | electric power |
| $\dot{E}q$ | rate of exergy due to heat transfer |
| $\dot{H}$ | heat rate |
| $L$ | exergy losses |
| $\dot{L}t$ | rate of exergy losses due to heat transfer |
| $\dot{L}r$ | rate of exergy losses due to dissipation |
| $\dot{L}f$ | rate of exergy losses due to fabrication |
| $\dot{L}tot$ | rate of total exergy losses |
| $\dot{Q}$ | heat rate or duty |
| $T$ | absolute temperature |
| $t$ | lifetime of the components |

Greek letters

| | |
|---|---|
| $\delta Q_{\beta}^{\alpha}$ | energy exchanged betwen $\alpha$ and $\beta$ |
| $\delta S^{i}$ | internal entropy creation |
| $\delta R$ | dissipation energy |
| $\Delta T_{HP}$ | temperature difference in the heat pump |
| $\Delta T_{min}$ | minimum temperature difference |
| $\Delta\theta_{min}$ | minimum Carnot factor difference |
| $\Delta\theta g_{min}$ | minimum global Carnot factor difference |
| $\theta$ | Carnot factor |
| $\theta r$ | dissipation Carnot factor |
| $\theta f$ | fabrication Carnot factor |

Subscripts

| | |
|---|---|
| a | atmosphere |
| h | hot stream |
| c | cold stream |
| k | heat pump compressor |
| p | pumps (ventilators, compressors) |
| st | steam turbine |
| cu | cold utility |
| $\alpha$ | substance $\alpha$ |
| $\beta$ | substance $\beta$ |

Superscripts

| | |
|---|---|
| + | given to the system |
| - | given by the system |

## 1. INTRODUCTION

The increasing concern for the environmental impacts of human activities has stimulated the development of new methods for the analysis of industrial processes and the implementation of energy conservation measures. One particularly powerful method is the so-called pinch technology for the energy integration of thermal processes, which has matured during the last ten years with major contributions from the research group at the University of Manchester with Linnhoff et al. (1982). One of the objectives of this powerful approach is to prioritize internal heat exchanges between process or site streams in order to make a rapid determination of realistic energy targets for the hot and cold utilities of the process. The method also provides clear guidelines for the placement of power units like cogeneration or heat pump units. A synthesis of both the energy needs and energy recovery opportunities of the process or of the site is represented in a temperature versus heat rate diagram (hot and cold composites, respectively). The most constrained region for internal heat transfer corresponds to the minimum vertical distance betwen the two composite curves ($\Delta T_{min}$) and is refered to as the pinch temperature level. The optimal value of $\Delta T_{min}$ is usually chosen by a trade-off between investment and operational costs. The ultimate goal of the analysis is to design the most energy efficient heat exchanger network.

## 1. LIMITATIONS OF THE ORIGINAL PINCH METHOD

In the initial pinch technology approach, the thermodynamic concepts of the Second Law appear essentially with regards to the heat transfer driving force and direction. Exergy losses resulting from the various heat exchanges between process streams can be represented by the area delimited by the composite curves if drawn on a Carnot factor versus heat rate diagram.

In most analyses published so far, the optimal value of $\Delta T_{min}$ is based on an economic criterion (e.g., yearly cost) with fixed values for the film heat transfer coefficient of each stream (e.g., supertargeting). Costs are primarily proportional to the heat transfer area of the heat exchangers; exergy losses due to pressure drops during the operation of the process itself or incurred earlier during the fabrication of the components are not considered. Except for the beneficial effects resulting from the reduction of the utility requirements, there is also no obvious path for relating the losses considered in a standard pinch technology approach with environmental factors. With regards to friction losses, a paper by Polley et al. (1990) takes into consideration the pressure drops in the retrofit of heat exchanger networks by using a relationship between heat exchange and pressure drop. The results however are based on arbitrarily allowable or existing values and are not readily based on exergy considerations.

Although process integration analyses often lead to the integration of heat pump or cogeneration units, the classical representation does not graphically highlight the resulting energy balance of electric power. For instance, a comparison of the composite curves before and after the integration of a heat pump graphically illustrates the increased internal heat transfer area with corresponding decreases in the utility loads; but electricity is not represented so that the existing indicators can be misleading when trying to make effective efficiency improvements (see figure 1). Even when considering only heat transfer losses, the original pinch method recommends for the particular case of multiple pinch points, the use of the same value for the pinch temperature difference at all pinch points without consideration for their corresponding temperature levels. Thermodynamics says however that for the same temperature drop, the entropy creation is considerably larger at low temperatures than at high temperatures. This effect is fully taken into account for when exergy losses instead of simple temperature difference criteria are considered, because the exergy losses, as shown by Eq. (1), are directly proportional to the entropy created (Borel, 1987).

$$L = T_a \, \delta S^i \qquad (1)$$

where $\delta S^i \equiv$ entropy creation due to heat transfer with a temperature drop

$$\delta S^i = \left(\frac{1}{T_\beta} - \frac{1}{T_\alpha}\right)\delta Q_\beta^\alpha \qquad (2)$$

and $\delta S^i \equiv$ entropy creation due to energy dissipation resulting from friction, pressure drop losses (see figure 2).

$$\delta S^i = \frac{\delta R}{T} \qquad (3)$$

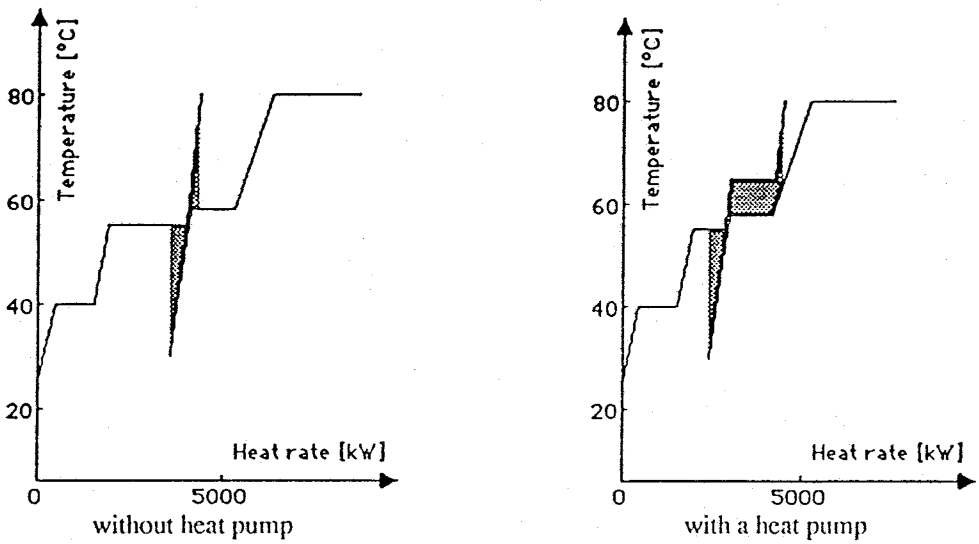


Figure 1: Composite curves obtained with the original pinch technology method; the electric power of the heat pump compressor is not shown (Favrat and Staine, 1991).

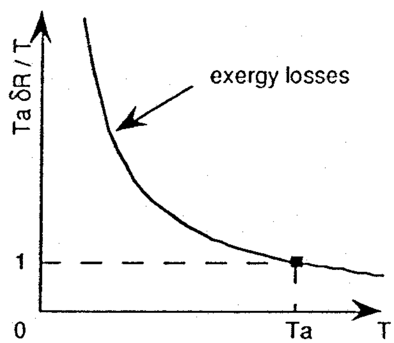


Figure 2: Variation of the exergy losses due to pressure drop versus the temperature associated with the corresponding energy dissipation.

## 3. THERMODYNAMIC OPTIMISATION OF A HEAT EXCHANGER

When designing heat exchangers, the objective, energetically speaking, is to minimize thermodynamic losses. For a fixed heat transfer area, this requires balancing competing heat transfer and fluid flow irreversibilities (Bejan, 1987). Further minimization of the cumulated operational losses can be achieved by increasing the heat transfer area which, at the limit, results in an infinite heat transfer area (see figure 3). Such a result is, of course, not feasible and even a large finite area may not be practical for economic reasons .

must be altered to also account for the exergy expended during the manufacturing process of the heat transfer equipment. This exergy called the gray exergy is proportional to the heat transfer area (Aceves-Saborio and Reistad, 1990) and includes the raw material exergy plus all the exergy delivered during the entire chain of processes associated with the fabrication of the exchanger minus, of course the exergy of potentially valuable wastes such as, for example, that of aluminium recycled chips. The fabrication process begins with the extraction of the raw material and continues with its transformation, machining and assembly. For most processes, exergy loss rates are considered so that the gray exergy is divided by the expected life time or the expected useful time which ever come first in order to be able to use it on a rate basis. The thermodynamic balance of competing heat transfer, dissipation and component fabrication losses results in an exergetically optimised heat exchanger with a finite heat transfer area (see figure 4).

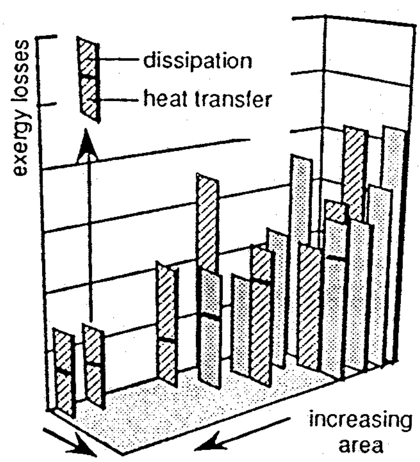


Figure 3 : Thermodynamic optimisation structure of a heat exchanger

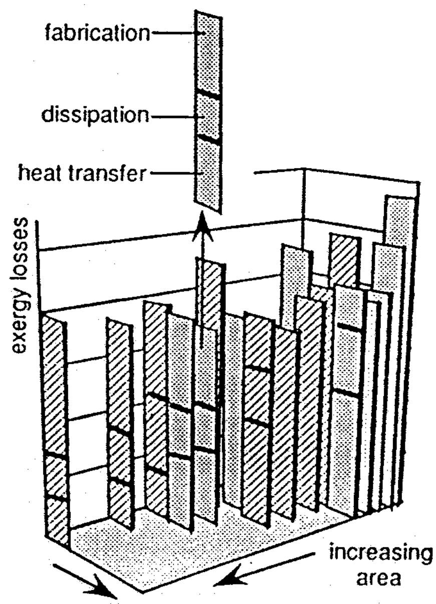


Figure 4 : Evolution of the exergetic losses of a heat exchanger including the gray exergy

However, even this balance is incomplete since it lacks the ecological considerations which may be pertinent not only during the use of the heat exchanger itself but during its fabrication as well. Effects on the environment due to pollution or relations accounting for the degree of future scarcity of the resources used can, to a great extent, be related to the exergetic losses as proposed by some recent papers (von Spakovsky et al., 1991). In the latter approach, pollution and scarcity factors (lower or equal to one) corresponding to each pollutant or resource are introduced as denominators to artificially increase the exergetic losses of a particular process. Although these parameters are not specifically introduced in the extended pinch method presented here, the present approach does pave the way for a later introduction of such considerations.

The graphs in figures 3 and 4 show the results of search type of thermodynamic optimisation for a shell and tube heat exchanger, where air (hot stream) circulates in the shell and water (cold stream) in the tubes. As mentioned above, the gray exergy is essentially proportional to the heat exchange area (A). For a shell and tube heat exchanger, this latter value corresponds approximately to 2500 MJ/m$^2$ (Staine, 1992). The lifetime chosen in this analysis is 30 000 h.

$$\dot{L}f = \frac{A \, . \, Ex_g}{t} \tag{4}$$

## 4. REPRESENTATION OF THE EXERGY LOSSES FOR INTERNAL HEAT EXCHANGERS

The representation of the exergy losses for internal heat exchangers described below is valid for the heat exchangers transferring heat between process streams. The main idea of this representation is the following: since the exergy losses due to heat transfer are represented as an area in a Carnot factor versus heat rate diagram, all other exergy losses will be represented in a similar way. For a given heat exchanger (with known duty, area, inlet and oulet temperatures and pressure drop for the hot and cold streams), the exergy losses due to heat transfer ($\dot{L}t$), dissipation ($\dot{L}r_h$ and $\dot{L}r_c$) and equipment fabrication ($\dot{L}f_h$ and $\dot{L}f_c$) can easily be calculated. The values for all the losses except $\dot{L}t$ are represented as additional areas immediately above and below, respectively, of the heat transfer loss area for the hot and cold streams, respectively of the composites. This is done in order to keep track of the location of the additional losses. The area corresponding to these losses spreads horizontally over the length corresponding to the heat rate ($\dot{Q}$) of the specific heat exchanger considered. The losses $\dot{L}t$ due to heat transfer is the dark gray area between the two curves $\theta_h$ and $\theta_c$ (see figure 5).

Two new curves $\theta r_h$ and $\theta r_c$, given by Eqs. (5) and (6) are used to construct the area which represents $\dot{L}r$. Normally associated with each hot and cold stream, $\dot{L}r$ corresponds then to the sum of the two areas $\dot{L}r_h$ (the light gray area between $\theta r_h$ and $\theta_h$) and $\dot{L}r_c$ (the light gray area between $\theta_c$ and $\theta r_c$), immediately above and below the area corresponding to $\dot{L}t$.

$$\theta r_h = \theta_h + \frac{\dot{L}r_h}{\dot{Q}} \tag{5}$$

$$\theta r_c = \theta_c - \frac{\dot{L}r_c}{\dot{Q}} \tag{6}$$

In a similar way, the fabrication losses $\dot{L}f$ can be distributed equally between the hot and the cold streams, the corresponding surfaces being added to define two new curves $\theta f_h$ and $\theta f_c$ given by Eqs. (7) and (8).

$$\theta f_h = \theta r_h + \frac{\dot{L}f_h}{\dot{Q}} \tag{7}$$

$$\theta f_c = \theta r_c - \frac{\dot{L}f_c}{\dot{Q}} \tag{8}$$

$\dot{L}f$ then, is the sum of the two areas $\dot{L}f_h$ (the diagonally hatched area between $\theta f_h$ and $\theta r_h$) and $\dot{L}f_c$ ( the diagonally hatched area between $\theta r_c$ and $\theta f_c$), above and below , respectively, the areas for $\dot{L}r_h$ and $\dot{L}r_c$.

In this way, all the exergy losses associated with a heat exchanger are brought together on the same diagram in a set of cumulated areas between $\theta f_h$ and $\theta f_c$, so that the relationship between each type of irreversibility is

clearly visible ( see figure 5 - the portions of the curves corresponding to the internal heat exchangers).
A heat exchanger is characterized not only by the parameter $\Delta T_{min}$ or $\Delta\theta_{min}$ relative to heat transfer alone, but also by the parameter $\Delta\theta g_{min}$, given by Eq. (9), relative to heat transfer, pressure drop and fabrication. It is proposed that the design of the heat exchanger network should preferably be based on this factor.

$$\Delta\theta g_{min} = \theta f_h - \theta f_c \tag{9}$$

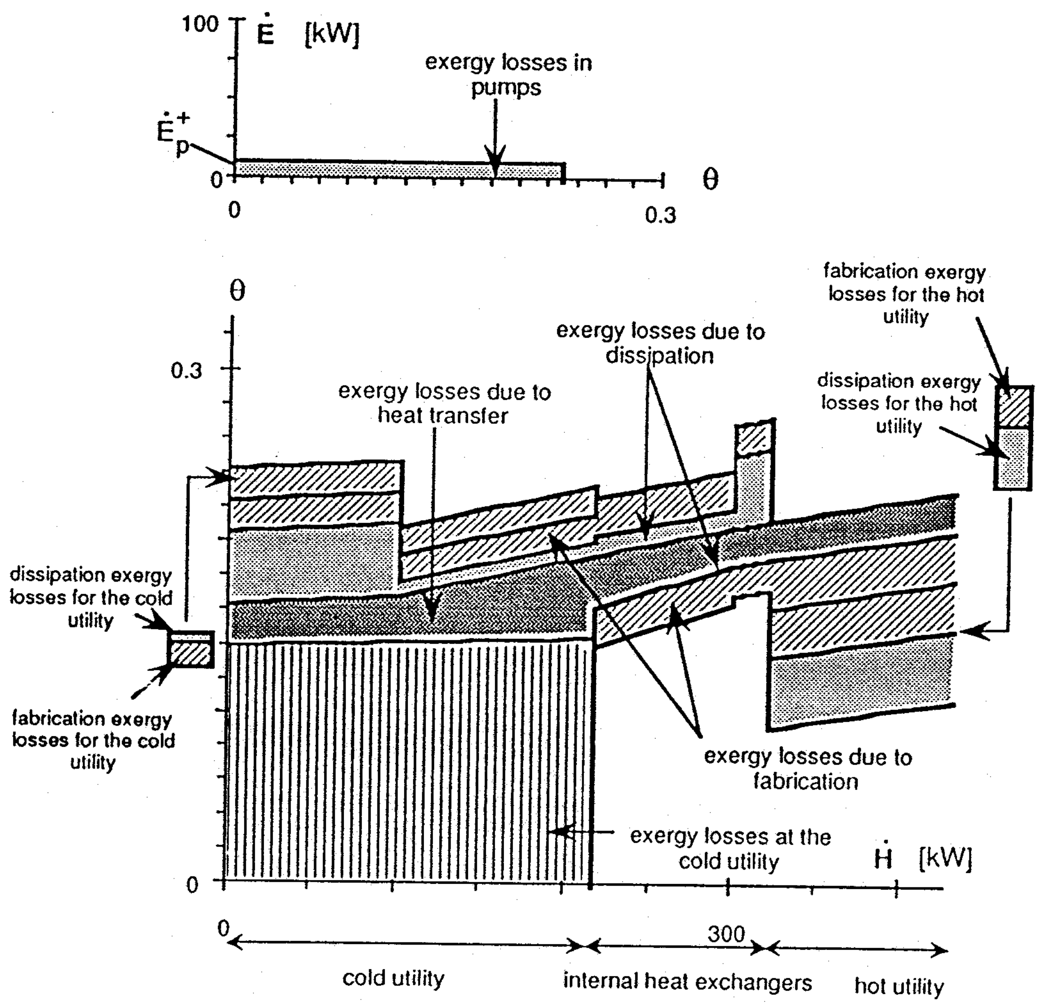


Figure 5 : Extended representation of the exergy losses of a simple process

## 5. REPRESENTATION OF THE EXERGY LOSSES FOR THE EXTERNAL HEAT EXCHANGERS OF THE COLD UTILITY

In this zone, characterized by an excess of energy, all the exergy of the hot streams is considered to be lost in so far as this exchanger does not allow any recovery of possible exergy benefits. These losses expressed by Eq. (10) are represented by the area (dark gray and vertically hatched areas) under the corresponding hot streams down to the horizontal axis where the Carnot factor is zero (T=Ta).
The dissipation and fabrication exergy losses $\dot{L}r_h$ and $\dot{L}f_h$ relevant to the hot streams remain unchanged from

the previous representation, but the areas $\dot{L}r_c$ and $\dot{L}f_c$ related to the utility streams are not drawn below but just above those of the hot streams (see figure 5 - the portion of the curves corresponding to the cold utility). Note that the dissipation losses $\dot{L}r_c$ are too small to be clearly visible in the figure. This way of representing the utility losses eliminates the superposition of surfaces which would negate any interest in this kind of area representation.

$$\dot{L}_{cu} = \dot{E}q_h^- = \int\left(1 - \frac{T_a}{T}\right)\delta\dot{Q}_h^- \qquad (10)$$

## 6. REPRESENTATION OF THE EXERGY LOSSES FOR THE EXTERNAL HEAT EXCHANGERS OF THE HOT UTILITY

In this zone, characterized by a lack of energy, the hot utility can be combustion gases from a fuel boiler, steam, heat release from a cogeneration unit, electricity, etc. If high quality energy (exergy) such as electricity is used, the corresponding Carnot factor is equal to one and the area between $\theta = 1$ and the top part of the cold composite represents the exergy losses due to heat transfer. As before, the previous representation for the exergy losses $\dot{L}r_c$ and $\dot{L}f_c$ of the cold streams is unchanged. By symmetry with what was done on the cold utility side, the areas $\dot{L}r_h$ and $\dot{L}f_h$ linked to the hot streams are drawn just below the corresponding cold stream areas (see figure 5 - the portion of the curves corresponding to the hot utility).

## 7. CHOICE OF THE OPTIMUM VALUE OF $\Delta T_{MIN}$ BASED ON A TOTAL EXERGY BALANCE

The sum of the exergy losses $\dot{L}tot$ (heat transfer, dissipation, component fabrication, pumps and turbines, minus the exergy gains) depend on the pinch value $\Delta T_{min}$. An example of the total loss trend is shown in figure 6 where the optimum corresponds to a pinch value of 5°C.

## 8. EXERGY BALANCE INCLUDING ELECTRICITY NEEDS AND COGENERATION PRODUCTION.

In a site analysis, it is often valuable to not only make a synthesis of the heat balance but also of the electricity balance. As is known, such considerations are of primary importance when discussing cogeneration. An additional area representation is proposed whereby one axis gives the absolute consumption of electricity and the other the Carnot factor $\theta$ such that the loss area of the specific component under consideration is represented between the two axes. The $\dot{E}$ versus $\theta$ diagram is placed immediately above the $\theta$ versus $\dot{H}$ diagram keeping the same scale for coherence. Let us look at some of the key components of industrial systems.

- Pumps, ventilators and compressors (referred to generally as pumps below) :

The pressure drop of each hot and cold stream in the heat exchangers must be compensated by some kind of pumping power generally given by a pump (see figure 7). Given the electric power on a pump $\dot{E}_p^+$ and its exergy losses $\dot{L}_p$, the length of the rectangle corresponding to these losses is equal to $\dot{L}_p / \dot{E}_p^+$ (see figure 5). For most processes, the electric power and rate of exergy losses of pumps will ultimately be grouped together to have only one global representative area. On the other hand, for a process where the integration of a heat pump is planned, the representation of the electric power needed for the compressor ($\dot{E}_k^+$) and the exergy losses in the compressor ($\dot{L}_k$) will be separated from the electricity rate for pumps to highlight it (see figure 9c, further on in the discussions).

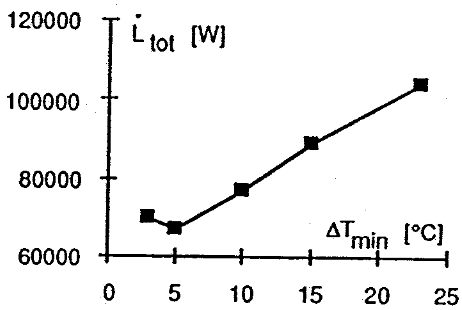


Figure 6 : Variations of the total exergy losses versus $\Delta T_{min}$ (example 1)

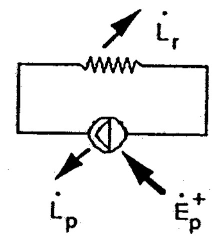


Figure 7 : Pump associated with each stream

- Turbines:

A turbine in a process either produces electricity or direct shaft power with the benefit of not having an equivalent amount of electrical consumption. This electricity or shaft power is a net gain in the exergy balance of the process minus of course, the exergy losses in the turbine itself. The losses are represented by the area $\dot{E}_{st}^- \times \dot{L}_{st} / \dot{E}_{st}^-$ in the $\dot{E}$ versus $\theta$ diagram (see figure 11, further on in the discussions). For new power components (pumps, turbines, etc.), the exergy losses linked to their fabrication can also be represented on the $\dot{E}$ versus $\theta$ diagram.

## 9. EXAMPLE 1 : ANALYSIS OF A SIMPLE PROCESS

In this example, a simple process is analysed with the proposed method. The optimum value of $\Delta T_{min}$, based on the minimum of the total exergy losses, is found by calculating the curve in figure 6. As mentioned above $\dot{L}tot$ has a minimum value for a $\Delta T_{min}$ of 5°C. Figures 8a and 8b show two representations of the exergy losses for two values of $\Delta T_{min}$. Even for small changes of the pinch temperature difference, a comparison of the loss areas linked to each type of irreversibility and a comparison of the required electric power are very useful and straight forward.

## 10. EXAMPLE 2 : INTEGRATION OF A HEAT PUMP

In this example, the fluids considered are air and water for the hot and the cold streams, respectively. The refrigerant chosen in the heat pump cycle is R123. The evaporator of the heat pump uses some of the exhaust air of the process and the condenser is used on the hot utility side. Figure 9a shows the classic composite curves, and figure 9b shows only the exergy losses due to heat transfer as represented in the original pinch method.

A representation of all the exergy losses and the exergy balance of the process is given in figure 9c where the different types of irreversibility are clearly visible. The exception is the area corresponding to the dissipation exergy losses of the cold streams which is too small to be seen. All the heat exchangers in this process have been optimised by minimizing the sum of the three types of exergy losses as explained before. It is assumed that each exchanger is a shell and tube heat exchanger for which the calculation of the heat exchange area, the film heat transfer coefficients and the pressure drop for both streams are based on the respective fluid velocities and on the geometry. The curves in figure 9c correspond to an integration of a heat pump with a temperature difference at the evaporator and at the condenser ($\Delta T_{HP}$) of 10°C. For this case, the exergy losses are higher than those with a $\Delta T_{HP}$ of 5°C (see figure 10).

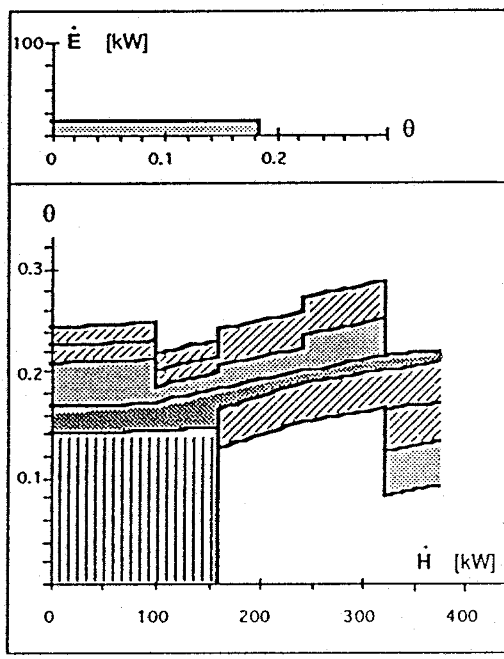


Figure 8a : Representation of the exergy losses with $\Delta T_{min}$ = 5°C

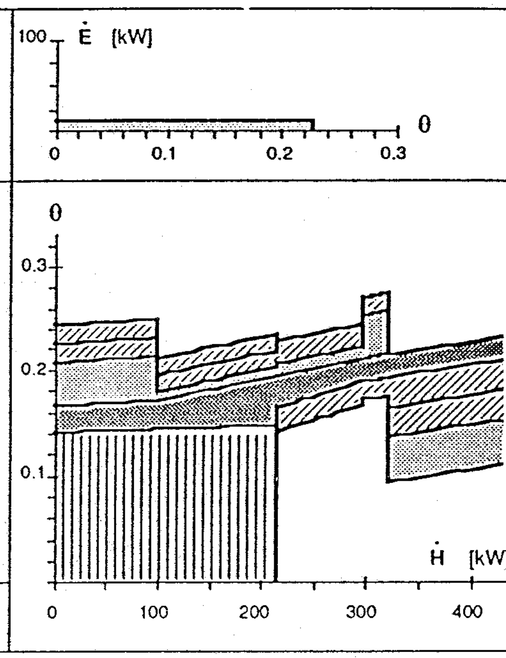


Figure 8b : Representation of the exergy losses with $\Delta T_{min}$ = 10°C

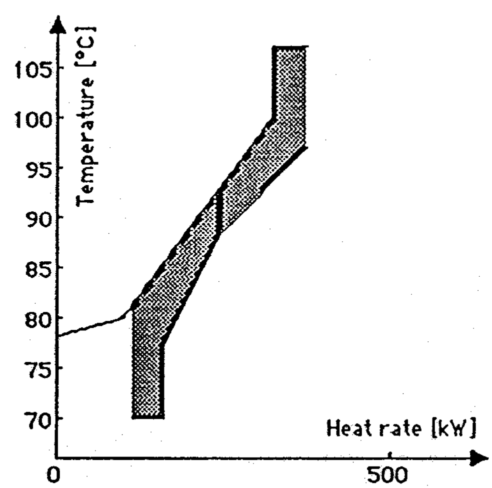


Figure 9a : Composite curves of the process with a heat pump ($\Delta T_{HP}$ = 10°C) (Favrat and Staine, 1991).

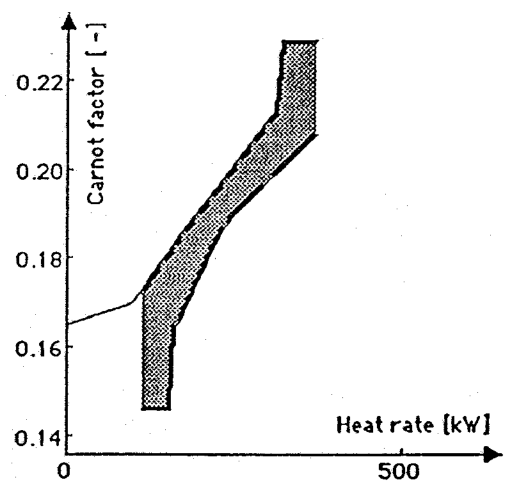


Figure 9b: Representation of the exergy losses due to heat transfer of the process with a heat pump ($\Delta T_{HP}$ = 10°C) (Favrat and Staine, 1991).

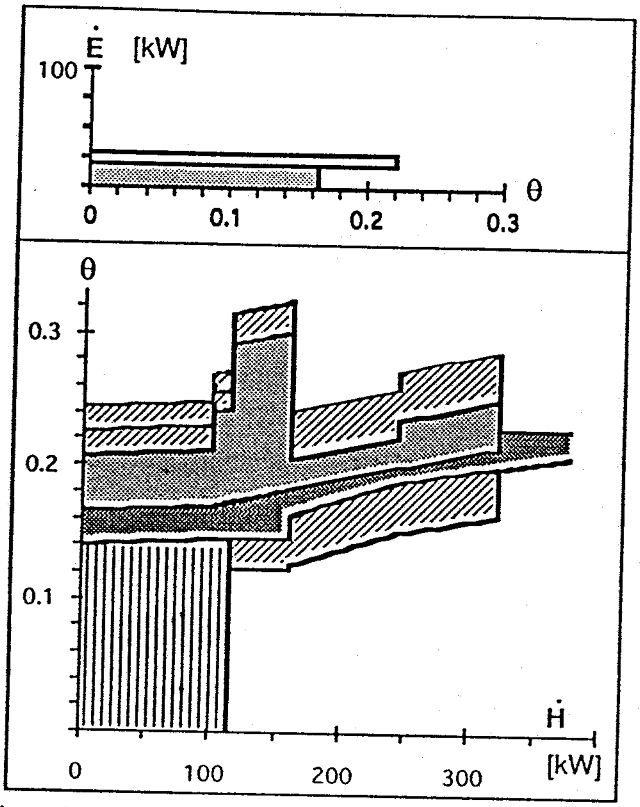


Figure 9c : Extended representation of the exergy losses of a process including a heat pump ($\Delta T_{HP} = 10°C$)

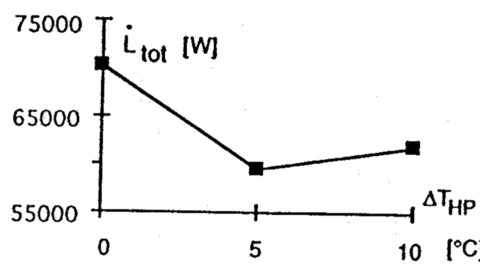


Figure 10 : Evolution of the total exergy losses versus $\Delta T_{HP}$ ($\Delta T_{HP}$ =0°C means without heat pump).

## 11. EXAMPLE 3 : INTEGRATION OF A STEAM TURBINE

In this example, we consider only the recuperator (steam generator) and the steam turbine of a combined plant. The exhaust gases from the gas turbine are cooled to produce steam at two levels of pressure. A representation of all the exergy losses and the exergy balance of this process is given in figure 11. For this case, the heat exchangers were not calculated based on a thermodynamic optimisation as was done earlier but were considered with fixed values for the film heat transfer coefficients and arbitrary values for the allowable pressure drop. The production of electricity for the cycle is equal to the difference $\dot{E}^-_{st} - \dot{E}^+_p$ . The exergy gain is significant when compared with the exergy losses in the steam generator (area about twice as big).

## 12. CONCLUSIONS

The extended pinch analysis method presented here is an important step towards a complete exergy balance of process integrated networks. Knowing that purely economic factors do not yet include environmental or proper natural resource management costs, it is of interest to analyse solutions for which the minimum pinch is determined on an overall exergy basis which includes equipment fabrication exergy (gray exergy). With the proposed method, the advantage of a graphical representation of losses is maintained along with the highlighting of the most critical areas for heat transfer and a complementary representation with the additional topping diagram proposed of the electrical balance which is particularly useful when introducing power units. When dealing with processes covering a broad temperature range with several pinch areas like in a combined power plant, the use as proposed of a global Carnot factor pinch provides a thermodynamically more rational criterion for network design. Further work will include among other, the actual design of heat exchanger networks based on the global Carnot factor pinch for practical examples, the determination of gray exergy values for power units and other components and the addition of pollution and resource scarcity factors.

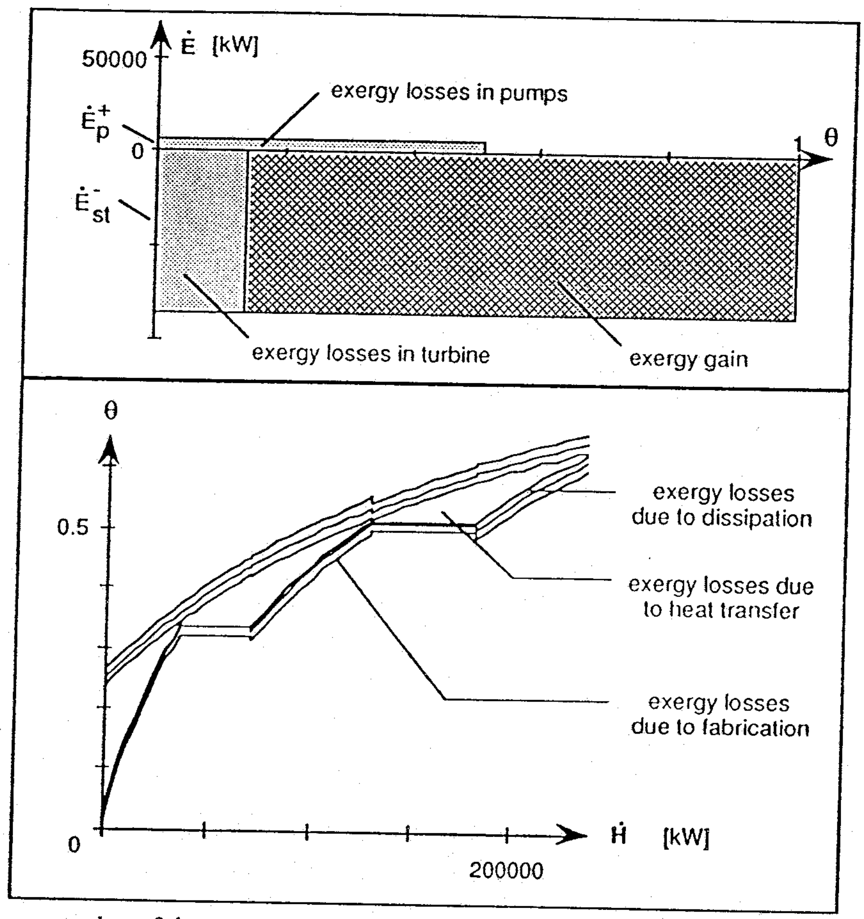


Figure 11 : Representation of the exergy losses of a steam generator and turbine of a combined power plant

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