Methodology and Thermo-Economic Optimization for Integration of Industrial Heat Pumps

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Abstract

This thesis presents a systematic methodology, based on pinch analysis and process integration techniques, to integrate heat pumps into industrial processes. The main goals are to decrease the energy consumption and the corresponding operating costs, and therefore to increase the energy efficiency of an industrial process.

The objective of this thesis is to identify heat pump opportunities, and to optimize simultaneously the energy conversion and utility system of a process. The process and utility integration is realized, using mixed integer linear programming (MILP). In order to find systematically the optimal operating conditions of heat pumps, an optimization framework combining linear and non-linear optimization methods is presented. Technologically feasible heat pumps are collected in a data base and proposed to the process. A multi-objective optimization, combined with process integration methods, gives the possibility to systematically identify optimal heat pump sizing and positioning solutions.

Pinch analysis is a promising tool, however several limits have been discovered, and the basic methodology has been extended to obtain more realistic solutions.

The first extension gives the possibility to integrate heat exchange restrictions due to industrial constraints (e.g. safety reasons, or long distances). A methodology, based on the decomposition into sub-systems, is developed to include heat exchange restrictions. The penalty of these heat exchange restrictions can be decreased by integrating intermediate heat recovery loops, which transfer heat indirectly between two sub-systems. A further developed extension, which enables to define sub-systems at different levels, makes the approach very flexible and useful for many different application cases.

The second extension is realized to integrate multi-period and multi-time slice problems. The main focus is the integration of storage units, to enable the heat recovery between different time slices. The interest in heat pumps can be increased when integrating storage units, because their working hours and profitability increase.

The methodology and its extensions are tested and validated with several real industrial case studies. It is shown that there is a great potential for industrial heat pumps.

Keywords: industrial heat pumps, process integration, pinch analysis, total site integration, energy efficiency, restricted matches, multi-period, multi-time slice, heat storage, mixed integer linear programming (MILP), multi-objective optimization

Résumé

Cette thèse présente une méthodologie systématique pour l'intégration de pompes à chaleur dans des procédés industriels. Elle est basée sur l'analyse de pincement et sur des techniques d'intégration des procédés. Les objectifs principaux sont de réduire les consommations d'énergies et les coûts opératoires associés et, par conséquent, d'augmenter l'efficacité d'un procédé industriel.

L'objectif de cette thèse est d'identifier les opportunités pour les pompes à chaleur, et d'optimiser simultanément le système de conversion d'énergie et les utilitaires d'un procédé. L'intégration du procédé et de ses utilitaires est réalisée en utilisant une formulation MILP (Mixed Integer Linear Programming). Afin d'optimiser systématiquement les conditions opératoires des pompes à chaleur, l'approche présentée utilise à la fois une optimisation linéaire et une optimisation non-linéaire. Des pompes à chaleur technologiquement réalisables sont regroupées dans une base de données utilisée lors des calculs. Une optimisation multi-objectifs, combinée avec des méthodes d'intégration de procédé, donne la possibilité d'identifier systématiquement les solutions optimales pour le positionnement et le dimensionnement de pompes à chaleur.

L'analyse de pincement est un bon outil, mais plusieurs limites ont été découvertes, et la méthodologie de base a été étendue pour obtenir des solutions plus réalistes.

La première extension donne la possibilité d'intégrer des échanges interdits permettant de prendre en compte les contraintes industrielles (par ex. raison de sécurité, ou longues distances). La méthodologie, basée sur la décomposition en sous-systèmes, est développée pour inclure les échanges interdits. La pénalité de ces échanges interdits peut être diminuée par l'intégration de réseaux intermédiaires qui peuvent transférer indirectement de la chaleur. Grâce à une extension, il est possible d'intégrer plusieurs niveaux de sous-systèmes, et par conséquent, l'approche devient très flexible et peut être utilisée pour de nombreuses applications.

La deuxième contribution permet d'intégrer des problèmes multi-périodes et multi-temps. L'objectif principal est l'intégration d'unités de stockage de chaleur, qui permettent de réutiliser celle-ci entre différents segments de temps. L'intérêt des pompes à chaleur peut être augmenté en intégrant des unités de stockage, car leur temps d'utilisation et leur rentabilité augmentent.

La méthodologie et ses extensions sont testées et validées par différents cas d'application industriels. Le grand potentiel des pompes à chaleur industrielles est ainsi vérifié.

Mots-clés: pompes à chaleur industrielles, intégration de procédé, analyse de pincement, intégration totale d'un site, efficacité énergétique, échanges interdits, multi-périodes, multi-temps, stockage de chaleur, mixed integer linear programming (MILP), optimisation multi-objectifs

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Short contents

Contents

Nomenclature

Latin letters

ΔT_{min} Minimum temperature difference [K]

$\dot{M}$ Mass flow [kg/s]

$\dot{E}$ Technical mechanical power [kW]

$\dot{e}$ Specific electric power [kW/kg]

$\dot{E}_q$ Heat exergy [kW]

$\dot{e}_q$ Specific heat exergy [kW/kg]

$\dot{E}_y$ Transformation exergy [kW]

$\dot{e}_y$ Specific transformation exergy [kW/kg]

$\dot{E}_{el,u}$ Consumed ($^+$) / produced ($^-$) nominal electricity by unit u [kW]

$\dot{E}_{f,u}$ Consumed ($^+$) nominal fuel by unit u [kW]

$\dot{L}$ Exergy losses [kW]

$\dot{Q}$ Heat power [kW]

$\dot{q}$ Specific heat power [kW/kg]

A Heat exchanger area [m^2]

AP Annualized profit [€/year]

B Net operating cost savings [€/year]

c_p Specific heat capacity [kJ/kgK]

c_{el} Electricity price for import ($^+$) or export ($^-$) [€/kWh$_{el}$]

c_f Fuel price [€/kWh]

c_u Nominal utility operating cost (excluding fuel and electricity costs) [€/h]

c_w Water price for fresh water ($^+$) and waste water treatment ($^-$) [€/kg]

cf_{hs} Conversion factor [h/s]

cy Number of cycles [year^{-1}]

d Yearly total operating hours of the process [h/year]

d_u	Yearly operating hours of unit u [h/year]
$e_{p_{el}}$	Primary energy of electricity [MJ/kWh$_{el}$]
e_{p_f}	Primary energy of natural gas [MJ/kWh]
E_p	Primary energy [MJ]
f	Installation factor [-]
f_{hl}	Heat loss calculation factor [-]
h	Specific enthalpy [kJ/kg]
h_{cz}	Specific total enthalpy [kJ/kg]
i	Interest rate for investment [-]
$InvC$	Investment costs [€]
$InvC_a$	Annual investment costs [€/year]
k	Specific co-enthalpy [kW/kg]
k_{CO_2}	Electricity to fuel CO_2 content ratio [-]
k_{cost}	Electricity to fuel price ratio [-]
k_{E_p}	Electricity to fuel primary energy ratio [-]
k_{hl}	Heat loss coefficient for storage tanks [kW/m^2K]
M	Mass [kg]
M_p	Tons of product [t/year]
M_{CO_2}	CO_2 emissions [kg]
m_{el}	CO_2 content of electricity [kg/kWh$_{el}$]
m_f	CO_2 content of natural gas [kg/kWh]
MC	Annual maintenance costs [€/year]
n	Expected life time of installations [year]
N_{min}	Minimum number of heat exchanger connections [-]
nf	Number of different fuels [-]
nk	Number of temperature intervals [-]
nl	Number of temperature levels of storage system [-]
nps	Number of parent sub-systems [-]
ns	Number of streams [-]
$nsub$	Number of sub-systems [-]
nu	Number of units [-]

OpC Annual operating costs [€/year]

P Pressure [Pa]

PB Payback period [year]

PR Pressure ratio [-]

Q Heat (energy) [kWh]

q Average heat load [kWh/t]

s Specific entropy [kJ/(kg K)]

T Temperature [K]

T_a Ambient temperature [K]

TC Total annual costs [€/year]

U Global heat transfer coefficient [kW/(m^{2}K)]

U_{cz} Total internal energy [kJ]

V Volume [m^3]

$\dot{\boldsymbol{E}}_{\boldsymbol{el}}$ Total electricity demand ($^+$) or excess ($^-$) [kW]

$\dot{\boldsymbol{Q}}_{\boldsymbol{c,env,k}}$ Heat load of fictive cold envelope stream in interval k [kW]

$\dot{\boldsymbol{Q}}_{\boldsymbol{h,env,k}}$ Heat load of fictive hot envelope stream in interval k [kW]

$\dot{\boldsymbol{Q}}_{\boldsymbol{hts(s),s,k}}$ Heat provided by the heat transfer system (-) to sub-system s or heat removed by the heat transfer system (+) from sub-system s in interval k [kW]

$\dot{\boldsymbol{Q}}_{\boldsymbol{hts+1,k}}$ Heat provided by the higher heat transfer system (-) or heat removed by the higher heat transfer system (+)

$\dot{\boldsymbol{R}}_{\boldsymbol{k}}$ Cascaded heat to lower interval k [kW]

$\dot{\boldsymbol{W}}_{\boldsymbol{k}}$ Cascaded water to lower interval k [kg/s]

$\boldsymbol{f_u}$ Multiplication factor of unit u [-]

$\boldsymbol{M_{0,l}}$ Initial water content of storage tank l [kg]

$\boldsymbol{M_{l,t}}$ Water content of storage tank l after time t [kg]

$\boldsymbol{y_{ij}}$ Integer variable representing the connection (1) or not (0) between hot stream i and cold stream j [-]

$\boldsymbol{y_u}$ Integer variable representing the existence (1) or not (0) of unit u [-]

Greek letters

δ Numerical precision parameter for optimization [-]

$\epsilon_{c,h}$ Combined cooling and heating efficiency [-]

ϵ_c Cooling efficiency [-]

ϵ_{el} Electrical efficiency [-]

ϵ_h Heating efficiency [-]

ϵ_{th} Thermal efficiency [-]

$\eta_{c,h}$ Combined cooling and heating exergy efficiency [-]

η_{COP} Carnot / Exergy efficiency [-]

η_c Cooling exergy efficiency [-]

η_h Heating exergy efficiency [-]

γ Exponent for investment costs estimation [-]

κ Weighting factor [-]

ρ Mass density [kg/m^3]

Θ Carnot Factor [-]

Indices

a Heat source at ambient temperature

c Cold stream

co Water consumption

cog Cogeneration unit

$comp$ Compressor

$cond$ Condenser

cw Cooling water

eva Evaporator

F Cold source

H Hot source

h Hot stream

hex Total heat exchanger area

hp Heat pump

in Inlet

lm Logarithmic

nw Network

out Outlet

pr Water production

rel Relative

$sink$	Heat sink
$source$	Heat source
$valve$	Expansion valve
env	Index for envelope composite curves
hts	Index for heat transfer system
k	Temperature interval / Quality interval
l	Temperature level of storage tank
$mean$	Mean value
o	Optimal value
p	Period
ref	Reference value
s	Index for sub-system
st	Storage system
t	Time
u	Index for unit

Superscripts

$*$	Corrected temperature domain / Instantaneous loads
$+$	Entering the system
$-$	Leaving the system
max	Maximum value
min	Minimum value

Acronyms

CHP	Combined heat and power
COP	Coefficient of performance for ($_h$) heating or ($_c$) cooling
CU	Common unit
DV	Decision variable
HEN	Heat exchanger network
HLD	Heat load distribution
HTS	Heat transfer system
HTU	Heat transfer unit
MER	Minimum energy requirement

MILP Mixed integer linear programming

MINLP Mixed integer non-linear programming

MOO Multi-objective optimization

PUO Process unit operation

TAM Time average model

TSM Time slice model

Conventions

bold characters Optimization variables

Chapter 1

Introduction

1.1 Context and motivation

With raising energy prices and more restricted environment regulations, the rational use of energy in industrial processes becomes more important. Also the reduction of CO_2 emissions, which stands as one of the main reasons for the climate change plays an important role. In 1997, the Kyoto protocol has been signed to reduce the global CO_2 emissions. Still being a topical issue, recent predictions of key technologies for reducing CO_2 emissions have been published (IEA, 2010). Figure 1.1 shows that the greatest part (38% of CO_2 emission reduction) could be achieved by the rational end-use of fuel and electricity.

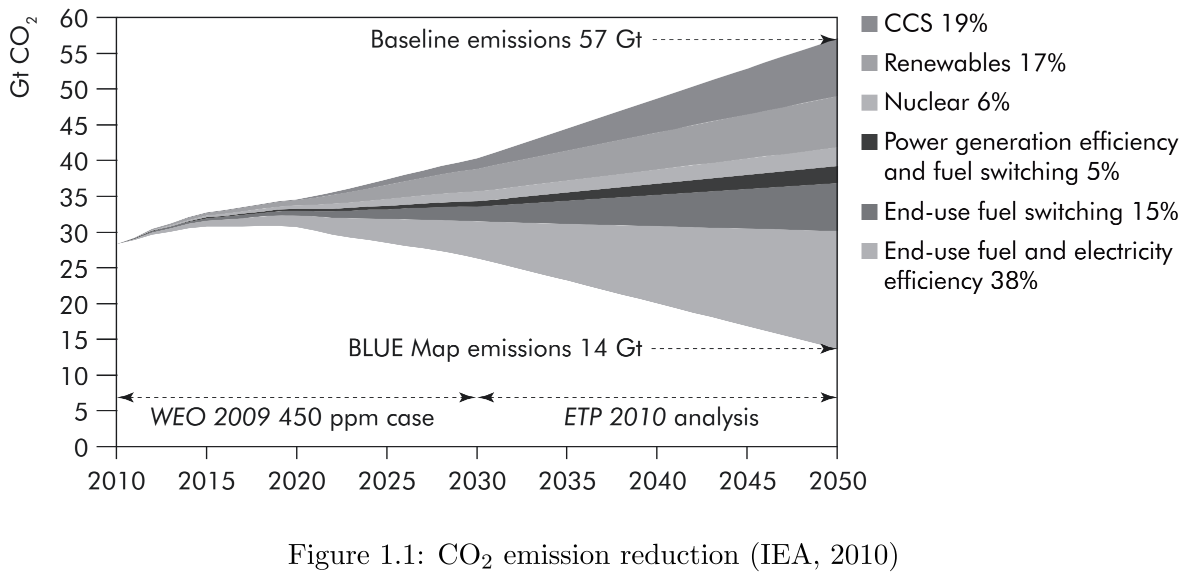


Figure 1.1: CO_2 emission reduction (IEA, 2010)

The industry is interested in the so called ”white certificates”. These documents certify that a certain reduction of energy consumption has been attained. In France, this environmental policy has been introduced in 2005 and the ADEME (Agence De l’Environnement et de la Maîtrise de l’Energie) delivers these certificates to the French industry.

In this context, heat pumps could increase the efficiency of a system or more specifically of an industrial process. They consume less primary energy than conventional heating systems, and thus, they are a promising technology for reducing CO_2 emissions. The overall environmental impact of electric heat pumps depends strongly on how the electricity is produced.

An industrial process can be represented with Figure 1.2. The utilities and the energy conversion units satisfy the process heat demand. It is important to note that the heat pump itself is part of the global system and belongs to the energy conversion units.

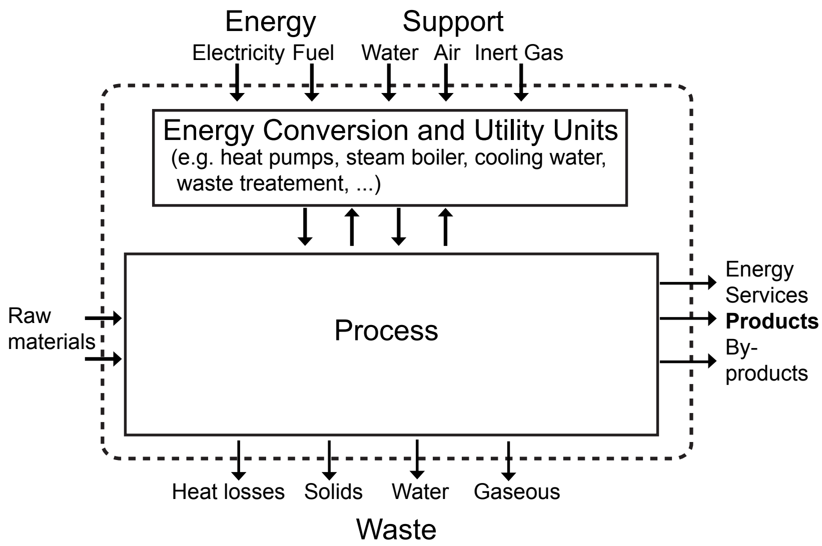


Figure 1.2: Process schematic representation

The optimal integration of energy conversion and utility units, and in particular the placement of industrial heat pumps, can be evaluated by pinch analysis. The goal of pinch analysis and process integration is to identify the heat recovery potential between the hot and cold streams of a system (Kemp, 2007). Process integration also concerns the integration of energy conversion technologies to supply the heating and cooling requirements of the process.

Heat pump technologies and process integration techniques will be combined in order to develop a systematic methodology and thermo-economic optimization to integrate industrial heat pumps.

1.2 Introduction to industrial heat pumps

Spontaneously and without energy input heat is transferred from a higher to a lower temperature. Working as a reverse heat engine, a heat pump device can force the heat to flow in the inverse direction and thermal energy from a lower temperature (heat source) is raised to a higher temperature (heat sink), by using an energy driver (e.g. mechanical work, heat or chemical energy).

A typical heat pump application in industry is to upgrade waste process heat at lower temperature or at the ambiance by electricity, to produce heat at higher temperature which can be used in the process. Heat pumping can also be used for cooling and refrigeration (refrigeration cycles) or to satisfy a cooling and a heating demand at the same time (thermo-frigopump). Thus, in this thesis, the word ”heat pump” will be used for heating, refrigeration and combined heating and cooling.

General information such as heat pump theory, heat pump components, design and industrial applications has been described by Reay and Macmichael (1979) in a practical handbook. Heat pumps are used as refrigeration cycles or to valorize ambient or waste heat. The most common examples for waste heat in industry are cooling water, effluents, condensates, and condensation heat from refrigeration cycles (chillers). Industrial applications show a great variation in heat pump sizes, operating conditions, heat sources and application types. The most attractive sectors are pulp and paper, chemical and food industry, where heat pumps are particularly interesting for drying, evaporation or distillation operations. Especially in food industry, the required moderate temperature levels (up to 120 °C) favor heat pump integration. However, relatively few heat pumps are currently installed in industry. Although there are many possible applications, heat pumps have not been installed, because of unfavorable economic conditions (Favrat, 2008) and the lack of understanding about the correct placement of heat pumps according to pinch analysis rules. Nowadays environmental regulations become stricter and properly integrated industrial heat pumps can become an important technology to reduce emissions, improve efficiency, and limit the use of cooling water.

Heat pump opportunities have been described by Flach-Malaspina and Perrotin (2006). Abou-Khalil (2008) developed a methodology to perform energetic and exergetic analysis in food industry. In his thesis, he focused on process analysis to collect necessary data. He studied the case of a cheese factory and the possibility of installing a heat pump to valorize effluents. In his work the methods of pinch analysis are used, but there is no systematical integration of energy conversion and utility systems. Particular installations of heat pumps in an industrial process have also been studied by Murr (2010). With the focus on an experimental setup, Assaf (2010) studied industrial heat pumps at high temperatures. An algorithmic approach based on process integration has also been developed by Dubuis (2007).

1.2.1 Heat pump technologies

In a heat pump, lower quality energy can be upgraded with electricity (e.g. compression heat pumps) or with heat (e.g. absorption heat pumps, ejectors, thermal engine).

Figure 1.3 gives a short overview of heat pump technologies with different energy drivers. Other more exotic heat pump types are for example thermo-electric or magnetic heat pumps, which like the dominant family of electrically driven heat pumps also use electricity as energy driver.

Figure 1.4 shows the distribution of heat pumps in industrial processes of 8 selected countries (Berntsson and Franck, 1997). Compression heat pumps represent the majority of the applications in which: 50% are mechanical vapour re-compression heat pumps, and 45% are closed cycle compression systems.

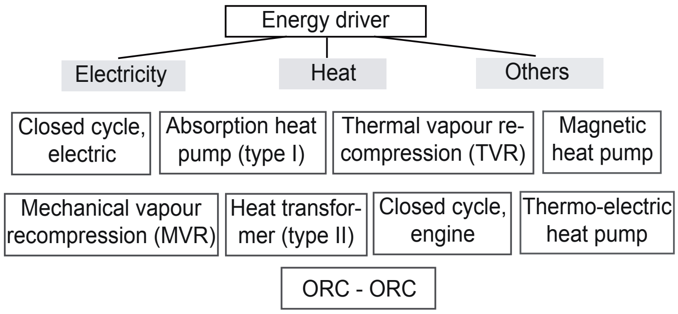


Figure 1.3: Heat pump types and their different energy drivers

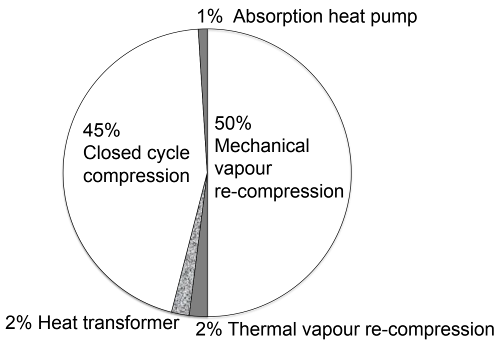


Figure 1.4: Distribution of heat pumps according to Berntsson and Franck (1997)

This thesis focuses mainly on electricity as energy driver. Closed cycle heat pumps and mechanical vapour re-compression units are analyzed in detail. The closed heat pump cycle (single stage) consists in one compression, one condensation, one expansion and one evaporation step. The heat exchange with the process occurs indirectly through the refrigerant. Multi-stage heat pumps have several compression operations and can be used to achieve higher temperature differences. On the contrary, open cycle heat pumps or mechanical vapour re-compressions (MVRs) use the process streams directly. They consist only in one compression and one condensation step.

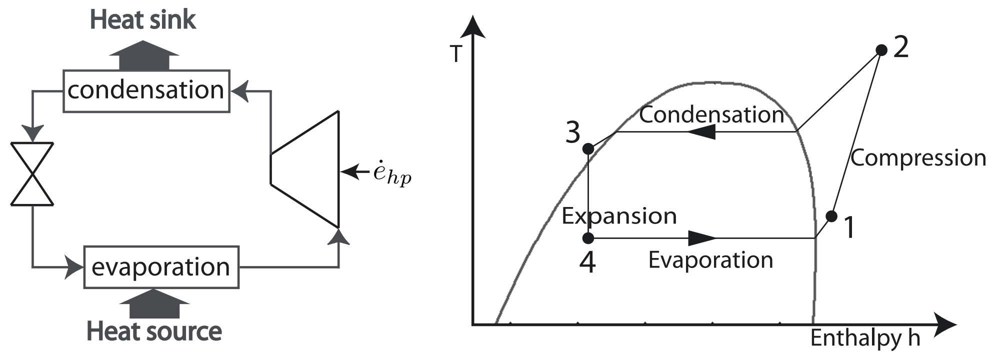
1.2.2 Thermodynamic principles

Heat pumps are characterized by the coefficient of performance (COP), given by Equation (1.1). It is defined as the ratio of heat delivered by the heat pump and the driver energy consumption.

$$COP = \frac{\text{useful heat [kW]}}{\text{energy driver consumption [kW]}} \tag{1.1}$$

Other key performance factors as the exergy efficiency, the economic indicators (operating and investment costs) and the environmental indicator (CO_2 emissions) will be described in detail in Chapter 2.

The illustration of a simple bi-thermal closed cycle heat pump driven by electricity is used to describe thermodynamic concepts (Figure 1.5). After evaporation at the lower pressure level, the fluid is sent to a compressor driven by an electric motor. When the higher pressure level is reached, the fluid is first cooled down to its saturation temperature and then condensation takes place. Heat is delivered at higher temperature. After desuperheating, condensation and subcooling, the fluid is expanded to the lower pressure level. During evaporation, heat is absorbed by the fluid.



(a) Simple HP cycle (b) Thermodynamic diagram

Figure 1.5: Simplified heat pump cycle

As an example, a heat pump using water from a lake (ambient energy as heat source) for heating is considered. This kind of heat pump with an electric driven compressor is installed at EPFL, where the working fluid is ammonia, the cold source is the Geneva Lake, and the hot source is the building heating network (Pelet et al., 1997).

1.3 Introduction to process integration

Process integration and pinch analysis techniques aim at maximizing the heat recovery and consequently minimizing the energy consumption of an industrial processes. In the following, a short overview on the basics of pinch analysis is given (Dumbliauskaite, 2009). A general state of the art is then presented in a second step.

1.3.1 Theory of pinch analysis

1.3.1.1 Process streams

Pinch analysis can be applied to processes defined by a set of mass and energy flows. The application of the pinch theory requires the definition of two types of thermal streams:

- hot streams, requiring cooling. They thus deliver heat.
- cold streams, requiring heating. They thus absorb heat.

Hot and cold streams represent the root of the pinch analysis, since it is possible to set up heat exchangers between these two matched streams, as their energy requirements are opposite. Thermal streams are generally defined by their input and output temperatures and enthalpies. The heat load required by a stream can be computed with Equation (1.2) by applying the first principle of thermodynamics for an open system (Borel and Favrat, 2010).

$$\frac{dU_{cz}}{dt} = \sum_k [\dot{E}_k^+] + \dot{E}_a^+ + \sum_i [\dot{Q}_i^+] + \dot{Q}_a^+ + \sum_j [h_{czj} \dot{M}_j^+] \tag{1.2}$$

When the system operates at steady state conditions and there are neither thermal losses to the atmosphere nor mechanical work, then Equation (1.2) becomes Equation (1.3). The variations of kinetic and potential energies are also not taken into account.

$$\frac{dU_{cz}}{dt} = 0 = \dot{Q}^+ + h_{in} \cdot \dot{M}^+ - h_{out} \cdot \dot{M}^+ \tag{1.3}$$

Thus, for a given stream, the computation of the corresponding heat load $\dot{Q}^+$ is given by Equation (1.3), which value is positive for cold streams and negative for hot ones. With the assumption of constant pressure, the more simplified Equation (1.4) can be used for real single phase fluids.

$$\dot{Q}^+ = \dot{M}^+(h_{out} - h_{in}) = \dot{M}^+ \cdot c_p(T_{out} - T_{in}) \tag{1.4}$$

1.3.1.2 Definition of ΔT_{min}

Figure 1.6 provides an example of two streams: a cold stream heated from $T_{c,in}$ to $T_{c,target}$ and a hot stream cooled from $T_{h,in}$ to $T_{h,target}$. A heat exchanger enables the heat transfer from the hot stream to the cold one. The quality of the exchange is characterized by the minimal approach temperature (ΔT_{min}) between the hot and the cold streams. The lower the ΔT_{min}, the greater the heat exchange between the streams, and the lower the external utility requirements. This can be seen on Figure 1.7, by shifting horizontally the curve of the cold stream (black arrow).

The graphical representation of the possible heat exchange as a function of the ΔT_{min} enables identification of the minimum energy requirements in heating and cooling (MER_{hot} and MER_{cold} respectively). The ΔT_{min} value of a heat exchanger is a function of its exchange area. A larger heat transfer area allows greater heat exchange and therefore corresponds to a lower value of ΔT_{min}. However, this results in higher investment costs. Thus, the value of ΔT_{min} represents a parameter used to determine the trade-off between the energy costs and the investment. It is a decision variable that enables the definition of the optimal size of the heat exchanger. In order to calculate the most desirable value of ΔT_{min}, it is necessary to express the objectives (operating and

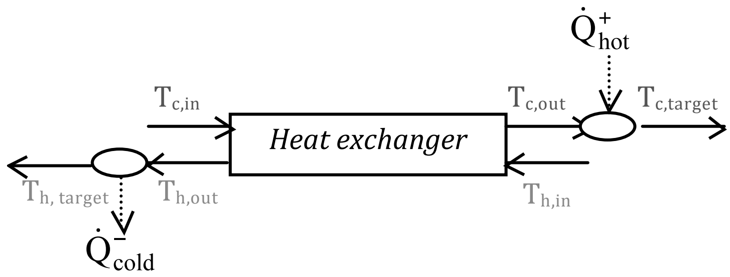


Figure 1.6: Counter current heat exchanger

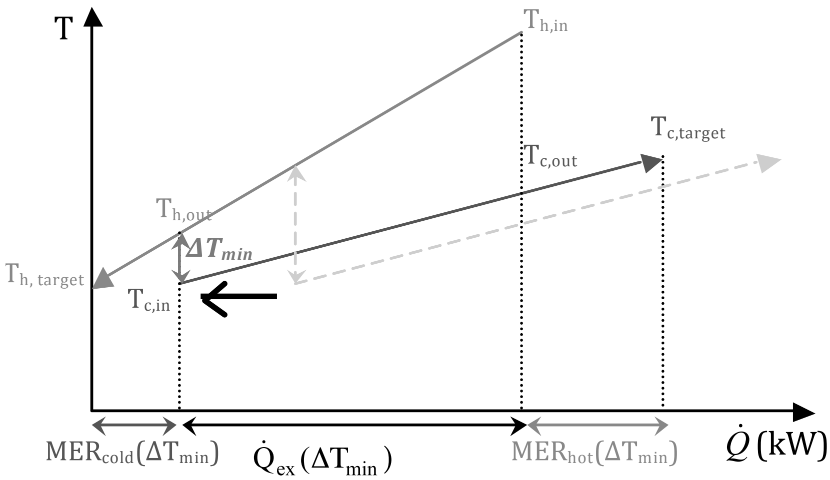


Figure 1.7: Representation of the counter current heat exchanger in a heat load temperature diagram

investment costs) as a function of this value. Finally, the optimal value of ΔT_{min} can be determined by the minimization of both investment and resulting operating costs, as it is shown in the example of Figure 1.8.

1.3.1.3 Composite curves

A graphical representation of the thermal requirements of a process has been developed by Linnhoff and Flower (1978) in order to have a systematic overview of its energetic demands.

As mentioned before, a process can be defined as a set of hot and cold streams. As the enthalpy is an additive quantity, it is possible to sum the heat loads of all hot and all cold streams for each temperature interval k $[T_k, T_{k+1}]$. The total available and required heat loads can therefore be drawn as a function of the temperature, corresponding to the hot and cold composite curves (Figure 1.9).

A value of $\Delta T_{min}/2$ can be associated with each stream. When representing the hot and cold

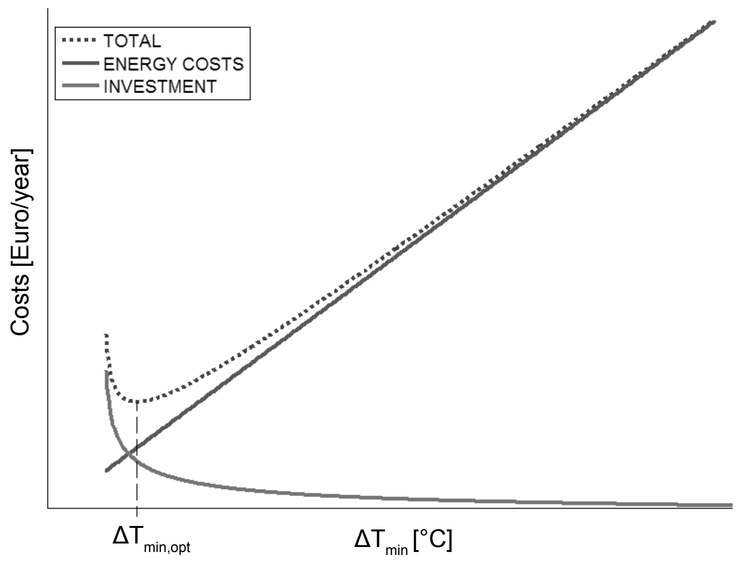


Figure 1.8: Choosing ΔT_{min}: trade-off between operating and investment costs

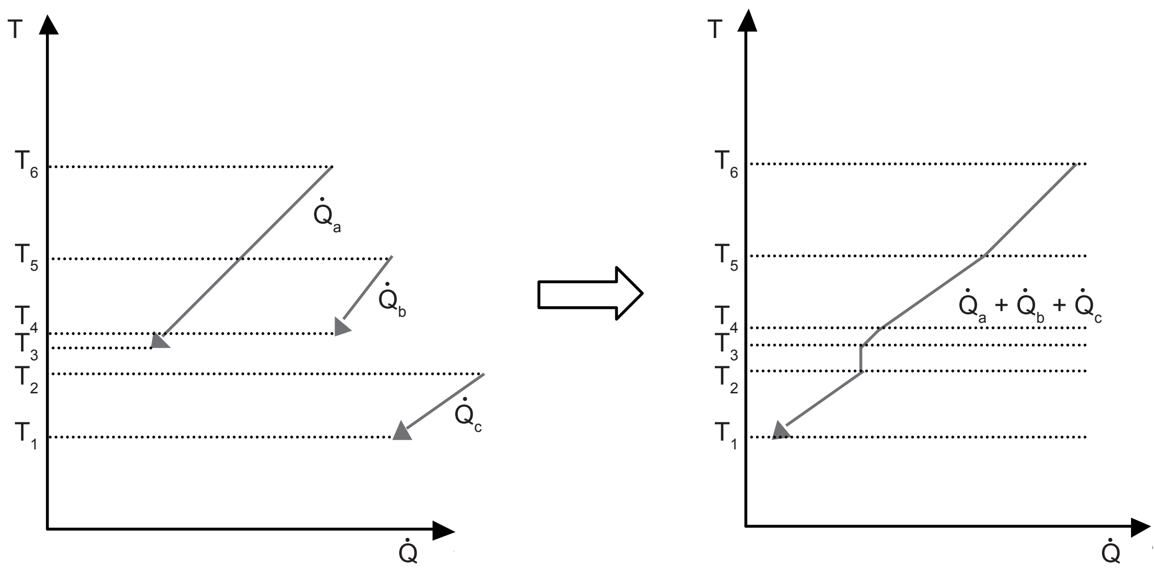


Figure 1.9: Hot composite curve construction

composite curves as a function of their corrected temperatures, namely $T_h^* = T - \Delta T_{min}/2$ and $T_c^* = T + \Delta T_{min}/2$, it is possible to define the corresponding grand composite curve, obtained by calculating the enthalpy difference between the hot and cold composite curves (Equation (1.5)).

$$\dot{Q}_k = \dot{Q}_{h,k} - \dot{Q}_{c,k} \qquad \forall k \tag{1.5}$$

The pinch point is located at the intersection of the hot and cold composite curves drawn as a function of their corrected temperatures, which results in the intersection of the grand composite curve with the temperature axis, as shown in Figure 1.10.

The pinch temperature splits the process into two parts: above the pinch temperature, the process generally requires heating (heat sink), whereas it requires generally cooling below the pinch (heat source).

When integrating utilities, defined as the external energy supply, the grand composite curve is used to determine the amount of heat that must be provided to the process or removed from it using energy conversion technologies.

In particular, utility integration must respect the following rules:

- Above the pinch temperature, hot streams are to be cooled without external cold utility.
- Below the pinch temperature, cold streams are to be heated without external hot utility.
- No heat exchange is allowed across the pinch point.

In each case, violation of the rule induces an increase in energy requirement above and below the pinch temperature.

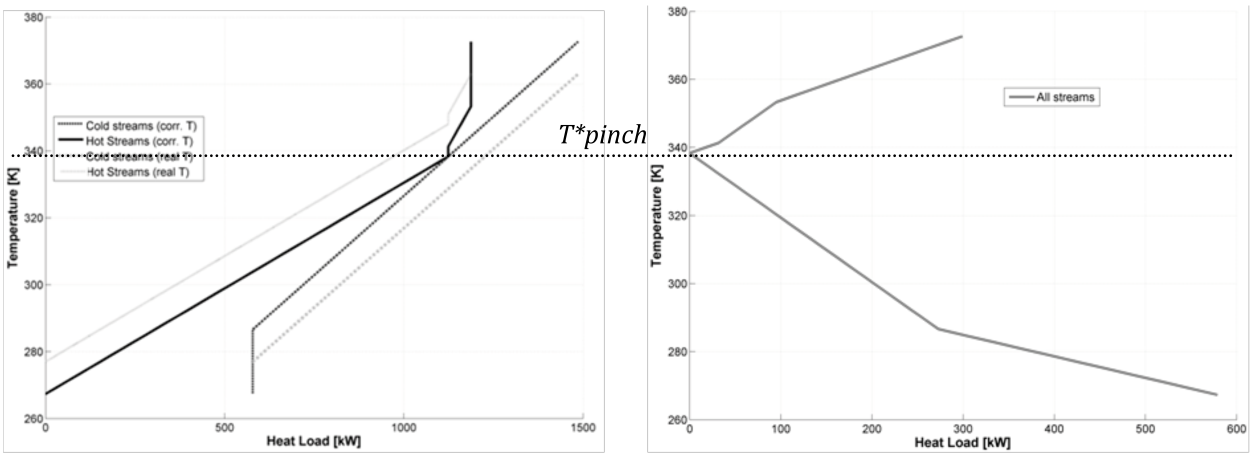


Figure 1.10: Hot and cold composite curves and corresponding grand composite curve

1.3.2 General state of the art of process integration

The interest in heat pumps and in process integration is strongly linked to the energy prices and started around 1980 (Figure 1.11).

1.3.2.1 Heat exchanger network synthesis - graphical approach

Process integration was originally developed by chemical engineers interested in the synthesis of heat exchanger networks. Umeda et al. (1978) combined the thermodynamic analysis with heuristic rules

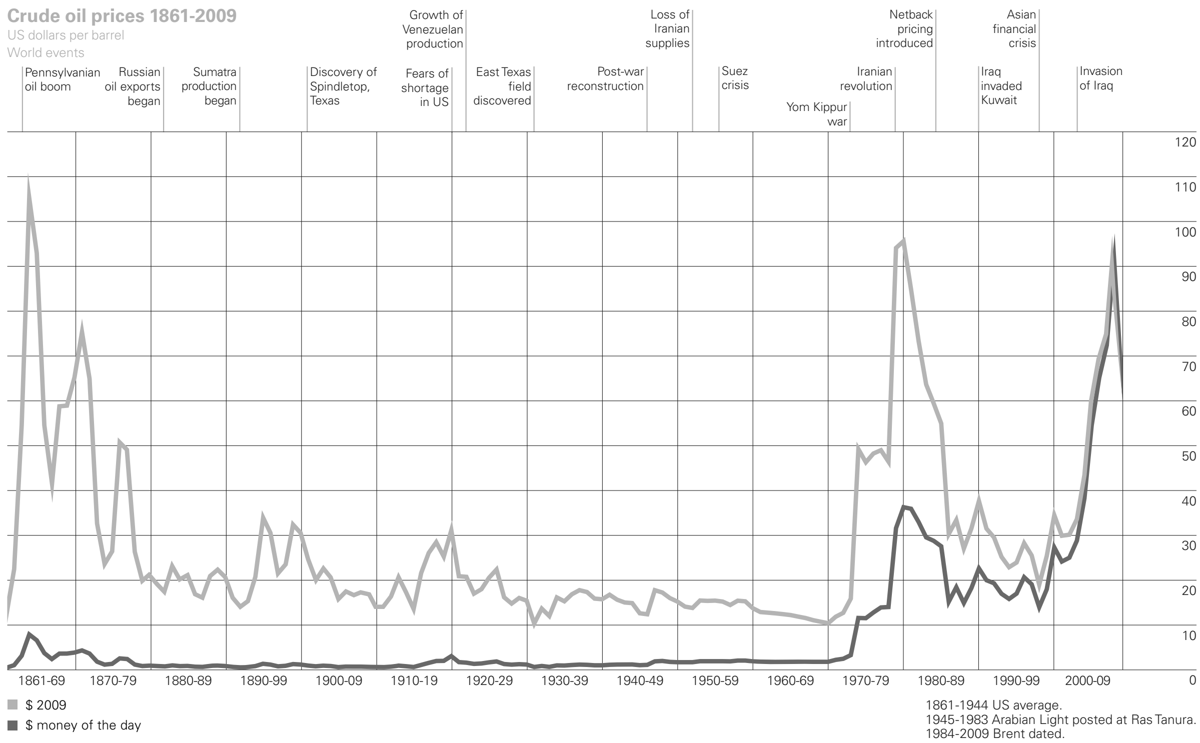


Figure 1.11: Oil price statistics (BP, 2010)

to optimize the heat exchanger network and observed the pinch point. Independently and at the same time, Linnhoff and Flower (1978) introduced the temperature interval method and presented an iterative method to solve the problem table. This method consists in a simple algorithm to obtain mathematically the minimum energy requirement. The resulting grand composite curve could therefore also be seen as the graphical representation of the problem table. That way, they systematically generated optimal heat exchanger networks, that satisfy the minimum hot and cold utilities. Basically the minimum required utilities are obtained by using thermodynamics.

A review of process synthesis was given by Nishida et al. (1981). They focused among others on heat exchanger networks.

Another general review can be found in Linnhoff (1993). Ongoing development areas were described (e.g. batch, total site integration). Linnhoff et al. (1992) wrote a user guide for process integration. General concepts of pinch analysis are explained in detail. The main focus is on the heat exchanger network, which is evaluated after the construction of the composite curves and the grand composite curve with the problem table method. Heuristic methods to find a solution for the heat exchanger network are presented, however for industrial processes with a large number of streams, it is difficult to apply these methods. A second edition has been published by Kemp (2007). Utility integration, process modification, batch problems, and more application cases have been presented.

Dhole and Linnhoff (1992) proposed the total site integration with combined heat and power production for the total system. Optimal utilities can be identified for the complete process, however self-sufficient pockets are neglected in this approach. Further work on the total site approach is summarized in a book by Klemeš et al. (2011).

Maréchal and Kalitventzeff (1996) proposed a graphical technique called ”integrated composite curves” to evaluate utility integration.

1.3.2.2 Mathematical programming

Pinch analysis was first developed for the synthesis of heat exchanger network and later enhanced for process integration of complete process sites including utility systems. Based on the approach of Linnhoff and Flower (1978), Cerda et al. (1983) presented the transportation problem which calculates the minimum utility by using linear programming. The classical transportation problem, where goods are shipped from several origins to several destinations at a minimum cost has been used. By analogy the authors proposed to ”transport” the heat from the hot streams to the cold streams at minimum utility cost. By solving this linear problem, the heat exchanger network of a given process is optimized. Moreover it is possible to integrate utilities with variable temperatures, but nevertheless no systematic utility integration is considered. The main disadvantage of this approach is that each cold stream has a fictive connection with each hot stream and therefore the problem becomes soon very complex for a high number of streams.

Papoulias and Grossmann (1983a) introduced in a series of 3 publications a mixed integer linear programming (MILP) approach for the systematic synthesis of utility systems. The integer variable is linked to the activation of an utility in the superstructure. The second part (Papoulias and Grossmann, 1983b) deals with the heat cascade and heat recovery networks. Also the possibility of restricted matches between two streams has been included in this formulation. Finally in the third part (Papoulias and Grossmann, 1983c), total process systems consisting in a chemical plant, heat networks and utility systems, has been solved simultaneously by a MILP formulation.

Further developments have been published. A new strategy for solving process integration problems has been presented by Maréchal and Kalitventzeff (1989). They suggested to optimize first the utility integration. Then, based on the fixed list of the process and utility streams, they calculated the heat load distribution (HLD), which is the first step to finally design the heat exchanger network (HEN). The same authors (Maréchal and Kalitventzeff, 1998a) adapted later the pinch analysis methods to solve industrial site scale problems including the combined heat and power production with the example of steam networks. A complete method, combining pinch analysis and mathematical programming tools is presented. The approach, beginning from the construction of composite curves and utility integration up to the HLD and the HEN is shown with the help of an application case on a chemical process. The integration of the utility system has to be considered simultaneously with the process integration (Maréchal and Kalitventzeff, 1998b). The big advantage of their approach is that it is not necessary to define explicitly all possible heat exchange connections between hot and cold streams to calculate the utility integration. Thus, the problem size is reduced and more complex problems can be solved. On the other hand, the disadvantage is that the previously skipped heat exchange connections are needed to calculate the heat load distribution and the heat exchanger network. Therefore, even if the utility integration can be calculated it is possible that the heat load distribution and the heat exchanger network can not be solved for complex problems.

For each chapter a separate detailed literature review is presented.

1.4 Introduction to case studies

In the thesis, a methodology to integrate heat pumps and tools will be developed. The development will be validated on available test cases taken from industrial applications. All details of each real case study are given in appendix.

- Dairy B.1
- Cheese factory B.2
- Brewery B.3

These test cases are from the food industry and particularly promising since the required moderate temperature levels promote heat pump integration.

1.5 Objectives and synthesis

The first goal is to optimize industrial processes in order to increase their energy efficiency and to reduce the energy consumption and consequently the operating costs. In this context industrial heat pumps are an important technology and therefore the focus is given to the development of a methodology to estimate and evaluate heat pump opportunities in industrial processes. The two main questions that have to be answered are summarized below:

- Where are the heat pump opportunities in a given process ?
- What are the optimal heat pump operating conditions ?

Chapter 2 will introduce heat pump performances and a preliminary methodology to rapidly evaluate a heat pump potential without doing a complete process integration approach.

The process integration approach will be described in Chapter 3. Particular attention will be given to the heat pump integration. A complete state of the art and the applied methodology based on the approach of (Maréchal and Kalitventzeff, 1998b) will be presented. The case study of a dairy will show the importance to consider simultaneously all process and utility streams. Especially, it will be shown that a heat pump combined with a refrigeration cycle is a good option to reduce the energy consumption.

Heat pump integration will be further analyzed in Chapter 4, where an optimization methodology will be developed to systematically test and integrate different heat pump types. For that, a heat pump data base will be defined, where technologically feasible heat pumps are collected. For example, the acceptable volumetric flow-rates and typical isentropic efficiencies of a given compressor type will be taken into account. Also available refrigerants and their temperature ranges will be considered. Finally, a multi-objective optimization combined with process integration will give the possibility to identify optimal heat pump solutions.

The used heat cascade formulations of Chapter 3 are quite powerful, but in order to have more realistic solutions further developments are necessary. For this, the heat cascade formulation will be extended in Chapter 5 to enable the integration of heat exchange restrictions due to industrial constraints. The new concept of ”envelope composite curves”, which identifies optimal indirect heat transfer networks, will be presented. Heat exchange restrictions can also be used to solve multi-period problems by defining heat exchange restrictions due to non-simultaneous process operations.

Chapter 6 will extend the heat cascade formulation to multi-period and multi-time slice problems. Furthermore, heat storage between different time slices become possible and the method enables in addition to size new heat pumps and storage tanks.

The implementation approach will be summarized in Chapter 7 and finally, Chapter 8 will draw the conclusions and gives an outlook to future projects.

Chapter 2

Potential of industrial heat pumps

This chapter describes heat pump performances and presents a preliminary analysis to evaluate heat pump potentials, without doing a complete process integration study.
Related publication:Becker et al. (2011a).

2.1 Introduction

The interest of heat pumps has been shown in the literature: industrial heat pump technologies, their integration into the process, and existing installations have been documented by Berntsson and Franck (1997). Zogg (2008) presented the history of heat pumps in Switzerland as well as historical and recent examples of application of heat pumps. The progress of heat pump technology and its applications have also been described by Favrat (2008). The technology is already quite popular in the domestic sector, but relatively few heat pump units are today installed in industrial sites, although these sites often present the advantage of continuous waste heat and heat demand. A market analysis on industrial heat pumps has been investigated by Lambauer et al. (2008). According to them, the main installation barriers are the low fuel prices, the lack of knowledge in heat pump technologies, the difficulties to find the optimal placement and the competition with other heat recovery systems.

Contrary to this widespread opinion (Lambauer et al., 2008), the competition with other heat recovery systems is not a real barrier. The choice between a heat pump, a direct heat recovery system or a combination of both depends on the process. This problematic is treated in Chapter 3, which focuses on the right placement of heat pumps by applying process integration and pinch analysis techniques.

Regarding current heat pump technologies, it can be added that a required optimal heat pump may not be available on the market even if, from the technological point of view, the heat pump is feasible.

This chapter shows the interest of heat pumps compared to conventional heating system (e.g. boiler or engine). Assuming that the heat source and the heat sink of a heat pump are defined, the main goal of the following approach is to evaluate a heat pump potential without having to analyze the whole process and to realize a complete process integration study. In the next sections, first the performance indicators will be presented, and then heat pumps will be compared to conventional heating systems.

2.2 Performance indicators

Figure 2.1 summarizes all possible heat pump cycles. They can be categorized by comparing the temperature levels of the heat source and the heat sink to the ambient temperature. Cases A and B correspond to a heating system, case C corresponds to a combined heating and cooling system and finally cases D and E correspond to refrigeration systems.

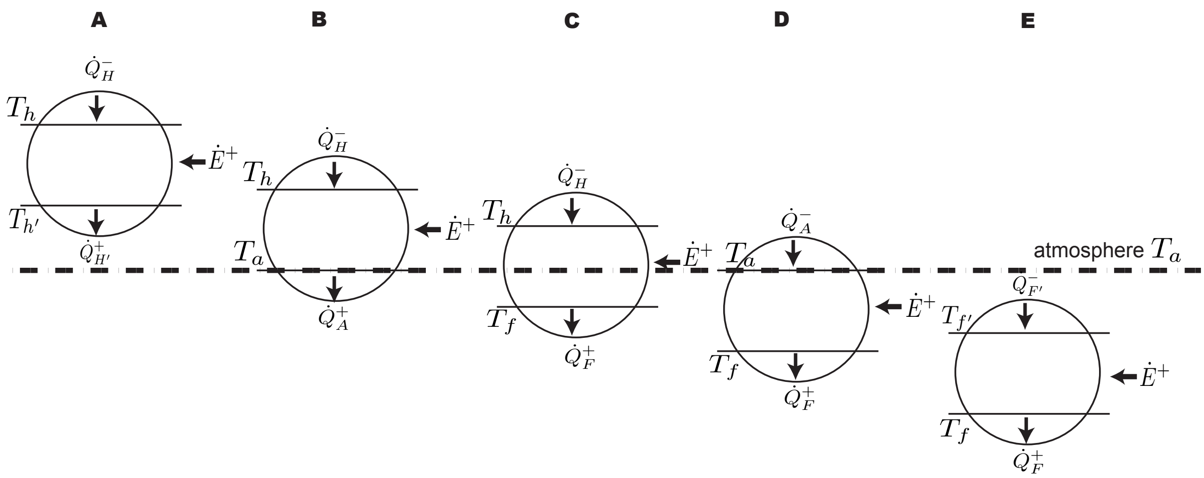


Figure 2.1: Typical heat pump cycles (Borel and Favrat, 2010)

2.2.1 COP definition

Heat pumps are characterized by their coefficient of performance (COP), which is generally defined as the ratio of the useful energy and the energy consumption of the driver. The COP is an important indicator to evaluate heat pump technologies, but it is also crucial to consider the overall efficiencies, including the process and utility units. A heat pump with a high COP can be counterproductive when it is not correctly integrated in the process (see Chapter 3).

To compare heat pump performances, the COP is used. This is mainly because it is not so easy to define a general efficiency. It can happen that the efficiency is not between 0 and 1, and thus it is better to call this number COP.

Usually, the COP of a heat pump is defined as shown in Equation (2.1). $\dot{Q}_H^-$ is the heat delivered by the heat pump and $\dot{E}^+$ is the electricity consumed by the compressor.

$$COP_h = \frac{\dot{Q}_H^-}{\dot{E}^+} \tag{2.1}$$

In the same way the COP for a refrigeration cycle is defined (Equation (2.2)), where $\dot{Q}_F^+$ is the heat extracted by the heat pump.

$$COP_f = \frac{\dot{Q}_F^+}{\dot{E}^+} \tag{2.2}$$

Although these definitions are widely used in industry, theoretically they are only correct for a heat pump having its heat source at ambient temperature (Case B) and a refrigeration cycle having its heat sink at the ambient temperature (Case D).

The COP definition has to be reviewed depending on the temperature demands achieved by the heat pump (heating, refrigeration, combined heating and cooling, ...). For example, taking the COP definition of case C: for an industrial heat pump, theoretically speaking, both energy demand should be accounted in the COP, because the process profits from both the heating and the cooling. Thus, a new definition of the COP could be defined like it is shown in Equation (2.3). Similar attempts could be done for Cases A and E.

$$COP_h = \frac{\dot{Q}_H^- + \dot{Q}_H'^+}{\dot{E}^+} \tag{2.3}$$

It is important to notice that the COP definition is a convention, which cannot evaluate the quality of an integrated heat pump cycle. In order to be compatible with the literature, in this thesis the COP will be related to a performance of the heat pump machine by using the definitions in Equations (2.1) and (2.2) for heating and cooling respectively. But to be rigorous, when comparing heat pump cycles, the exergy efficiency presented in the next section is preferred and used.

Assuming that the temperature levels of the heat source and heat sink are known, the COP can be defined as a first approximation by considering an efficiency, according to the theoretical value (η_{COP}). The COP of a heat pump is defined by Equation (2.4) and may be estimated considering the Carnot factor η_{COP}. According to rules for good practice, 55% can be considered for this factor, which represents the exergy efficiency of a cycle.

$$COP = \frac{\dot{Q}}{\dot{E}} = \eta_{COP} \frac{T_{sink}}{T_{sink} - T_{source}} \tag{2.4}$$

2.2.2 Exergy analysis

Another important indicator is the exergy efficiency. This one is based on the second principle of thermodynamics, and evaluates the quality of the thermodynamic cycle. The exergy concept is briefly summarized below (Borel and Favrat, 2010). First, the co-enthalpy (Equation (2.5)), the Carnot factor (Equation (2.6)), the received heat exergy (Equation (2.7) and the received transformation exergy in steady state condition (Equation (2.8)) are defined.

$$k = h - T_a \cdot s \tag{2.5}$$

$$\Theta = 1 - \frac{T_a}{T} \tag{2.6}$$

$$\dot{E}_q^+ = \int \Theta \cdot \delta \dot{Q}^+ \tag{2.7}$$

$$\dot{E}_y^+ = \sum (k \cdot \dot{M}^+) \tag{2.8}$$

Equation (2.9) gives the exergy losses $\dot{L}$ and the overall exergy balance.

$$\sum (\dot{E}_k^+) + \sum (\dot{E}_q^+) + \sum (\dot{E}_y^+) = \dot{L} \geq 0 \tag{2.9}$$

$\dot{E}_k^+$ is the work received by the system from a machine (e.g. electrical or mechanical work).

2.2.2.1 Example of exergy analysis for a heat pump device

The exergy analysis is shown with the help of a typical heat pump cycle, which satisfies a hot and a cold demand of a process. Figure 2.2 shows the studied heat pump installation.

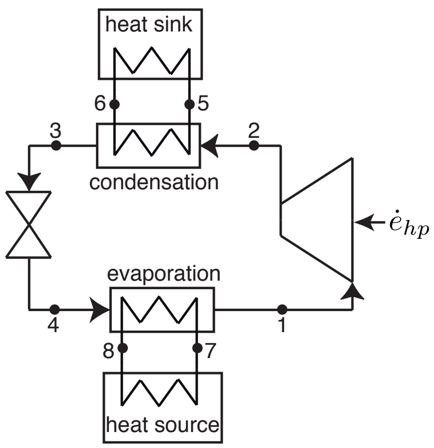


Figure 2.2: Heat pump installation for exergy analysis

The corresponding thermodynamical cycle is shown in a T-h diagram (Figure 2.3) and can be decomposed in 4 operations:

- 1 - 2: Mechanical compression.
- 2 - 3: Heat supply to the process. Thermodynamically, this operation consists in the desuperheating, the condensation and the subcooling.
- 3 - 4: Expansion.
- 4 - 1: Heat supply from the process (cooling). Thermodynamically, this operation consists in the evaporation and the superheating to avoid droplets in the compressor.

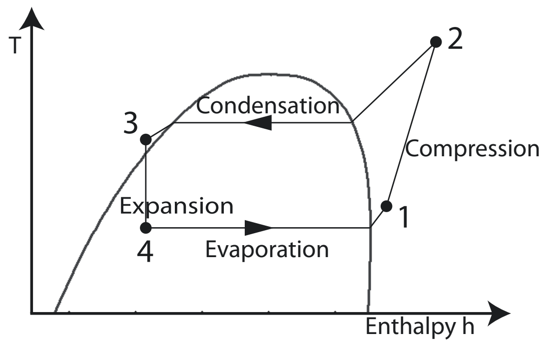


Figure 2.3: Thermodynamic heat pump cycle

It is considered that the thermodynamic state (T, h, s) is known in each point. The total exergy received by the heat pump is given in Equation (2.10). It is assumed that the heat sink and the heat source are above the ambient temperature.

$$\dot{L}_{hp} = \dot{M}_{hp} \cdot e^{+}_{hp} - (1 - \frac{T_a}{T_{sink}})\dot{Q}^{-}_{H} + (1 - \frac{T_a}{T_{source}})\dot{Q}^{+}_{F} \tag{2.10}$$

$\dot{Q}^{-}_{H}$ (Equation (2.11)) is the heat delivered to the heat sink and $\dot{Q}^{+}_{F}$ (Equation (2.12)) is the heat supplied from the heat source.

$$\dot{Q}^{-}_{H} = \dot{M}_{hp}(h_2 - h_3) \tag{2.11}$$

$$\dot{Q}^{+}_{F} = \dot{M}_{hp}(h_1 - h_4) \tag{2.12}$$

The exergy losses of the compressor (Equation (2.13)), the condenser (Equation (2.14)), the expansion valve (Equation (2.15)) and the evaporator (Equation (2.16)) are given below. The exergy losses in the hot and cold network are expressed by Equations (2.17) and (2.18).

$$\dot{L}_{comp} = \dot{M}_{hp} \cdot e^{+}_{hp} - \dot{M}_{hp}(k_2 - k_1) \tag{2.13}$$

Table 2.1: Thermodynamic state of R245fa (points 1-4), hot water from condensation (nw1, point 5-6) and cooling water of evaporation (nw2, point 7-8), $T_a = 15$ °C, $T_{sink} = 55$ °C, $T_{source} = 45$ °C

Point	P	T	h	s	k	$\dot{M}$
	[bar]	[°C]	[kJ/kg]	[kJ/ kg K]	[kJ/kg]	[kg/s]
1	1.789	40	436.78	1.7872	-78.20	30
2	5.372	74.3	462.96	1.8040	-56.86	30
3	5.372	60	281.27	1.2673	-83.90	30
4	1.789	30	281.27	1.2750	-86.12	30
5	1	58	242.88	0.8061	10.60	651.2
6	1	56	234.51	0.7807	9.55	651.2
7	1	44	184.33	0.6254	4.12	558.1
8	1	42	175.97	0.5990	3.39	558.1

Table 2.2: Exergy losses of the example heat pump

	$\dot{L}_{hp}$	$\dot{L}_{comp}$	$\dot{L}_{cond}$	$\dot{L}_{valve}$	$\dot{L}_{eva}$	$\dot{L}_{nw1}$	$\dot{L}_{nw2}$
[kW]	561.0	145.2	127.4	66.6	169.8	19.4	32.6
[%]	100	25.9	22.7	11.9	30.2	3.5	5.8

$$\dot{L}_{cond} = \dot{M}_{hp}(k_2 - k_3) - \dot{M}_{w1}(k_5 - k_6) \tag{2.14}$$

$$\dot{L}_{valve} = \dot{M}_{hp}(k_3 - k_4) \tag{2.15}$$

$$\dot{L}_{eva} = -\dot{M}_{hp}(k_1 - k_4) + \dot{M}_{w2}(k_7 - k_8) \tag{2.16}$$

$$\dot{L}_{nw1} = \dot{M}_{w1}(k_5 - k_6) - (1 - \frac{T_a}{T_{sink}})\dot{Q}_H^- \tag{2.17}$$

$$\dot{L}_{nw2} = \dot{M}_{w2}(k_7 - k_8) + (1 - \frac{T_a}{T_{source}})\dot{Q}_F^+ \tag{2.18}$$

To illustrate the concept, a simple example of a R245fa heat pump and its corresponding thermodynamic conditions are given in Table 2.1.

Table 2.2 and the Sankey diagram in Figure 2.4 show the numerical results.

The overall exergy efficiency of the heat pump can be calculated by Equation (2.19).

$$\eta_{hp} = \frac{(1 - \frac{T_a}{T_{sink}})\dot{Q}_H^-}{\dot{E}_{hp}^+ + (1 - \frac{T_a}{T_{source}})\dot{Q}_F^+} = 0.54 \tag{2.19}$$

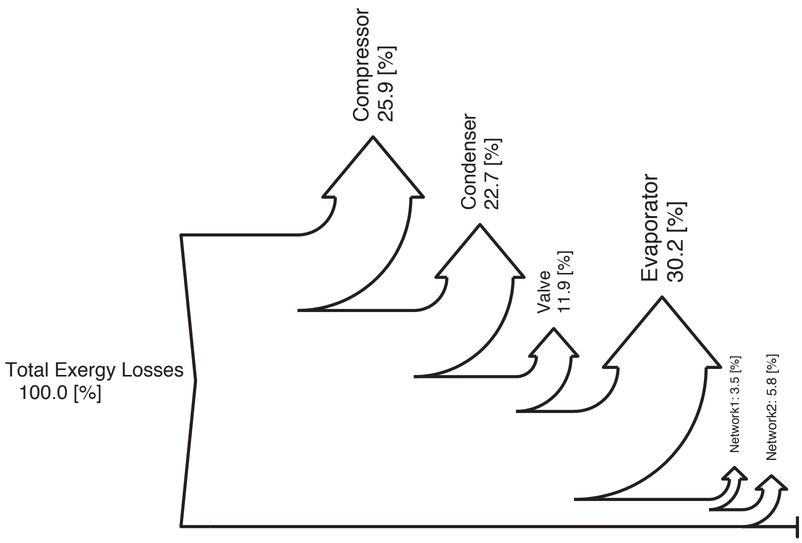


Figure 2.4: Typical exergy losses of a heat pump

2.2.2.2 Exergy efficiencies of heat pump cycles

Borel and Favrat (2010) tried to define efficiencies for all possible heat pump configurations shown in Figure 2.1. According to them, the only correct way to describe the performance of heat pump cycles is the use of exergy efficiencies, which are shown in the following.

Equation (2.20) shows the exergy efficiency for heating cycles above the ambient temperature (case A). A typical example is an industrial heat pump taking waste heat above the ambient temperature and upgrading it to a higher temperature required by a process demand. This type of heat pump satisfies not only the heat demand of the process, but also the cooling demand of the waste heat.

$$\eta_h = \frac{\dot{E}^-_{qH}}{\dot{E}^+ + \dot{E}^+_{qH'}} \tag{2.20}$$

Equation (2.21) shows the exergy efficiency for heating cycles with the cold source at the ambient temperature (case B). Typical examples are given by domestic heat pumps or by the EPFL heat pump supplying heat to the university, using the Geneva lake as heat source.

$$\eta_h = \frac{\dot{E}^-_{qH}}{\dot{E}^+} \tag{2.21}$$

Equation (2.22) shows the exergy efficiency for heating cycles and refrigeration cycles, with the hot source above the ambient temperature, and the cold source below the ambient temperature (case C). As examples, refrigeration cycles with the use of the condensation heat can be mentioned.

$$\eta_{c,h} = \frac{\dot{E}^-_{qH} + \dot{E}^-_{qF}}{\dot{E}^+} \tag{2.22}$$

Equation (2.23) shows the exergy efficiency for refrigeration cycles, with the hot source at the ambient temperature (case D). It is typical example for refrigeration cycles.

$$\eta_c = \frac{\dot{E}^-_{qF}}{\dot{E}^+} \tag{2.23}$$

Equation (2.24) shows the exergy efficiency for refrigeration cycles below the ambient temperature (case E). This is the rare case of cryogenic systems.

$$\eta_c = \frac{\dot{E}^-_{qF}}{\dot{E}^+ + \dot{E}^+_{qF'}} \tag{2.24}$$

In cases A and B, the heat pump is in competition with conventional heating systems like boilers or co-generation units, whereas in cases C, D and E, a cooling demand below the ambient temperature has to be satisfied. There is no direct competition for heat pumps working as refrigeration cycles and thus, they are well developed and successfully installed. On the contrary, heat pump installations for heating still encounter resistances because of profitability or operation risks, even if the energy saving potentials are often promising.

2.2.3 Operating costs, CO_2 emissions and primary energy savings

Heat pump integration will be evaluated by the operating costs savings. Typically, the integration of a heat pump driven with electricity and satisfying a part of the heating demand, results in less fuel consumption but higher electricity consumption. The benefit can be calculated by comparing the results to a reference case. Obviously, this depends strongly on the fuel and electricity prices.

Similarly, the savings in CO_2 emissions and primary energy can be evaluated. In this thesis, the CO_2 emissions will be used as the main environmental indicator. In order to have a complete analysis of the environmental impact, other indicators and life cycle analysis would be necessary.

2.3 Heat pump versus conventional heating systems

2.3.1 Heat pump profitability

In the following, we will calculate the profitability of a heat pump, considering that the heat delivered by the heat pump will substitute the same amount of heat ($\dot{Q}_{th}$), originally supplied by a co-generation system with a thermal efficiency (ϵ_{th}) and an electrical efficiency (ϵ_{el}). Its cost is expressed by Equation (2.25). When a boiler is used, $\epsilon_{el} = 0$.

$$OpC_{cog} = \dot{Q}_{th} \cdot d \cdot \left(\frac{c_f}{\epsilon_{th}} - \frac{c_{el} \cdot \epsilon_{el}}{\epsilon_{th}} \right) \tag{2.25}$$

The operating costs of the heat pump for a process duration d are given in Equation (2.26).

$$OpC_{hp} = \dot{Q}_{th} \cdot d \cdot \left(\frac{c_{el}}{COP} \right) \tag{2.26}$$

Depending on annualized investment costs (as a function of the investment costs $InvC$, expected life time n and interest rate i) and maintenance costs MC, the profitability condition of integrating a heat pump is defined by Equation (2.27). With the equality condition, this equation enables to calculate the break even cost for new investments. The operating costs of a previously needed cooling unit are not included in the approach, as they are insignificant. However, they would increase even more the heat pump interest.

$$InvC \cdot \left(\frac{i(i+1)^n}{(1+i)^n - 1} \right) + MC + \dot{Q}_{th} \cdot d \cdot \left(\frac{c_{el}}{COP} - \frac{c_f}{\epsilon_{th}} + \frac{c_{el} \cdot \epsilon_{el}}{\epsilon_{th}} \right) \leq 0 \tag{2.27}$$

In the same way, it would be possible to define the annualized profit (AP) in Equation (2.28). B is the benefit in operating costs, when the conventional heating system is replaced by a heat pump.

$$\begin{aligned} AP = B - InvC \cdot \left(\frac{i(i+1)^n}{(1+i)^n - 1} \right) - MC = \\ \dot{Q}_{th} \cdot d \cdot \left(\left(\frac{c_f}{\epsilon_{th}} - \frac{c_{el} \cdot \epsilon_{el}}{\epsilon_{th}} \right) - \frac{c_{el}}{COP} \right) - InvC \cdot \left(\frac{i(i+1)^n}{(1+i)^n - 1} \right) - MC \geq 0 \end{aligned} \tag{2.28}$$

The factor k_{cost} is introduced as the ratio of electricity and fuel price (Equation (2.29)), and Equation (2.27) is transformed in Equation (2.30).

$$k_{cost} = \frac{c_{el}}{c_f} \tag{2.29}$$

$$InvC \cdot \left(\frac{i(i+1)^n}{(1+i)^n - 1} \right) + MC + \frac{\dot{Q}_{th} \cdot d \cdot c_f}{\epsilon_{th}} \left(k_{cost} \left(\frac{\epsilon_{th}}{COP} + \epsilon_{el} \right) - 1 \right) \leq 0 \tag{2.30}$$

The profitability of the heat pump is therefore defined by Equation (2.31). It depends on the investment costs and the way it is annualized (varying with expected life time n and interest rate i), the maintenance costs, the fuel to electricity price ratio, the COP of the system and the efficiencies ($\epsilon_{th}, \epsilon_{el}$) of the present heating system (e.g. co-generation engine or boiler).

$$\frac{InvC \cdot \left(\frac{i(i+1)^n}{(1+i)^n - 1} \right) + MC}{\left(1 - k_{cost} \left(\frac{\epsilon_{th}}{COP} + \epsilon_{el} \right) \right)} \leq \frac{\dot{Q}_{th} \cdot d \cdot c_f}{\epsilon_{th}} \tag{2.31}$$

The right term of Equation (2.31) corresponds to the present fuel costs of the system.

2.3.2 Savings potentials

From Equation (2.30) the operating cost savings related to the fuel and electricity consumption can be deduced.

$$\Delta OpC = \left(\frac{c_f}{\epsilon_{th}} - \frac{c_{el} \cdot \epsilon_{el}}{\epsilon_{th}} - \frac{c_{el}}{COP}\right) \dot{Q}_{th} \cdot d \tag{2.32}$$

Equation (2.33) gives the relative operating cost saving potential as a function of k_{cost}.

$$\Delta OpC_{rel} = \left(1 - k_{cost}(\frac{\epsilon_{th}}{COP} + \epsilon_{el})\right) \tag{2.33}$$

The operating costs saving potential is increasing with a lower thermal efficiency, a higher COP and a lower fuel to electricity price ratio (k_{cost}).

Considering the specific CO_2 emissions and primary energy of both electricity and fuel, the heat pump integration may result in considerable primary energy and CO_2 emissions savings. According to the IEA (International Energy Agency) Heat Pump Center, heat pumps could save up to 5% of the total CO_2 emissions in industry. CO_2 emissions and primary energy savings depend respectively on:

- CO_2 content of the fuel (m_f in kg_{CO2}/kWh_{LHV}); (when not specified, natural gas is used as fuel).
- CO_2 content of the driving energy (m_{el} in kg_{CO2}/kWh_{el}); (here electricity when not specified); It depends strongly on the electricity mix (e.g. CO_2 content of electricity-mix in France: 0.092 kg/kWh_{el}; CO_2 content of electricity-mix in US: 0.745 kg/kWh_{el}).
- Primary energy of the fuel (e_{el_f} in MJ/kWh_{LHV}); (when not specified, natural gas is used as fuel).
- Primary energy ($e_{p_{el}}$ in MJ/kWh_{el}) of the driving energy (here electricity when not specified); as for the CO_2 emissions, the primary energy depends strongly on the electricity mix (e.g. in Switzerland: 8.094 MJ/kWh_{el} and in US: 12.399 MJ/kWh_{el}).

Using heat supplied from a heat pump, the CO_2 saving is expressed by Equation (2.34).

$$\Delta M_{CO_2} = \left(\frac{m_f}{\epsilon_{th}} - \frac{m_{el} \cdot \epsilon_{el}}{\epsilon_{th}} - \frac{m_{el}}{COP}\right) \dot{Q}_{th} \cdot d \tag{2.34}$$

By analogy to k_{cost}, k_{CO_2} is introduced as the ratio of CO_2 content of electricity and fuel. The relative CO_2 emissions reduction is expressed by Equation (2.35).

$$\Delta M_{CO_2 rel} = \left(1 - k_{CO_2}(\frac{\epsilon_{th}}{COP} + \epsilon_{el})\right) \tag{2.35}$$

The interest of heat pumps is therefore increasing with a higher COP, a smaller thermal efficiency (ϵ_{th}) and a lower k-factor for CO_2 emissions.

In the same way, primary energy savings can be introduced by Equations (2.36) to (2.37).

$$\Delta E_p = \left(\frac{e_{p_f}}{\epsilon_{th}} - \frac{e_{p_{el}} \cdot \epsilon_{el}}{\epsilon_{th}} - \frac{e_{p_{el}}}{COP}\right) \dot{Q}_{th} \cdot d \tag{2.36}$$

$$\Delta E_{p_{rel}} = \left(1 - k_{E_p}(\frac{\epsilon_{th}}{COP} + \epsilon_{el})\right) \tag{2.37}$$

The primary energy saving potential is increased with a lower thermal efficiency, a higher COP and a lower primary energy k-factor (k_{E_p}).

Details on the equations and more results are given in Appendix A.1.

2.3.3 Saving potential maps depending on geographical location

The saving potential of a heat pump can be estimated when the temperature levels of the heat pump cycle are known. The COP of a heat pump depends on the temperature lift and the sink temperature as shown in Equation (2.4) and may be estimated considering the Carnot factor (η_{cop} = 55%).

With Equation (2.4), Equation (2.33) becomes Equation (2.38).

$$\Delta OpC_{rel} = \left(1 - k_{cost} \cdot (\frac{\epsilon_{th}}{\eta_{COP}} \cdot \frac{T_{sink} - T_{source}}{T_{sink}} + \epsilon_{el})\right) \tag{2.38}$$

Equations (2.35) and (2.37) can be transformed by analogy. According to IEA (IEA, 2009), Eurostat (Eurostat, 2009) and Ecoinvent (Frischknecht et al., 2005), Table 2.3 shows the energy prices, the CO_2 emissions for the electricity mix and the corresponding primary energy in different countries. To compare, an other reference (ADEME, 2005) indicates a CO_2 content of the electricity mix in France between 0.06 and 0.12 kg/kWh$_{el}$ and a mean value for the electricity mix in Europe of 0.34 kg/kWh$_{el}$.

Table 2.3: Cost (IEA, 2009) (Eurostat, 2009), CO_2 emissions (Frischknecht et al., 2005) and primary energy (Frischknecht et al., 2005) for given electricity mix

	c_f €/kWh	c_{el} €/kWh$_{el}$	m_{el} kg/kWh$_{el}$	$e_{p_{el}}$ MJ/kWh$_{el}$
FR	0.039	0.062	0.092	11.788
DE	0.050	0.108	0.631	10.945
CH	0.036	0.069	0.113	8.094
US	0.018	0.052	0.745	12.399

The CO_2 content of natural gas is considered to be 0.202kg/kWh and the primary energy for natural gas has a value of 4.5MJ/kWh. Table 2.4 defines the corresponding k-factors.

Table 2.4: k-factors

	k_{cost}	k_{CO_2}	k_{E_p}
FR	1.58	0.46	2.62
DE	2.16	3.12	2.43
CH	1.92	0.56	1.80
US	2.89	3.69	2.76

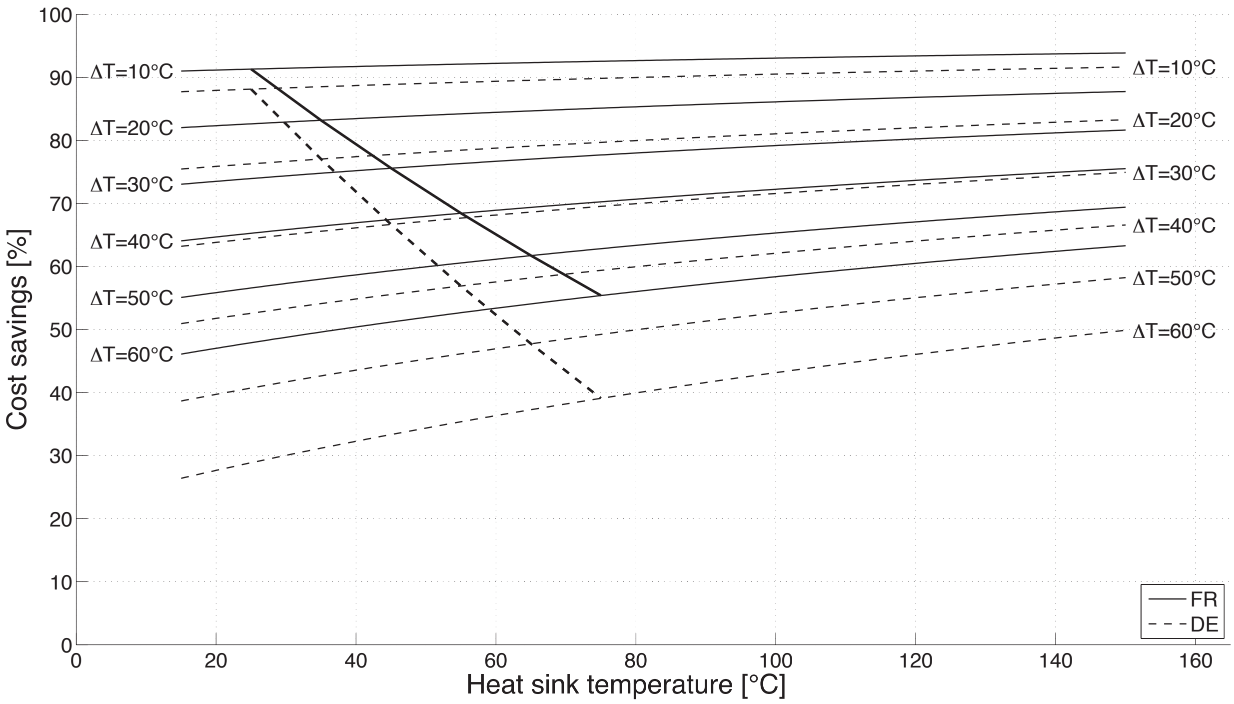


Figure 2.5: Relative operating cost savings (Equation (2.38) / (2.33)) as a function of the heat sink temperature and the temperature lift (ΔT)

The equations for relative CO_2 emissions and primary energy savings and the complete graphical maps for France, Germany, Switzerland and US are presented in A.2. Figures 2.5 and 2.6 show the relative saving potential for operating costs and CO_2 emissions for France and Germany. The ambient temperature is supposed to be 15 °C. The bold curves separate the case where the heat source is below the ambient temperature (on the left) from the case where it is above ambient temperature (on the right); ($\Delta T = T_{sink} - T_{source}$).

Having a lower k-factor regarding the operating costs and the CO_2 emissions, the saving potential is higher for a heat pump installed in France than for example in Germany. However heat pumps in Germany are also promising when regarding the potential of operating cost savings. Considering in a second step the CO_2 emissions, the interest of heat pumps is significantly higher in France, due to the low CO_2 content of the French electricity, which is mainly produced by nuclear power

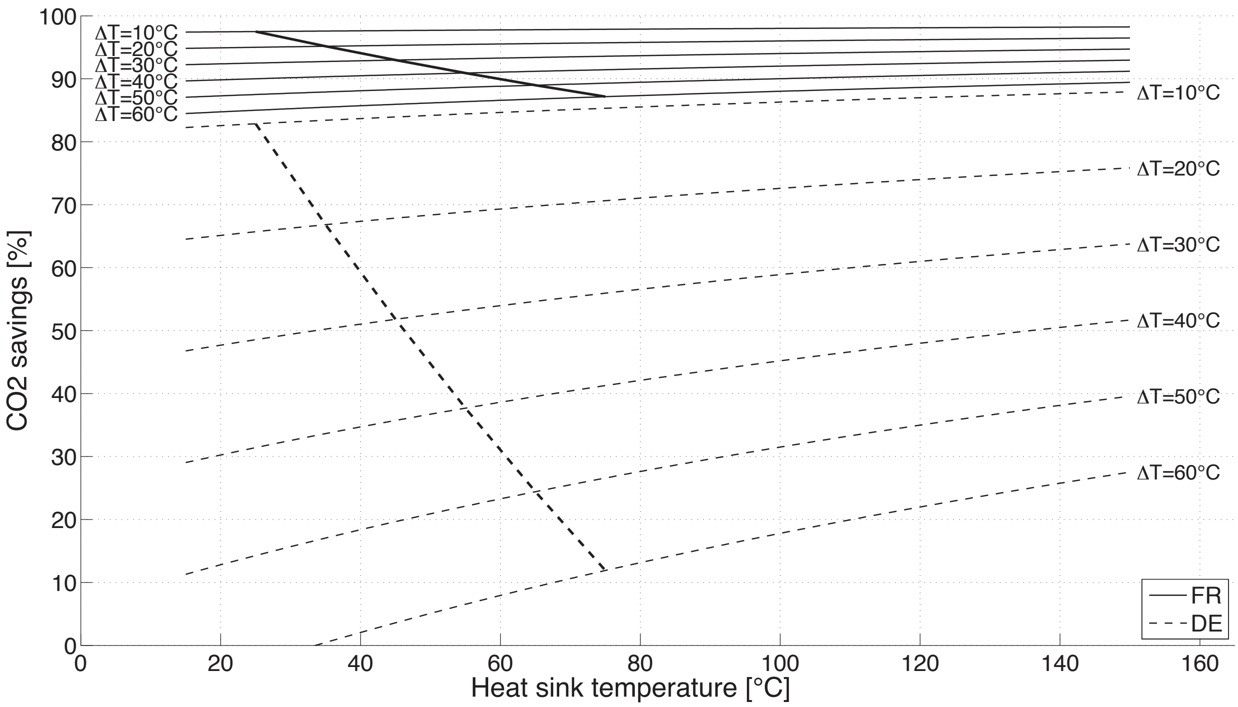


Figure 2.6: Relative CO_2 emission savings (Equation (2.35)) as a function of the heat sink temperature and the temperature lift (ΔT)

plants. Other environmental impacts of nuclear power plants have not been taken into account.

Generally speaking, heat pumps can be considered as a promising technology and even in countries with high k-factors, big saving potentials are shown. For example in the US, correctly integrated heat pumps can save more than 80% of operating costs, CO_2 emissions and primary energy.

2.4 Evaluation of concrete projects or existing installations

The saving potentials of concrete heat pump projects compared to a co-generation unit can be evaluated in terms of operating costs (Equation (2.32)), CO_2 emissions (Equation (2.34)) and primary energy (Equation (2.36)).

As a first example, the EPFL heat pump is considered (Schmid, 2005). Its cold source, located at a depth of 68 m in lake Geneva, has a quasi-constant temperature between 6 °C and 7 °C all year long. Its hot source between 35 °C and 51 °C (mean temperature 40 °C) is used for heating all the buildings on the university campus and some auxiliary locations. In winter, supplementary gas turbines are needed to provide the required heat demand. Considering a heat pump with a COP of 4.83 and supplying 4.8 MW heat during 8000 hours per year instead of using a boiler ($\epsilon_{th} = 0.9, \epsilon_{el} = 0$) the saving potential can be calculated (Table 2.5).

A mechanical vapour re-compression in a chemical process in Switzerland can be considered as a second example. The heat source is quite high (105 °C) and the heat sink is around 135 °C. The specialty of this installation is that the twin-screw compressor has been installed 25 years ago, and except short maintenance interruption, the unit is still used. The main problem is that spare parts become rare and the same type of compressor is not produced anymore. The heat pump has been installed in 1984 when the high fuel prices promoted heat pump installations. Unfortunately, the technology of that time is not available anymore on the market. With an electrical power of 328 kW and the characteristics of the compressor given by the company, a COP of 8.5 can be estimated. The yearly operating hours are estimated to 8000 hours/year.

Table 2.5: Saving potential of existing installations

	ΔCost [M€/year]	ΔCO_2[Tons/year]	$\Delta E_{pr}[GJ/year]$	COP	η_h
Heat pump EPFL	0.987	7720	128	4.83	0.39
MVR Lonza	0.709	4698	90	8.50	0.90

2.5 Conclusions

Heat pump savings in operating costs, CO_2 emissions and primary energy are shown in the context of different countries. The thermodynamic analysis also defines an indicator of profitability of the heat pumping system by estimating the break even cost for new investments and the annualized profit.

The saving potential of heat pumps has been shown. Knowing the temperature levels of the heat pump, relative saving potential maps give the possibility to rapidly evaluate a heat pump project.

However, in order to reach all the saving potentials presented above, heat pumps need to be placed correctly in a given industrial site. This can be done by applying process integration techniques, which will be shown in the next chapter.

Chapter 3

Process and heat pump integration

This chapter describes the basic methodology of process integration. The importance to consider simultaneously all utility and energy conversion units together with the process units, is shown. In this context, particular attention is given to the integration of industrial heat pumps.
Related publications: Methodology / Case study: Becker et al. (2011a), Becker et al. (2011c).

3.1 Introduction

In the last chapter, a methodology to evaluate heat pump potentials has been presented. For this, the optimal operating conditions of a heat pump (especially the temperature levels) must be known in advance. Thus, the main interest in this chapter is to identify heat pump opportunities and their optimal operating conditions for an industrial process: for this purpose, process integration tools will be used to determine the process heat requirements and the corresponding temperature levels.

The objective is to present the basic methodology and to apply it on case studies. A thermo-economic analysis will be included, that not only takes into account the saving opportunities but also the related investment costs.

3.2 State of the art

3.2.1 Industrial heat pump integration

In the previous chapter, it has been shown that a refrigeration cycle is a also a heat pump cycle. Therefore the literature review is also extended to refrigeration optimization.

One of the first important new insights has been the fact to consider the heat pump as a part of a complete system or industrial process. Instead of searching for local heat pump opportunities, Townsend and Linnhoff (1983a) considered the global system in their approach based on the temperature interval analysis. Also Loken (1985) pointed out that heat pumps have to be considered with the global process, and he preferred to use the ”process effect factor” rather than the ”heat pump effect factor”, also known as the COP. As highlighted by Wallin and Berntsson (1994), characteristics of both industrial process and heat pumps must be taken into account.

It has been demonstrated that the local heat pumping performance is not sufficient to reach the best energy efficiency of a process. Thus, the integration has to be assessed at the system level to fully exploit the heat pump potential.

As for process integration in general, lots of publication about industrial heat pumps integration were written after the energy prices increased drastically in the mid-70s. Due to low energy prices later on, heat pump integration was not anymore demanded, since the main focus was not on the reduction of the energy consumption. Nowadays, heat pumps become interesting again because of raising fuel prices and the increased concern for global warming.

In the following literature review, graphical and mathematical approaches, both based on pinch analysis, can be distinguished.

3.2.1.1 Graphical approaches

The rules to graphically identify the optimal placement of heat engines and heat pump in an industrial process have been introduced by Townsend and Linnhoff (1983a). Figure 3.1 shows that ideally heat pumps upgrade heat from the heat source below the pinch to the heat sink above the pinch point. In other words, only heat pumps across the pinch point are appropriately integrated. From the system integration point of view, heat pumps above the pinch point become electrical heaters of the process and, even worse, heat pumps below the pinch point increase the cold utility, which can be seen as electrical heaters of the environment. Design and selection of appropriate heat engines are described in a second publication (Townsend and Linnhoff, 1983b). Using the grand composite curve, solutions are graphically identified and assessed regarding their practical feasibility. The streams of the heat engines are then added to the problem table to calculate the remaining hot and cold utility. A systematic analysis can become very complex, when considering the combination of several heat engines and/or heat pumps. Moreover, the utility pinch points cannot be detected and therefore not all heat pump opportunities can be exploited. The same graphical approach based on the grand composite curves is used by (Hindmarsh et al., 1985) to understand the heat interaction between the process and heat engines. They demonstrated for example how the choice of evaporation levels in a refrigeration cycle can reduce the electricity consumption of the process.

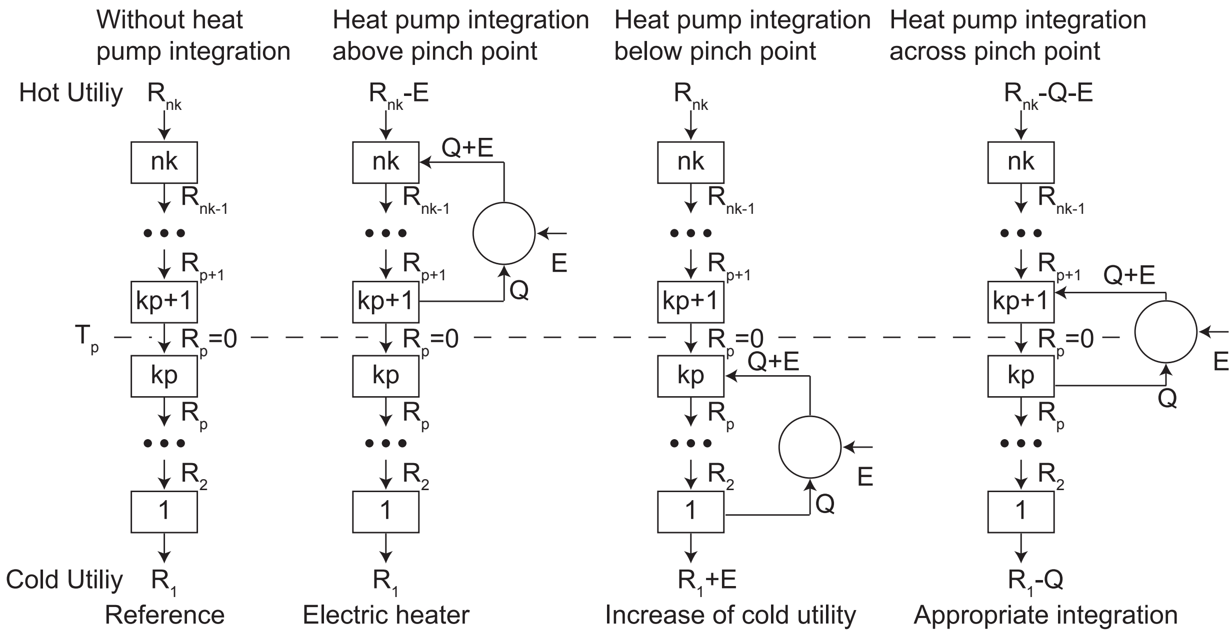


Figure 3.1: Heat pump integration

Based on pinch analysis and on the composite curves, Wallin and Berntsson (1990) proposed a methodology to optimize the temperature levels and the size of heat pumps, by repeating the approach for several minimum temperatures. They pointed out that beside the composite curves, the energy prices and the fuel to electricity price ratio are important factors for the heat pump integration. Therefore, the energy savings related to heat pump integration differ according to the economic context of a given country. They analyzed typical process composite curves on Figure 3.2, where different types of heat pumps are plotted together with the grand composite curves of industrial processes. The synthesis is however incomplete, since absorption systems are tri-therm systems and the utility pinch point normally does not occur twice at the condenser and at the evaporator. No evaporation is needed for mechanical vapour re-compression, because the process stream is directly sent to the compressor to be upgraded to a higher temperature. The complete synthesis of ideal composite curves is shown on Figure 3.3 for heat transformers and absorption heat pumps and on Figure 3.4 for compression heat pumps.

Berntsson (2002) summarized the current situation of industrial heat pumps. He mentioned that heat pumps below or above the pinch can be economically interesting, for example when heat exchange gets expensive due to large distances between streams. This case could correspond to an inappropriate choice of the ΔT_{min}, and shows the need to introduce heat exchange restrictions and intermediate heat transfer as discussed in Chapter 5.

The total site approach first presented by Dhole and Linnhoff (1992) focused on systematic utility integration. Based on this approach, Pavlas et al. (2010) presented heat pump integration for a biomass gasification process, and recently Kapil et al. (2011) proposed a methodology for low grade

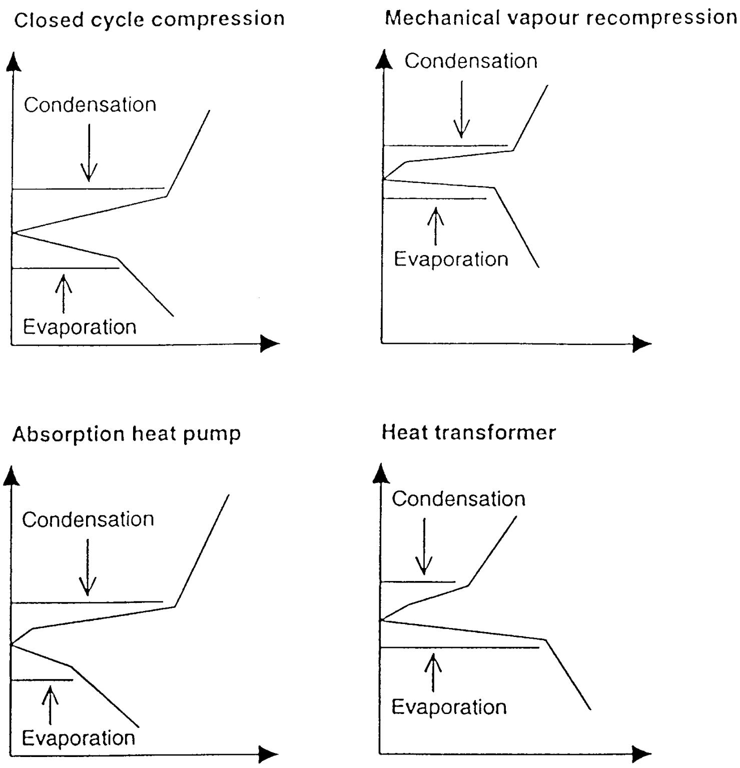


Figure 3.2: Incomplete representation of ideal grand composite curves for heat pump types according to Wallin and Berntsson (1994)

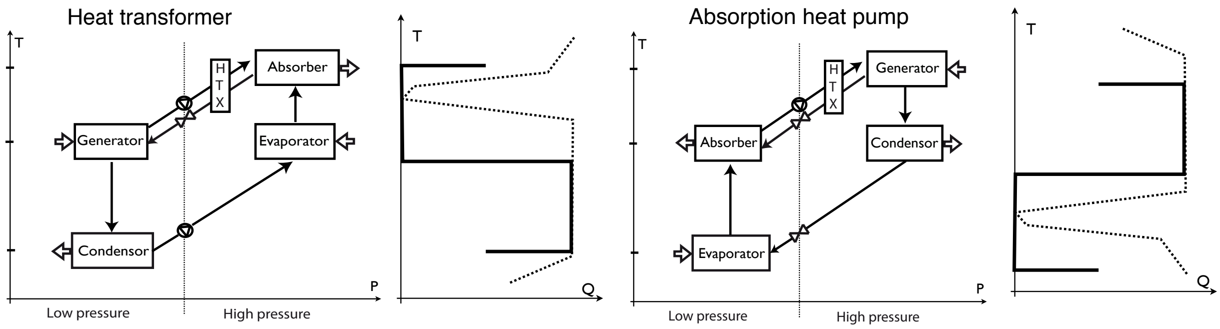


Figure 3.3: Ideal grand composite curves for absorption heat pump and heat transformer

heat recovery, by combining the total site approach with heat recovery units such as heat pumps, organic Rankine cycles or absorption refrigeration. However, this approach has the drawback of not taking into account self-sufficient pockets. Heat pump opportunities are missed, since the potential of self-sufficient pockets for combined heat and power cannot be exploited.

3.2.1.2 Mathematical programming

A computer program, which systematically allows to test heat pump option by adding their fixed streams to the process streams, has been developed by Loken (1985). Total process energy con-

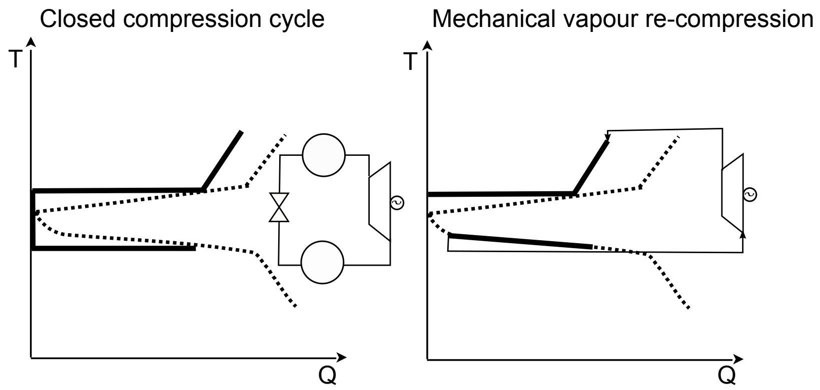


Figure 3.4: Ideal grand composite curves for closed cycle heat pumps and mechanical vapour re-compressions

sumption decreases but the heat pump integration may not be optimal, as utility pinch points may be activated. Thus, a systematic and simultaneous heat pump and utility integration would be necessary.

Later, Shelton and Grossmann (1986a) proposed a mixed integer linear programming model to show the economic potential of properly integrated heat recovery networks and refrigeration systems. They defined a superstructure for refrigeration cycles, which can be included in the structural optimization method presented by Papoulias and Grossmann (1983c). Single and multi-stage refrigeration cycles are integrated with one hot and one cold process stream, the so called ”superhot” and ”supercold” streams that are obtained by summing all hot and cold streams. For a given number of temperature intervals, all possible connections between refrigeration and process are defined. The authors proposed three optimization steps. First, a linear optimization problem to optimize the utility cost is presented. Then, mixed integer linear programming is used to optimize the capital cost or the total annual cost, where the integer variable determines the activation of compressors. The main disadvantage is the complexity when a high number of temperature intervals is required. Finally the heat recovery network can be designed. In the second publication (Shelton and Grossmann, 1986b), the authors presented an enumeration procedure by solving an inner and an outer problem. A selection of active refrigeration levels is fixed in the outer problem, while the inner problem is solving the heat cascade using MILP problem formulations. Problems with a large number of refrigeration levels can be solved, even if the resolution of the problem is complex due to its high combinatorial nature. Using the superstructure introduced by Shelton and Grossmann (1986a), Vaidyaraman and Maranas (1999) accounted for refrigeration cycle configuration and the refrigerant selected from a predefined list. Non-linearities are eliminated and modeling improvements (e.g. eliminating binary variables) are presented.

Combining economics and heat pump integration based on the grand composite curves, Ranade (1988) presented a general equation which defines the maximum economic temperature lift. Profitable heat pump units have to be placed between a minimum evaporation and a maximum condensation temperature, but no systematic approach to integrate heat pump units or/and other utility units has been considered.

Colmenares and Seider (1987) used a non-linear programming strategy for appropriate placement of heat engines and heat pumps. A synthesis of refrigeration systems to define the appropriate placement and the type of refrigerant is given in a second publication (Colmenares and Seider, 1989a). A general formulation for the synthesis of an integrated utility system is presented by the same authors (Colmenares and Seider, 1989b). Using temperatures as decision variables, the problem becomes discontinuous and non-linear. Their model integrates process and utility units simultaneously, by using a two stage optimization similar to the enumeration procedure of Shelton and Grossmann (1986b). The outer problem optimizes temperatures and pressures, while the inner problem optimizes the flow-rates of working fluids. Depending on the linearity of the objective function, the inner problem can be solved by linear programming.

Swaney (1989) proposed a transportation model to determine the optimal heat load of heat pumps by using fixed temperature levels. The heat loads are optimized without specifying a priori a detailed configuration before.

A mathematical formulation for the optimal integration of heat pumps is presented by Holiastos and Manousiouthakis (2002). The formulation evaluates the minimum hot/cold/electric utility cost for heat exchange networks. The approach includes a global heat pump/ heat engine network and additional hot and cold utility costs. They found an optimal case, where the heat pump does not cross the pinch point. The corresponding example integrated a combination of heat pumps and heat engine and this could be explained by the activation of utility pinch points.

Maréchal et al. (2002) have developed a tool to optimize the integration of refrigeration systems. Using a MILP formulation, this approach optimizes the flow-rates in refrigeration systems. The utility integration and combined heat and power integration are considered simultaneously. Therefore, different utilities can be evaluated in the same problem. Compared to previous approaches, the utility flow-rates can be calculated without defining explicitly all possible heat exchanger network connections.

Bagajewicz and Barbaro (2003) considered temperature levels as decision variables. A fine interval partition enables to find good solutions but can also give non-realistic ones, due to the fact that generally, industrial heat pumps only have one condenser and one evaporator. They also made a difference between assisting heat pumps (situated above or below the pinch) and effective heat pumps (situated across the pinch point). Economically, an assisting heat pump in combination with an effective heat pump can be optimal, which demonstrates the interest of integrating multi-levels heat pumping systems.

3.2.1.3 Heat pump integration case studies

Case studies, demonstrating heat pump integration, are also presented in the literature. Söylemez (2005) studied the integration of a heat pump in a dairy. However, the author focused on the local integration of the pasteurization unit without considering the global process. Although the solution appears to be locally optimal, there is no proof that this solution is optimal from the system point of view, since the system pinch point location and the global heat recovery options are not considered.

Multi-objective optimization has been used to optimize heat pump integration in a district heating network (Molyneaux et al., 2010) and for heat pump integration in a cheese factory (Murr et al., 2011). It has been shown that heat pump integration decreases the energy consumption. Nevertheless, process integration is not considered and optimal heat pump opportunities may be missed. Moreover heat pump integration is not compared to the case with optimal heat recovery.

Heat pump integration based on pinch analysis is also presented in the literature (e.g. distillation columns (Rivera-Ortega et al., 1999), slaughter and meat processing (Fritzson and Berntsson, 2006), cheese production plant (Kapustenko et al., 2008)).

3.2.2 Process modifications

Although the process recipe and its main constraints are kept, it could be possible to change the operating conditions of a process system. Thus, before analyzing the possible heat pump integration opportunities, it can be interesting to analyze if process modifications leads to a better heat recovery potential. For example, Périn-Levasseur et al. (2008) have analyzed the integration of heat pumps in a multi-effect evaporators system. Before calculating the size of the heat pumps, they first applied the "plus-minus" principle of the pinch analysis (Kemp, 2007) to identify the modifications of the operating conditions which maximize the heat recovery potential in the system. They also suggested to carefully analyze the value of the minimum temperature difference between hot and cold streams, that could make a heat exchange feasible. As a last step, they proposed a three level heat pumping system (here mechanical vapour re-compression), in which the optimal flows are calculated using a mixed integer linear programming formulation. The system analysis showed that only a part of the available heat load has to be pumped and that the heat pumps deliver their energy savings only if they are considered simultaneously. This study only focused on the multi-effect evaporators systems but did not consider the whole system, which could lead to miss leading solutions as discussed by Périn-Levasseur (2009).

Also for Gundersen et al. (2009), pressures and temperatures of process streams became design variables in order to optimize the heat recovery system. Their research was extended to the appropriate placement of re-compression and expansion in the process system. For example, process streams can be used in open cycle heat pumps to improve hot and cold utility, by varying the inlet temperature of the compressor. Therefore the distinction between process and utility streams is not evident anymore.

Maréchal et al. (2002) suggested to modify the pressure levels of process streams in order to optimize the combined heat and power integration. Especially, the integration of the refrigeration cycle is considered, which leads to significant energy savings of the system.

3.2.3 Exergy analysis and process integration

The first law of thermodynamics does not take into account the energy quality, whereas the exergy concept, a combination of the first and second law of thermodynamics, accounts for irreversibility and qualifies the energy conversion realized in the system. When heat is exchanged, exergy is always destroyed due to the temperature difference between the hot and the cold stream. A part of these exergy losses could be conserved by considering the possibility to produce work in addition to the heat exchange.

The use of exergy analysis in the process integration studies helps to identify the optimal integration of the energy conversion system. A short literature review of exergy analyis in process integration is given below. Dhole and Linnhoff (1992) have first introduced the exergy composite curves. Wall and Gong (1995) considered an exergy concept in addition to pinch technology for optimizing heat pump integration.

Staine and Favrat (1996) included exergy factors to process integration. For this, they proposed a graphical representation method to show the main exergy losses, which is particularly useful when introducing heat pumps or co-generation units. Based on this approach later Maréchal and Favrat (2005) discussed the application of exergy concepts to design the optimal energy conversion systems for given processes.

More recently, Wang et al. (2009) pointed out that exergy analysis combined with pinch technology can improve the efficiency of industrial processes, especially when self-sufficient pockets are exploited. However there were no significant new insights and they did not use a systematic approach for process and utility integration.

As described in the previous chapter (Section 2.2.2), the exergy analysis can evaluate the quality of a heat pump cycle (Borel and Favrat, 2010). The exergy ($\dot{E}_q$) of a hot stream delivering heat ($\dot{Q}$) from T_{in} to T_{out} is computed by Equations (3.1) and (3.2) when the specific heat capacity is assumed to be constant.

$$\dot{E}_q = \dot{Q}(1 - \frac{T_a}{T_{lm}}) \tag{3.1}$$

$$T_{lm} = \frac{T_{in} - T_{out}}{ln(\frac{T_{in}}{T_{out}})} \tag{3.2}$$

T_a is the ambient temperature and all temperatures are expressed in degrees Kelvin [K]. The multiplication factor ($\theta = 1 - T_a/T$) is also called the Carnot factor. It is also possible to evaluate the exergy efficiency of a complete system including an industrial process and its utilities.

Figure 3.5 shows the Carnot composite curves of a given process where the Carnot factor is used on the y-axis. Areas in this graph represent the required exergy to be supplied or removed, by hot and cold utilities and the exergy losses in the recovery system that could be potentially valorized in the energy conversion system.

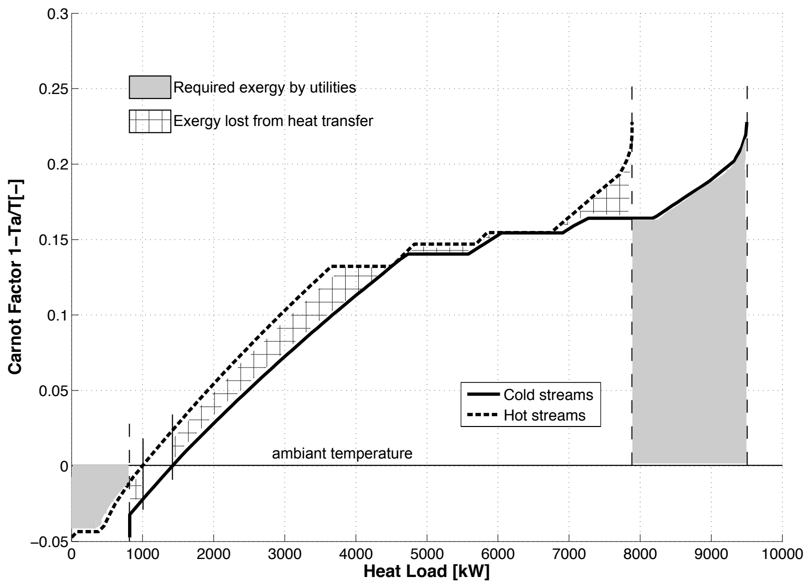


Figure 3.5: Hot and cold Carnot composite curves

The ambient temperature ($\theta = 0$) divides the Carnot composite curves into two parts.

Above the ambiance, the hashed area in this graph represent the exergy delivered by the hot streams in the systems minus the exergy required by the cold stream in the system. Below the ambiance, the hashed area represents the exergy delivered by the cold streams minus the exergy required by the hot streams. It represents therefore the exergy losses in the heat recovery system. This represents as well the exergy available in the self-sufficient pockets of the system (Figure 3.6). Exergy losses of a heat exchange between hot streams above and cold streams below the ambient temperature (white area between hot and cold composite curves on Figure 3.5) have to be considered separately since both the hot and the cold streams deliver exergy.

In the representation of Figures 3.5 and 3.6, the Carnot factor is calculated using the corrected temperature. This means that the exergy losses associated with the ΔT_{min} assumption are already considered as being lost. Smaller ΔT_{min} values decrease the exergy losses but on the contrary the investment costs related to heat exchangers will increase.

Above the ambient temperature, the area between the x-axis (positioned at y=0) and the cold composite curve represents the minimum exergy required to heat up the cold process streams. This requirement corresponds to the work that would be consumed by reversible heat pumps to heat up the cold process streams using the ambient heat as cold source. By analogy, a reversible Rankine cycle would deliver work, using the cold streams as hot source and the environment as cold source.

Below the ambient temperature the exergy required by the hot streams is equal to the work required by a reversible heat pump (or refrigeration) cycle. And by analogy the exergy delivered by the cold streams can be represented as the work of a reversible Rankine cycle with the environment as hot source and the hot streams as cold source.

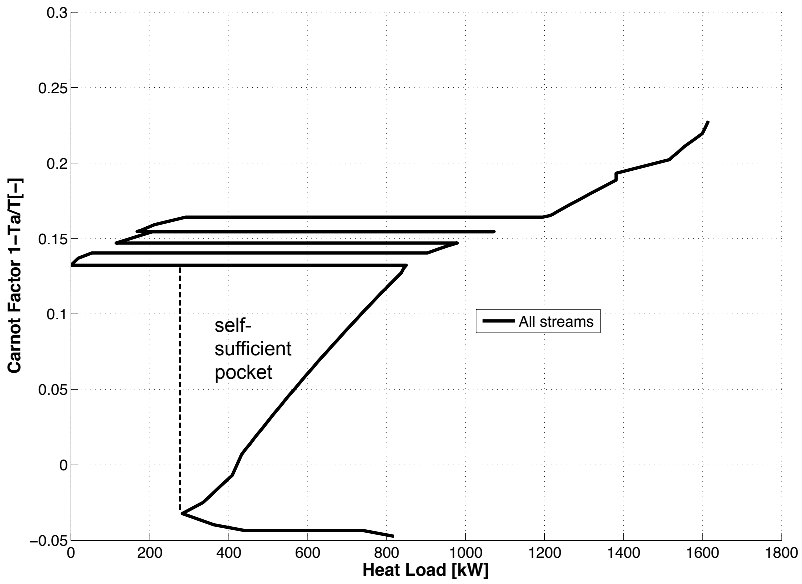


Figure 3.6: Carnot grand composite curves

3.3 Synthesis of the state of the art

The methodology proposed in this work is based on the heat cascade formulation of Maréchal et al. (2002). A two stage approach with a targeting and synthesis step is applied. First, the flows of the utility streams and their operating conditions are optimized applying the concept of the heat cascade and without considering the heat exchanger network design problem. The optimization can be realized by minimizing the operating costs or the total costs including an estimation of the annualized investment costs of new installations. The utility streams are optimized in this step without using a superstructure as proposed by Shelton and Grossmann (1986a). The problem size is reduced but the heat exchanger network is important for the final setup. Therefore, in the synthesis step the heat load distribution based on the fixed process and utility streams is first optimized with a MILP problem formulation. The activated connections between hot and cold streams and their exchanged heat quantities are resulting. With this, the heat exchanger network could be designed, but a non-linear optimization becomes necessary to define the appropriate operating conditions (in and outlet temperature and flow-rates) of each heat exchanger.

The method allows to solve the problem of heat pump integration, together with the integration of other energy conversion units. Heat pumps and other utility units are therefore optimized at the same time and their respective flow-rates are defined by activating the appropriate utility pinch point in order to minimize the objective function. Since the utility mass flow-rates are optimized simultaneously, new insights are discovered with this method. When heat pumps and refrigeration cycles are combined, heat pumps can be integrated above and below the pinch point due to activation of utility pinch points. Self-sufficient pockets can be better exploited and the exergy efficiency of a process system can be increased.

As shown in the literature, the problem becomes non-linear when the evaporation and condensation temperatures of a heat pump are part of the decision variables. In a first step, as presented below, the temperature will be fixed manually, by analyzing the shape of the grand composite curve. In a second step, the temperature levels could also be optimized by discretization or by using a multi-objective non-linear optimization strategy based on an evolutionary algorithm (Dumbliauskaite et al., 2010). Chapter 4 will show a systematic heat pump integration based on a master and slave problem decomposition (outer and inner problem), where the master problem will be solved by a multi-objective optimization. The temperature levels will be used as decision variables of the master problem, while the flow-rates will be calculated by solving the linear heat cascade in the slave problem.

Process modifications can increase the energy and exergy efficiency of an industrial process, but also the complexity of the problem. Process modifications are suggested with the example of an evaporation unit in Section 3.5.2.2, which is particularly interesting when mechanical vapour re-compression is integrated.

Exergy analysis is also useful when integrating heat pumps and co-generation engines. As said before, the exergy efficiency will be used in the following to evaluate the quality of heat pump cycles but also the quality of the whole process.

The approach described below has already been successfully applied to several case studies: a dairy process (Becker et al., 2011a) given in Appendix B.1, a cheese factory (Becker et al., 2011d) given in Appendix B.2 and a brewery process (Dumbliauskaite et al., 2010) given in Appendix B.3.

3.4 Methodology

Figure 3.7 summarizes the applied methodology. Because heat pumps are often integrated in processes using water, like food or pulp and paper industry, a combined water and energy analysis should be applied.

The first step is the data collection. It consists in defining the temperature - enthalpy profile of the heat transfer requirements and the quality - flow-rate profile of the water usage of the process unit operations. Considering the system boundaries, a lot of heat is found in the effluent streams which are systematically cooled down to the ambient temperature, in order to recover this heat.

In the second step, the heat cascade is solved and the maximum heat recovery and the minimum energy requirement in hot and cold utility is calculated. By analogy, the minimum water demand can be determined.

Utility and more particular heat pump units are proposed to the process in the third step. This targeting step identifies optimal utility units and optimizes their flow-rates. Finally, the saving potential can be analyzed by doing a thermo-economic analysis.

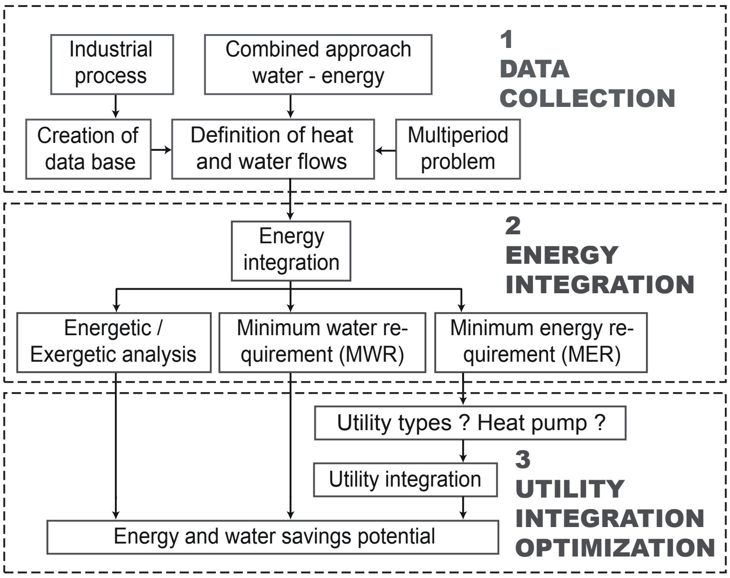


Figure 3.7: Heat pump integration methodology

3.4.1 Identification of process unit operations

After having defined the system boundaries, the first step is to identify the most significant energy consuming process units. Starting from the global energy bill, the relevant process operations for the energy integration are identified and for each identified process operation its contribution to the global energy is calculated. Thus, step by step the global energy consumption can be explained. The non-identified part of the energy bill can be represented in the utility representation (see Section 3.4.2), since the corresponding process operation units could not be identified or have an insignificant energy contribution. At this stage of the analysis, it is important to concentrate only on process unit operations with a significant energy consumption, in order to keep the problem as simple as possible and not to waste time for modeling.

The overall process including auxiliary units (e.g. hot water production, heating, ...) has to be considered. Any type of industrial process can be represented with Figure 3.8.

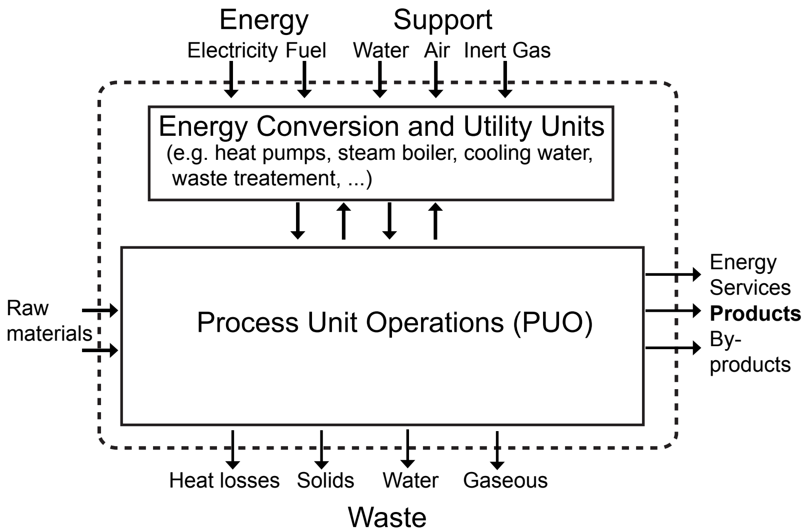


Figure 3.8: Site scale process schematic representation

Considering first the horizontal streams, raw materials enter the system, are transformed by several process unit operations and finally leave the system either as products, byproducts or energy services. The vertical streams represent the driving force of the process, (e.g. the utility and support streams) to satisfy the process demand. For this, energy in the form of electricity or fuel and support (e.g. water) enters the system. These input energy resources are then converted in the energy conversion and distribution system into useful process energy. On the other side, waste heat and/or waste water is leaving the system. Waste water and waste heat treatment may require energy and has therefore to be considered as a part of the system. Keeping the horizontal streams as they are, the main goal of process integration is to minimize the energy and support flows and the waste production which correspond to minimize the vertical streams of Figure 3.8.

3.4.2 Process requirements and their different representations

The applied approach includes the process analysis methodology of Muller et al. (2007). Once the process unit operations are identified, they are modeled in order to define their enthalpy temperature profiles of hot and cold streams for the process integration. Furthermore, all process and waste streams leaving the system are systematically cooled down to the ambient temperature, in order to maximize the heat recovery potential of the system.

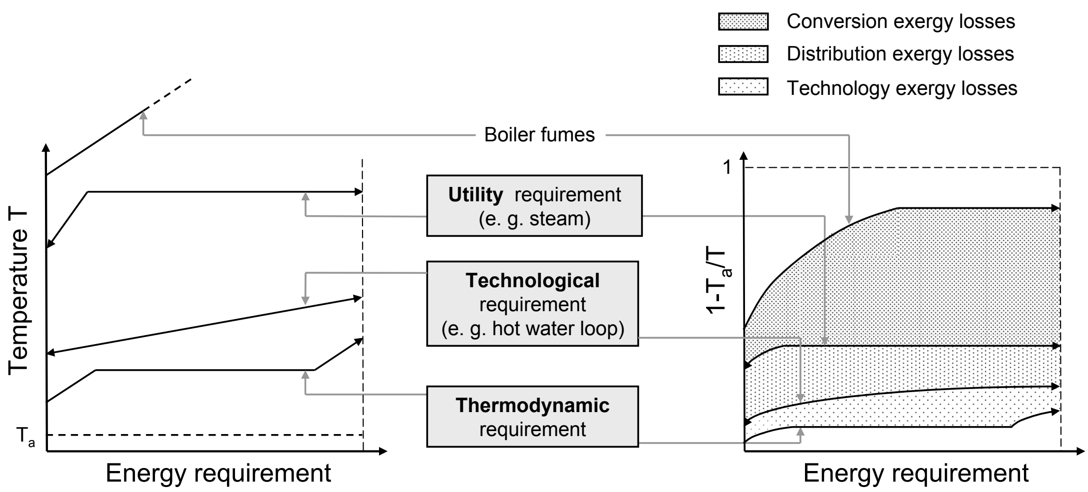


Figure 3.9: Triple representation of a process requirement Muller (2007)

According to different levels of process modifications that could be realized, Figure 3.9 summarizes the possible representations to define the heat requirements of the process unit operations.

- The thermodynamic representation corresponds to the process heat requirements where the hot streams have the highest possible temperature and the cold streams the lowest possible temperature. The enthalpy temperature profile results from the thermodynamic analysis of the process unit operation. This representation is chosen when the process technology and heat supply can be modified.

- The technology representation defines hot and cold streams as they currently exist in the process. The unit is modeled as a black-box and only the heat supply through heat exchangers can be modified, but not the process unit itself.

- The utility representation corresponds to the process heat requirements by defining their hot and cold utility streams. In this case neither the process nor the utility supply can be modified.

With the assumption of no heat losses, the representations are characterized by the same amount of heat, but with different temperature profiles. The Exergy losses can be visualized in Figure 3.9 on the right, using the Carnot composite curves (Staine and Favrat, 1996).

3.4.3 Hot and cold streams definition

For each process stream, the in- and outlet temperatures, the heat load (enthalpy temperature profile) and the $\Delta T_{min}/2$ have to be defined to be able to solve mathematically the heat cascade (cf. Section 1.3.1). The corrected temperatures are obtained by Equations (3.3) and (3.4) for hot and cold streams respectively.

$$T_h^* = T_h - \frac{\Delta T_{min}}{2} \qquad \forall h \in \{hot\ streams\} \tag{3.3}$$

$$T_c^* = T_c + \frac{\Delta T_{min}}{2} \qquad \forall c \in \{cold\ streams\} \tag{3.4}$$

Process integration was originally developed for continuous problems. But since many processes have non-continuous process operations, Linnhoff et al. (1988) introduced the time slice model and the time average model to be able to solve multi-period problems. The stream definition and some required adaptations for three particular cases are described below: continuous problems (Section 3.4.3.1) and non-continuous problems using the time slice approach (Section 3.4.3.2) or the time average approach (Section 3.4.3.3).

3.4.3.1 Continuous problems

The heat loads are defined with their instantaneous values, which are constant during the considered period.

$\Delta T_{min}/2$ can be defined by the observed temperature difference in the existing heat exchangers or by estimating an optimal temperature difference, depending on the trade-off between operating and new investment costs (cf. Section 1.3.1.2).

3.4.3.2 Time slice approach

In this approach, the process production can be divided into several operating periods with simultaneous operations. Thus, for each time slice the heat cascade can be calculated separately. The utility units can only be optimized separately for one time slice, but solving the complete problem including the heat recovery between time slices through storage tanks needs a more complex approach. This problematic will be treated in Chapter 6.

The heat loads are defined with their instantaneous values of the considered time slice.

Since instantaneous values are used, the $\Delta T_{min}/2$ can be defined in the same way as for continuous problems (Section 3.4.3.1).

3.4.3.3 Time average approach

The time average approach uses average values over all periods and is valid when making the assumption that the heat recovery is realized by using storage tanks. It could therefore be possible to define the process streams with their mean heat loads $\dot{Q}_{h/c}$ over all periods (Equation (3.5)).

$$\dot{Q}_{h/c} = \dot{Q}^{*}_{h/c} \frac{d_u}{d} \tag{3.5}$$

$\dot{Q}^{*}_{h/c}$ is the instantaneous heat load of a hot or cold stream belonging to unit u with an effective operating time d_u. d is the total operation time.

Another possibility to avoid multi-period problems is to express the energy consumption by a mean value related to the energy consumption of 1 ton of the final product (kWh/t). The specific heat loads are defined by Equation (3.6). q_u is the specific heat load of unit u, related to its instantaneous heat load $\dot{Q}^{*}_{u}$ during the operating time d_u. M_p is the product quantity produced during the total operation time.

$$q_{h/c} = \dot{Q}^{*}_{h/c} \cdot \frac{d_u}{M_p} \tag{3.6}$$

Heat exchanges become more difficult for multi-period processes as they are indirect through heat transfer units and storage tanks. Furthermore, the $\Delta T_{min}/2$ has to be adapted to include the fact, that the real instantaneous heat loads ($\dot{Q}^{*}_{h/c}$) will be higher than the mean heat loads ($\dot{Q}_{h/c}$). Thus, a higher heat exchange area for the same heat amount will be required.

If $\Delta T_{minref}/2$ is the minimum temperature difference which corresponds to the optimum trade-off in terms of operating and investment costs for the case when streams are simultaneous, the new value of $\Delta T_{min}/2$ will be calculated by considering the instantaneous heat load instead of the mean heat load. Assuming the same heat exchanger area, the new $\Delta T_{min}/2$ is obtained by Equation (3.7).

$$\Delta T_{minu}/2 = \frac{\dot{Q_{h/c}}^{*}}{UA} = \frac{\dot{Q}^{*}_{h/c}}{\dot{Q}_{h/c}} \Delta T_{minref}/2 \tag{3.7}$$

Combining Equations (3.5) and (3.7), the $\Delta T_{minu}/2$ can be approximated with Equation (3.8).

$$\Delta T_{min_u}/2 \simeq \frac{d}{d_u} \Delta T_{min_{ref}}/2 \tag{3.8}$$

3.4.4 Mathematical formulations

3.4.4.1 Heat cascade

This MILP formulation proposed by Maréchal and Kalitventzeff (1998a) solves the heat cascade and calculates the maximum heat recovery together with the optimal integration of energy conversion units by minimizing the cost of supplying the heat and cold requirements of the process. The combined heat and power production is therefore maximized. The objective is to minimize the operating costs (Equation (3.9)).

$$F_{obj} = min(d \cdot (\sum_{f=1}^{nf}(c_f^+ \sum_{u=1}^{nu} \boldsymbol{f_u} \dot{E}_{f,u}^+) + c_{el}^+ \dot{\boldsymbol{E}}_{\boldsymbol{el}}^+ - c_{el}^- \dot{\boldsymbol{E}}_{\boldsymbol{el}}^- + \sum_{u=1}^{nu} \boldsymbol{f_u} c_u)) \tag{3.9}$$

Equation (3.10) states that the total electricity import and the electricity produced by the process have to be greater or equal to the process electricity consumption. The overall electricity balance is given by Equation (3.11). Both equations are necessary to distinguish the price for electricity import and export.

$$\sum_{u=1}^{nu} \boldsymbol{f_u} \dot{E}_{el,u}^+ + \dot{\boldsymbol{E}}_{\boldsymbol{el}}^+ - \sum_{u=1}^{nu} \boldsymbol{f_u} \dot{E}_{el,u}^- \geq 0 \tag{3.10}$$

$$\sum_{u=1}^{nu} \boldsymbol{f_u} \dot{E}_{el,u}^+ + \dot{\boldsymbol{E}}_{\boldsymbol{el}}^+ - \dot{\boldsymbol{E}}_{\boldsymbol{el}}^- - \sum_{u=1}^{nu} \boldsymbol{f_u} \dot{E}_{el,u}^- = 0 \tag{3.11}$$

The corresponding thermodynamical feasibility is ensured by Equation (3.12).

$$\dot{\boldsymbol{E}}_{\boldsymbol{el}}^+ \geq 0 \qquad \dot{\boldsymbol{E}}_{\boldsymbol{el}}^- \geq 0 \tag{3.12}$$

For the electricity cost, c_{el}^+ is the purchase cost and c_{el}^- is the selling price. c_f^+ is the fuel price. $\dot{E}_{f,u}^+$ is the nominal energy (or heating value) delivered to unit u by the fuel (e.g. natural gas) and $\dot{E}_{el,u}$ is the nominal electricity demand $^{(+)}$ or excess $^{(-)}$ of unit u. c_u is the nominal operating cost per hour of unit u (excluding the fuel and electricity costs of unit u). A typical example of c_u is the nominal operating costs corresponding to a water consumption for cooling. Thus, the objective function not only includes the fuel and electricity costs, but also a cooling water consumption or even the possibility of selling heat surpluses, by defining a utility unit with a negative nominal operating cost c_u for a nominal heat surplus at a given temperature. Furthermore, maintenance of fixed operating costs can also be added in the objective function.

In the heat integration model, each process unit, being process or utility unit, is defined with its nominal conditions and requirements. A multiplication factor $\boldsymbol{f_u}$ represents the usage rate of unit u. When it is a process unit the flow-rate is fixed ($\boldsymbol{f_u} = 1$), whereas if it is a utility unit, the flow-rate is variable in a certain range. Thus, each utility unit is defined by a minimal and a maximal multiplication factor (f_u^{min} and f_u^{max}), fixing the lower and the upper bound of the heat load range. Moreover, the associated integer variable $\boldsymbol{y_u}$ defines if the utility unit u is added to process ($\boldsymbol{y_u} = 1$) or not ($\boldsymbol{y_u} = 0$).

$$\boldsymbol{y_u} \cdot f_u^{min} \leq \boldsymbol{f_u} \leq \boldsymbol{y_u} \cdot f_u^{max} \tag{3.13}$$

If necessary, nominal fuel, electricity consumption or electricity surplus (e.g. for heat pumps or steam network) and additional nominal operating costs (e.g. for cooling water) of the corresponding nominal thermal streams can be defined. In the process integration step, the multiplication factors are optimized and the necessary flow-rates, fuel and electricity consumption and additional operating costs for the utility units are calculated.

Without considering heat exchange restrictions, the general heat cascade for each temperature interval k in the corrected temperature domain is given by Equation (3.14), where $\dot{Q}_{h/c,k,u}$ is the nominal heat load of hot or cold stream h/c in temperature interval k and belonging to unit u. $\dot{\boldsymbol{R}}_{\boldsymbol{k}}$ is the cascaded heat from the temperature interval k to the lower temperature intervals.

$$\sum_{h_k=1}^{ns_{h,k}} \boldsymbol{f_u}\dot{Q}_{h,k,u} - \sum_{c_k=1}^{ns_{c,k}} \boldsymbol{f_u}\dot{Q}_{c,k,u} + \dot{\boldsymbol{R}}_{\boldsymbol{k+1}} - \dot{\boldsymbol{R}}_{\boldsymbol{k}} = 0 \quad \forall k = 1..., nk \tag{3.14}$$

$$\dot{\boldsymbol{R}}_{\boldsymbol{1}} = 0 \quad \dot{\boldsymbol{R}}_{\boldsymbol{nk+1}} = 0 \qquad \dot{\boldsymbol{R}}_{\boldsymbol{k}} \geq 0 \quad \forall k = 2..., nk \tag{3.15}$$

Equations (3.9) to (3.15) form the set of equations for the MILP formulation ((Maréchal and Kalitventzeff, 1998a)) without restricted matches. An illustration is given in Figure 3.10.

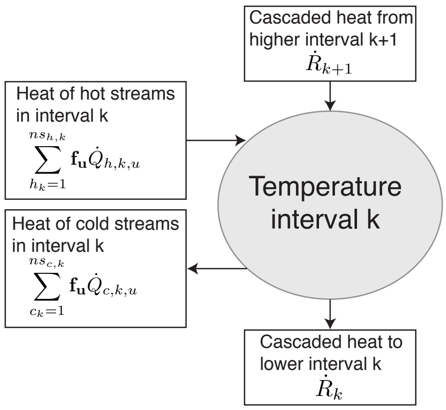


Figure 3.10: Graphical heat cascade representation for MILP formulation without heat exchange restrictions

In this chapter, this formulation is applied to two case studies, where its potential and also its limits will be discussed. The extension of this formulation will be presented in Chapter 5 (heat exchange restrictions) and Chapter 6 (non-continuous process operations). Having well defined the process and its thermal streams, the maximum heat recovery and the minimum energy requirement is calculated. Suitable utilities can be integrated and particular attention will be given to the heat pump integration.

3.4.4.2 Combined energy and water approach

Waste water minimization is important in industrial processes and directly linked to energy efficiency. Linnhoff (1993) already showed the analogy of energy and water integration (Table 3.1).

Combined energy and mass integration has been studied in the literature (Bagajewicz, 2000) (Dunn and El-Halwagi, 2003), using mathematical programming procedures.

In this work, the combined energy and water approach is based on the following principles:

1. The heat recovery is maximized since all effluents will be systematically cooled down to the ambient temperature.
2. The water recovery is determined at ambient temperature considering the water cascade: each process or utility unit can be a water consumer or a water producer. Each water stream (e.g. waste water) has a quality level that defines for instance the concentration of pollutants: the higher the concentration, the lower the water quality. The quality intervals are ordered in a way that streams with a higher quality can always be used to satisfy a water requirement with a lower quality. The analogy between the thermal energy and the water, shown in Table 3.1 enables to write the water cascade equations.

The used approach is a simplified version since it does not allow recycling by mixing waste water and does not take into account multiple contaminant constraints. For this, the approach could be extended with the formulation proposed by Brown et al. (2004).

Table 3.1: Analogy energy - water integration

Thermal Energy cascade	Water cascade
Temperature = energy quality	Index = water quality
Heat load	Water mass flow
Hot stream	Water producer
Cold stream	Water consumer

By analogy to the presented heat cascade in the previous section, Equations (3.19) to (3.20) describe the water or mass cascade without considering the network (or the superstructure) at the targeting stage. However the main disadvantage remains the practical feasibility since the water network should be designed in a next step.

The water cascade consists in water producer (pr) and water consumers (co) and is computed for each water quality index interval. The objective function minimizes the purchase cost of water with a given quality ($\dot{W}^+$) and the cost to evacuate water with a given contamination ($\dot{W}^-$) (Equation 3.16). $\dot{W}_k$ is the cascaded waste water to the lower quality interval k. d is the operating time.

$$F_{obj} = min(d \cdot c_w^+ \dot{\boldsymbol{W}}^+ + c_w^- \dot{\boldsymbol{W}}^-) \tag{3.16}$$

$$\sum_{u=1}^{nu} \boldsymbol{f_u} \dot{W}_u^+ + \dot{\boldsymbol{W}}^+ - \sum_{u=1}^{nu} \boldsymbol{f_u} \dot{W}_u^- \geq 0 \tag{3.17}$$

$$\sum_{u=1}^{nu} \boldsymbol{f_u} \dot{W}_u^+ + \dot{\boldsymbol{W}}^+ - \dot{\boldsymbol{W}}^- - \sum_{u=1}^{nu} \boldsymbol{f_u} \dot{W}_u^- = 0 \tag{3.18}$$

$$\sum_{h_k=1}^{ns_{pr,k}} \mathbf{f_u} \dot{M}_{pr,k,u} - \sum_{c_k=1}^{ns_{co,k}} \mathbf{f_u} \dot{M}_{co,k,u} + \dot{\mathbf{W}}_{\mathbf{k+1}} - \dot{\mathbf{W}}_{\mathbf{k}} = 0 \quad \forall k = 1..., nk \tag{3.19}$$

$$\dot{\mathbf{W}}_{\mathbf{1}} = 0 \quad \dot{\mathbf{W}}_{\mathbf{nk+1}} = 0 \qquad \dot{\mathbf{W}}_{\mathbf{k}} \geq 0 \quad \forall k = 2..., nk \tag{3.20}$$

$\dot{W}_u^+$ is the nominal water consumption ($^+$) or production ($^-$) of unit u. It is important to note that the flow-rates and the multiplication factors (f_u=1) of the process streams are the same for the heat and water integration. Thus, it becomes possible to consider energy and mass minimization in one single problem and to optimize simultaneously the utilities for the heat and the waste water integration.

3.4.4.3 Heat load distribution

The targeting phase defines all flow-rates of the utility streams in the system. Therefore, the complete list of process and utility hot and cold streams is defined, which enables to calculate the heat load distribution by using the formulation presented by Maréchal and Kalitventzeff (1989). It defines for each hot stream the heat that is exchanged with the cold streams. The objective function aims at minimizing the number of connections $\boldsymbol{y_{ij}}$ between hot and cold streams (Equation (3.21)).

$$F_{obj_{hld}} = min(\sum_{j=1}^{ns_c} \sum_{i=1}^{ns_h} \boldsymbol{y_{ij}}) \tag{3.21}$$

Equations (3.22) and (3.23) describe the heat balances of the hot and cold streams. Q_{ijk} is the exchanged heat between hot stream i and cold stream j in temperature interval k. Q_{ik} is the heat load of hot stream i in temperature interval k. And Q_j is the heat load of cold stream j.

Equation (3.24) shows the existence of a connection between hot stream i and cold stream j. Each heat load must be positive (Equation (3.25)).

$$\sum_{j=1}^{ns_c} \boldsymbol{Q_{ijk}} = Q_{ik} \qquad \forall i = 1, ..., ns_h, \forall k = 1..., nk \tag{3.22}$$

$$\sum_{i=1}^{ns_h} \sum_{r=k}^{k_{ui}} \boldsymbol{Q_{ijr}} - \sum_{r=k}^{k_{uj}} Q_{jr} \geq 0 \qquad \forall j = 1, ..., ns_c, \quad \forall k = k_{lj}, ...k_{uj}, \tag{3.23}$$

$$\sum_{k=k_{lij}}^{k_{uij}} \boldsymbol{Q_{ijk}} - \boldsymbol{y_{ij}} \cdot Q_{max_{ij}} \leq= 0 \qquad \forall i = 1, ..., ns_h, \forall j = 1, ..., ns_c \tag{3.24}$$

$$Q_{ijk} \geq 0 \qquad \forall j = 1, ..., ns_c, \forall j = 1, ..., ns_h, \forall k = 1, ..., nk \tag{3.25}$$

The heat load distribution represents the step before designing the heat exchanger network. The latter is not treated in this thesis.

3.4.5 Thermo-economic evaluation & results analysis

In addition to the operating costs, the thermo-economic evaluation requires the estimation of the investment costs. These could be evaluated by an available equipment cost database, based on cost estimations of Ulrich (1984) and Turton et al. (1998). A two stage procedure is applied where the equipments are first sized according to the flow-rates and operating conditions. The investment costs are then deduced by correlations obtained with statistical analysis of similar and known investment costs from the equipment market. These approaches are typically used in the process engineering community to realize the preliminary cost estimation of process development projects.

In the academic research, it is difficult to find realistic and updated investment costs of equipments. Moreover, installation costs are specific to a given process and a given site.

Used in industry, Equation (3.26) gives a simple and recent relationship (base on data from 2010) to roughly estimate the investment costs (InvC) of industrial heat pumps (Vuillermoz and Guillotin, 2011). The sizing parameter used for the cost estimation is the electrical power of the compressor (in kW) and the investment costs is expressed in €. f corresponds to an installation factor with a typical value of 1.5. This factor takes into account installation costs as for example the material of new equipments.

$$InvC_{hp} = f \cdot 1500 \cdot 160^{0.1} \cdot \dot{E}_{hp}^{0.9} \qquad [Euro] \tag{3.26}$$

The investment costs of new co-generation units according to Dumbliauskaite (2009) are given in Equation (3.27). A reciprocating engine fed with natural gas is considered. Its combined heat and power system produces mechanical power and recovers heat from both exhaust gases and cooling water.

$$\begin{aligned} InvC_{cog} = 0.68 \cdot (&-0.0391 \cdot \dot{E}_{cog}^2 + 850.89 \cdot \dot{E}_{cog} + 306016 \\ &+125 \cdot \dot{E}_{cog}^{0.8}) \cdot 1477.7/1068.3 \qquad [Euro] \end{aligned} \tag{3.27}$$

The investment costs related to new heat exchangers can also be taken into account. Once all flow-rates are determined by the MILP optimization, the vertical intervals are built in order to calculate the total heat exchanger area A and the minimum number of connections N_{min}. The mean area is then calculated with Equation (3.28) and the investment is estimated with Equation

(3.29). With an exponent $\gamma = 0.65$, the reference costs ($InvC_{ref}$) is 6300 € for a reference area (A_{ref}) of 1m^2.

$$A_{min} = \frac{A}{N_{min}} \tag{3.28}$$

$$InvC_{hex} = N_{min} \cdot InvC_{ref} \cdot (\frac{A_{mean}}{A_{ref}})^{\gamma} \qquad [Euro] \tag{3.29}$$

The total investment costs can be calculated by summing all investment costs related to new equipments (Equation (3.30)). And the benefit in operating costs (B) can be determined in Equation (3.31) by comparing the new operating costs (OpC) to the current operating costs (OpC_{ref}).

$$InvC = \sum_{hp=1}^{nhp} InvC_{hp} + \sum_{cog=1}^{ncog} InvC_{cog} + InvC_{hex} \tag{3.30}$$

$$B = OpC_{ref} - OpC \tag{3.31}$$

The corresponding payback period PB can be evaluated with Equation (3.32). Maintenance costs are neglected for this estimation.

$$PB = \frac{InvC}{B} \tag{3.32}$$

As the profitability is strongly influenced by electricity and fuel prices, a sensibility analysis for these parameters is necessary.

As already discussed in Chapter 2.3.1, another important indicator for the heat pump profitability is the annualized profit (AP). It is defined by the annual profit B minus the annualized investment costs in Equation (3.33), where i is the interest rate and n is the life-time of the new installation. The profitability is estimated with typical values (e.g. an interest rate of 5% and a life-time over 20 years). However in order to have a global understanding, a sensitivity analysis on these parameters would be necessary.

$$AP = B - InvC \cdot \left(\frac{i(1+i)^n}{(1+i)^n - 1} \right) \geq 0 \tag{3.33}$$

It is also important to note that the increasing energy prices which would lead to a better heat pump profitability are not considered in this equation.

As environmental indicators, the released CO_2 emissions and the primary energy consumption are estimated and compared to a reference case.

3.5 Approach illustrated by case studies

The approach is illustrated by two different case studies. The first application concerns a dairy process. This test case illustrates the complete process integration approach, including the combined heat and water integration. Moreover the modeling of closed and open cycle heat pumps is discussed and a sensitivity analysis on the energy prices is shown. The second application is a cheese factory process and is in particular used to illustrate the importance of process modifications. The case studies are described in detail in Appendix B.1 (dairy) and B.2 (cheese factory).

For the dairy, a typical period (1 day) with continuous and constant process heat loads has been chosen. The complete list of streams is given in Table B.2. For the cheese factory the heat loads have been expressed in kWh per ton of produced cheese. The streams are given in Table B.10.

3.5.1 Dairy process

3.5.1.1 Minimum energy requirement

Knowing the heat flows, the heat recovery potential between hot and cold streams can be identified. Based on the definition of a minimum temperature difference ΔT_{min} (Section 3.4.3.2), the minimum energy requirement is calculated, by introducing a default hot and cold utility when solving the heat cascade formulation (Equations (3.9) to (3.15)).

The composite curves of Figure 3.11 shows three zones: the minimum hot and cold utility requirements and the heat recovery potential.

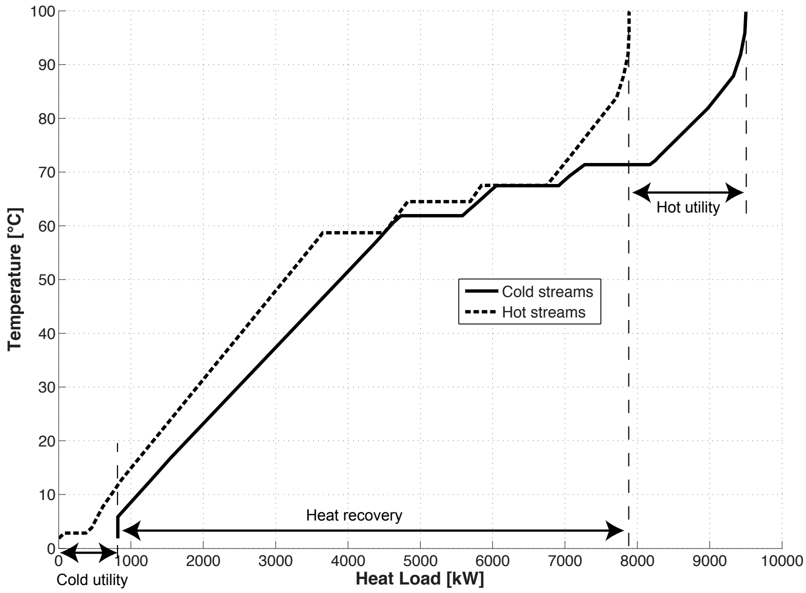


Figure 3.11: Hot and cold composite curves in the corrected temperature domain

The cold utility requirement can be divided into two parts: above the ambient temperature cooling requirements can be realized using a cold utility stream from the environment (e.g water from a river, atmosphere), whereas below the ambiance, the cold utility requires refrigeration. Refrigeration cycles are integrated in the same way than heat pumps, bringing the extracted heat above the ambient temperature or to the ambiance.

Note that the corresponding Carnot composite curves is already given as a previous example in Figure 3.5. Instead of representing temperature on the y-axis, the Carnot factor ($\theta = 1 - T_a/T$) is used. Areas in this graph represent the required exergy to be supplied or removed by hot and cold utilities, and the exergy losses in the heat recovery system, that could be potentially valorized in the energy conversion system (e.g. by a Rankine cycle).

Graphically the maximum heat recovery can be illustrated by shifting the cold composite curves horizontally until the smallest distance between hot and cold composite curves corresponds to the defined ΔT_{min}. Thus, the process pinch point is activated where the Delta T min occurs.

Mathematically, the pinch point can be calculated by solving the heat cascade formulation. The pinch point is located at the intersection of two temperature intervals where the residual heat from the upper interval is 0 ($\dot{R}_k = 0$).

The grand composite curve of Figure 3.12 is obtained when the heat load difference between the hot and cold composite curve is calculated for each temperature (c.f. Section 1.3.1.3). It represents the cascaded heat ($\dot{R}_k$) as a function of the temperature. The pinch point at the corrected temperature of 59 °C separates the process into two parts.

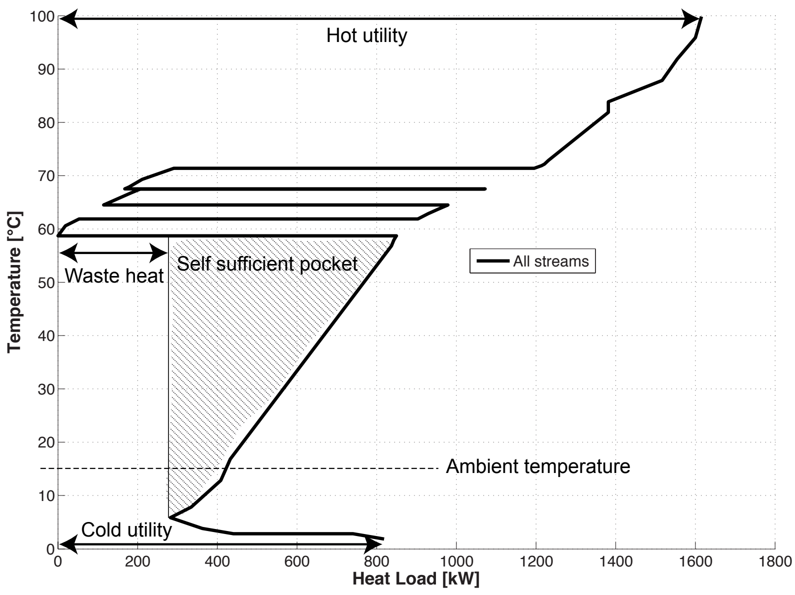


Figure 3.12: Process grand composite curve

Above the pinch point, the process globally requires heat and below the pinch point, the process has globally a heat excess. Following the pinch rules, no hot utility should be integrated below the pinch point and no cold utility above the pinch point. Therefore, theoretically, a heat pump is only appropriately integrated across the pinch point so that a part of the hot and a part of the cold utility are satisfied at the same time.

Based on the graphical analysis of the grand composite curve, the conventional pinch analysis shows that the heat pump could upgrade the waste heat of about 280 kW as indicated on Figure 3.12. Later it will be shown that the simultaneous utility integration will give more heat pump opportunities, by better using the potential of the self-sufficient pocket.

3.5.1.2 Water pinch

For the case study of the dairy, the water pinch has been performed. This section gives a brief description of the application and the major results.

Table 3.2 presents the list of water consumption and production of the process unit operations.

Table 3.2: Water consumption & production

	Water consumptions [kg/s]				Water productions [kg/s]		
	Used water level 2	Used water level 1	Fresh water	Soft water	Dirty	Used water level 2	Used water level 1
Index	50	100	150	200	10	60	110
CIP solution			-0.475		0.475		
CIP hot water		-0.5				0.5	
CIP cold water		-0.55					0.55
Milk evaporation						0.4	
Boiler				-0.26			
Hot water			-1.5		1.5		
Other	-1				1		

It should be noted that the water cascade does not account for the water used in the cooling system. Different levels of water quality have been defined by categorizing the quality requirements (soft water, fresh water, used water). For each consumption, a quality index between 0 and 200 has been attributed. The index of water production is slightly higher to enable water reuse, which corresponds to a similar concept as the introduction of the ΔT_{min} value for the heat recovery concept. This means water with an index of 110 (used water level 1) can satisfy a water consumption with an index 100 or lower.

Figure 3.13 shows the minimum water requirement and it its corresponding water quality.

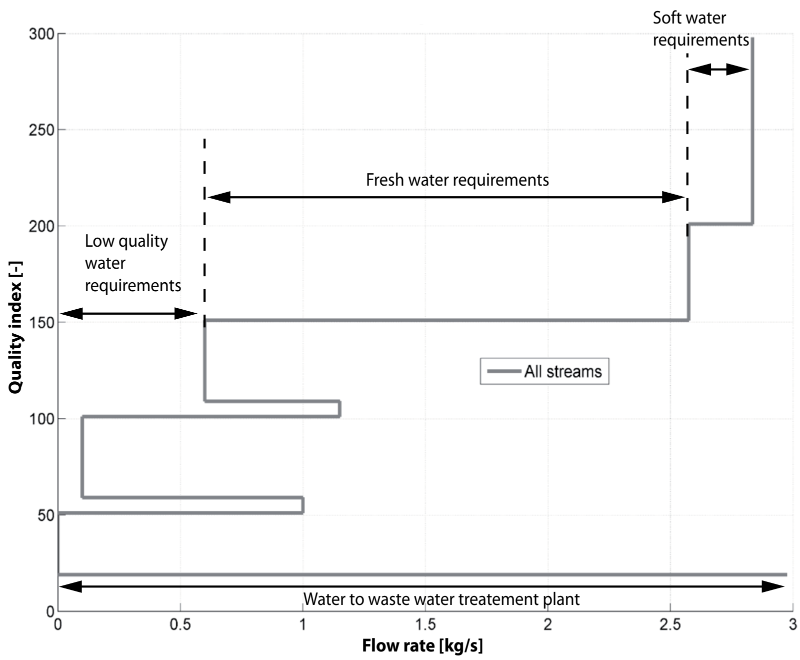


Figure 3.13: Water grand composite curve

In the present situation, the water consumption is about 4.03 kg/s. The calculation of the water cascade shows a minimum water requirement of 2.83 kg/s. The calculation of the water consumption corresponds therefore to a process water saving of 30 %. In addition, the analysis of the water grand composite curve shows that the water requirements can be satisfied by different water quality, (e.g. 0.25 kg/s as soft water, 1.98 kg/s as fresh water and 0.60 kg/s as low quality water).

3.5.1.3 Utility integration and thermo-economic analysis

By analyzing the shape of the grand composite curve in Figure 3.12, the required temperature levels for the hot and cold utilities can be identified. The hot utility demand does not exceed a temperature level of 100 °C. The cooling requirement is basically required at two different temperature ranges: first, waste heat at almost 60 °C is considered as a cooling requirement since all effluents have to leave the system at the ambient temperature and the second cooling demand corresponds to refrigeration demands of the process.

The basic methodology of Section 3.4 will be applied for different scenarios of the dairy in the French context. The utility choice and the operating conditions of the heat pump for each scenario are selected graphically by analyzing the grand composite curve of Figure 3.12.

Case 0 The current situation for the given period is used as reference.

Case 1 Maximum heat recovery is calculated and the on-site available refrigeration cycle and conventional boiler are integrated.

Case 2 On-site available utilities like in Case 1 are integrated. The evaporation temperatures of the refrigeration cycle are adapted to the process cooling demand.

Case 3 As Case 2 but additionally a co-generation engine is integrated.

Case 4 As Case 2 but additionally, a heat pump is integrated.

Case 5 As Case 4 but temperature levels of refrigeration cycle and heat pump are adapted.

Case 6 Heat pump and refrigeration cycles with adapted temperature levels, co-generation and other on site available utilities (e.g. steam boiler) are integrated simultaneously.

The integrated utility curves and detailed results for each case are given in Appendix B.1. The results of all cases are summarized in Table 3.3 and the comparison of operating costs and energy savings is illustrated in Figure 3.14.

Table 3.3: Results of the dairy (Cases 1-6)

	Unit	Case 0	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
OpC	[k€/year]	269	199	195	186	132	128	118
E_f	[MWh/year]	6164	4404	4563	8416	2772	2562	4671
E_{el}	[MWh/year]	443	429	252	-2900	382	439	-1308
Q_{cw}	[MWh/year]	n.a.	2283	2249	2311	767	635	645
M_{CO_2}	[t/year]	1286	929	945	1433	595	558	823
E_p	[GJ/year]	32954	24881	23504	3683	16979	16703	5595
$InvC$	[k€]		316	326	1517	486	489	1291
PB	[year]		4.5	4.4	18.3	3.6	3.5	8.6
AP	[k€/year]		44	48	-39	98	102	47

The operating costs, the fuel, electricity and cooling water consumption as well as CO_2 emissions and primary energy consumption are compared to the current case (Case 0). For France, following values are considered: the fuel price of 0.039 €/kWh, the electricity price of 0.062 €/kWh$_{el}$, and the electricity mix with 0.092 kg/kWh$_{el}$ of CO_2 emissions and 11.788 MJ/kWh$_{el}$ of primary energy (see Table 2.3).

The investment costs and payback period are evaluated by considering the cost of new heat exchangers (Equation (3.29)) and when necessary heat pump and co-generation units (Equations (3.26) and (3.27) respectively). For calculating the annualized profit, the interest rate is supposed to be 5% and the life time of new installations is considered to be 20 years. Already on site available heat exchangers are not accounted. It is important to note that the payback period is only evaluated for the savings of a specific time slice. The new installed heat pump or co-generation units could also be useful in other time slices and the payback time can be decreased.

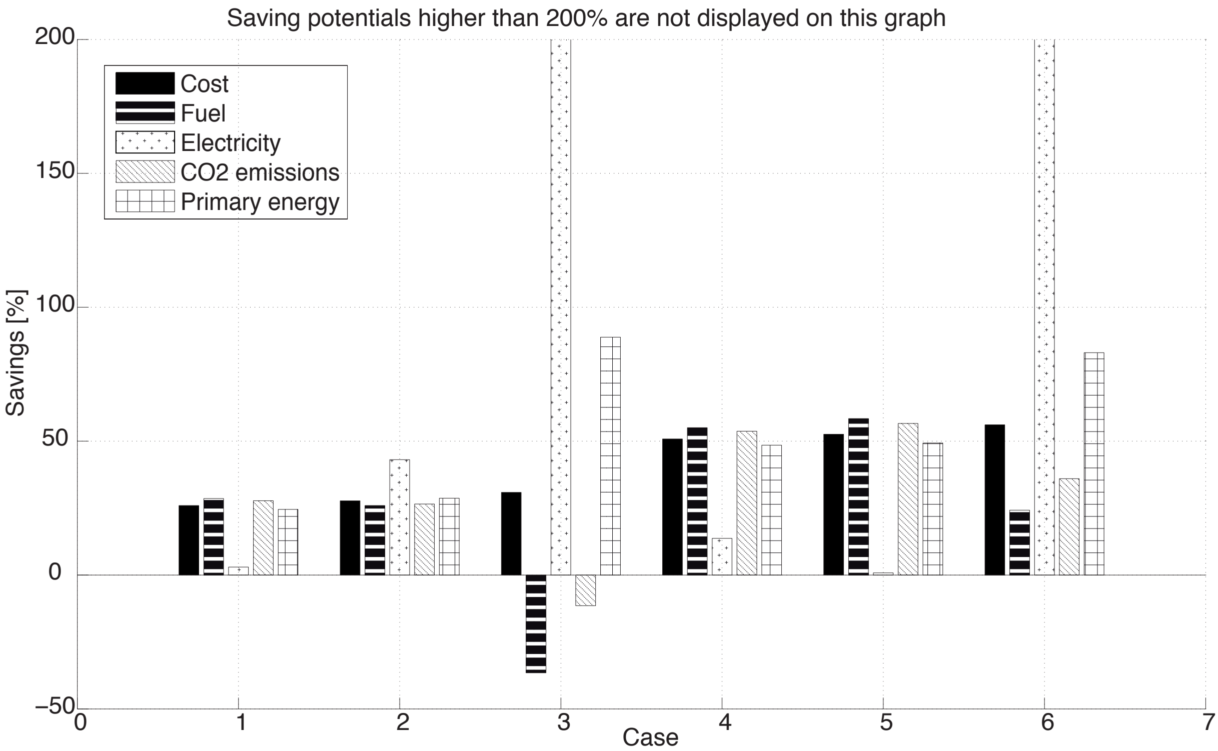


Figure 3.14: Comparison of savings for Case 1 to 6

Heat recovery between hot and cold streams is important and leads to saving potential of about 25% of operating costs, fuel consumption, CO_2 emissions and primary energy consumption. The saving potential for the electricity consumption is rather low.

Compared to Case 1, in Case 2 only the evaporation temperature of the refrigeration cycle is adapted to the process demand and a higher saving potential in electricity consumption can be observed (reduction of 41% of electricity consumption for refrigeration). To ensure the practical feasibility, the ΔT_{min} between the process and the refrigeration includes the temperature difference of the heat exchangers and when necessary of the distribution network.

Case 3 considers the integration of a co-generation engine. The fuel consumption is increased but the electricity balance shows the possibility of exporting electricity to the grid. The global operating costs are not drastically smaller than in Case 2. Regarding the payback period and the annualized profit, Case 3 is not profitable.

The electricity consumption of Case 4 is increased compared to Case 2 due to the integration of a heat pump unit. The operating costs and the fuel consumption are significantly decreased, which results in reduction of fuel consumption of 39% and CO_2 emissions of 37%. According to these results the COP can be estimated to 12.6 for a heat pump temperature difference of 20 °C. Graphically, the utility integration can be evaluated by the integrated composite curves (Maréchal and Kalitventzeff, 1996) in Figure 3.15. It can be seen that more heat than predicted with conventional pinch analysis is upgraded by the heat pump. Thanks to the combined utility integration, the condensation of the refrigeration cycle, enters into the self-sufficient pocket and heats up cold streams at low temperature. Consequently, it enables the heat pump to upgrade more heat.

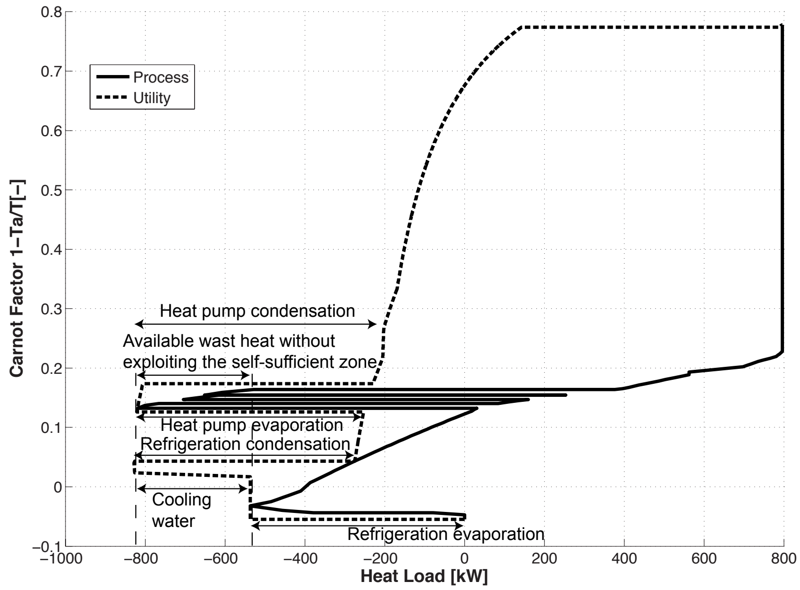


Figure 3.15: Integrated Carnot composite curves of Case 4 with refrigeration cycle, heat pump and boiler

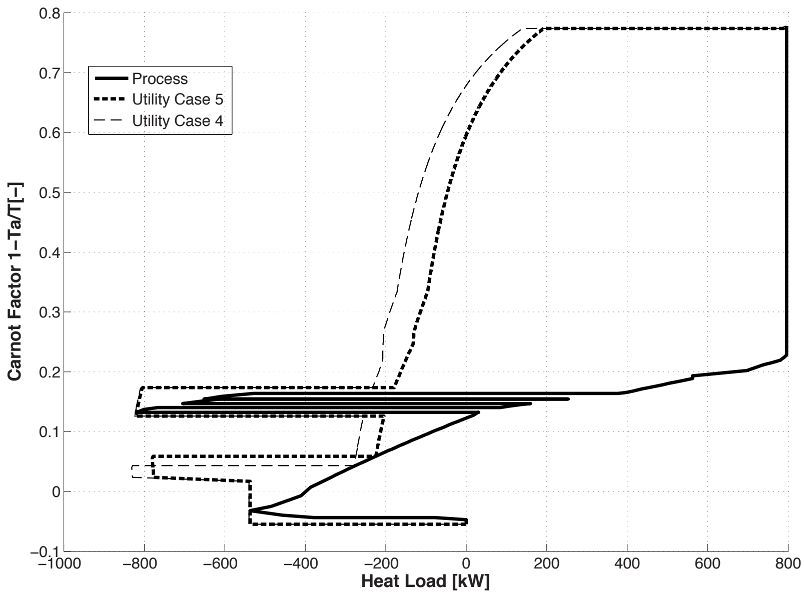


Figure 3.16: Integrated Carnot composite curves of Case 5 with adapted refrigeration cycle, heat pump and boiler

Case 5 improves the heat pump integration by adapting the temperature levels of the refrigeration. Even more heat can be upgraded by the heat pump, since the condensation of the refrigeration cycle enters at higher temperature into the self-sufficient zone and satisfies a bigger part of its hot demand (see Figure 3.16). The results can also be presented on the following two figures:

Figure 3.17 shows the integrated Carnot hot and cold composite curves and Figure 3.18 shows the integrated Carnot grand composite curve. This case offers the most promising payback times as well as the best energy savings.

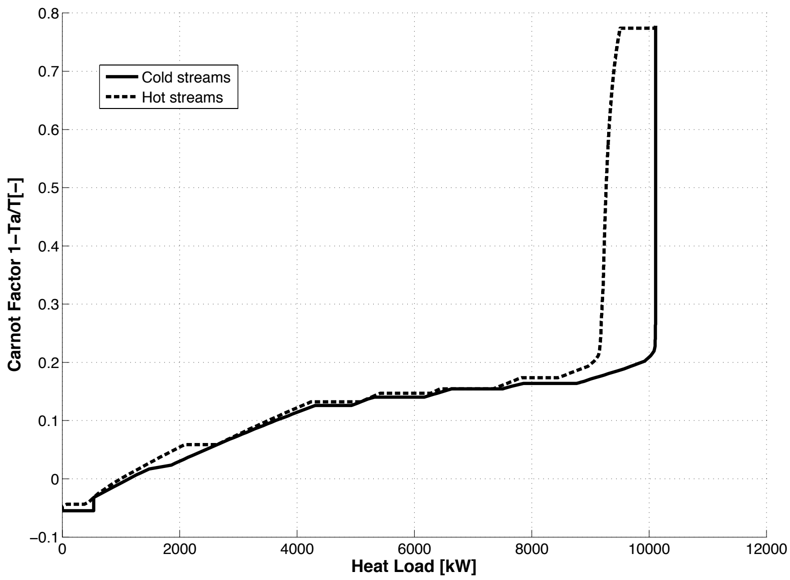


Figure 3.17: Integrated Carnot hot and cold composite curves of Case 5

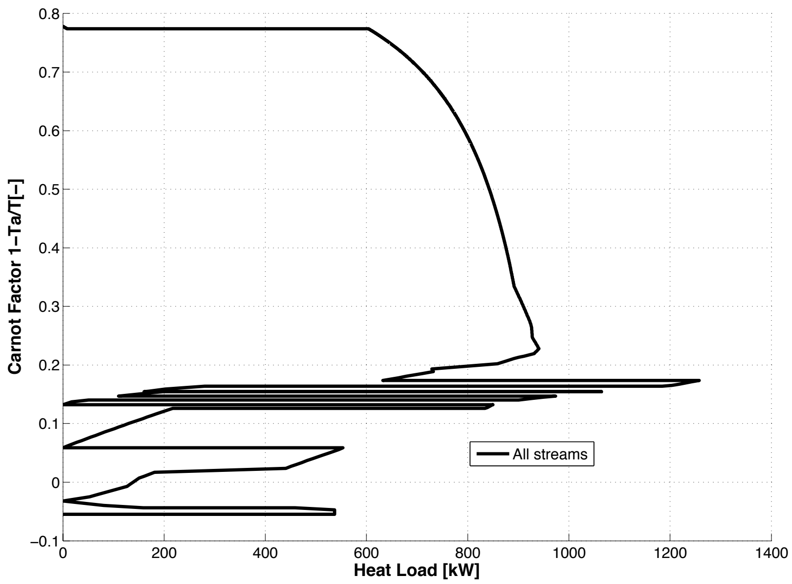


Figure 3.18: Integrated Carnot grand composite curve of Case 5

The last case (Case 6) where heat pump and co-generation units are integrated, has the lowest operating costs, but investment costs are relatively high due to the co-generation engine.

Table 3.4 compares the sizes and related investment costs of heat pumps, refrigeration cycles and co-generation engines. It has to be mentioned that the size of the different units can vary considerably for different scenarios. Especially, the size of the co-generation unit is divided by a factor 1.8 when it is integrated together with a heat pump.

Table 3.4: Detailed results of the dairy (Cases 1-6)

	Unit	Power [kW]	InvC [k€]
Case 1	ref	162	-
Case 2	ref	95	-
Case 3	ref	95	-
	cog	-3152	1222
Case 4	ref	95	-
	hp	49	124
Case 5	ref	112	-
	hp	53	134
Case 6	ref	112	-
	hp	54	136
	cog	-1750	821

3.5.1.4 Sensitivity analysis

The electricity and fuel prices play an important role to chose the optimal utilities. Therefore, this section focuses on their impacts and a sensitivity analysis is presented. Heat pumps do not have the same interest in different countries and a two dimensional sensitivity analysis on the electricity and natural gas price has been performed. For each calculated point, the utility choice among different heating equipments has been observed.

As a result, Figure 3.19 shows when heat pumps and co-generation engines have more opportunities, as a function of natural gas and electricity prices.

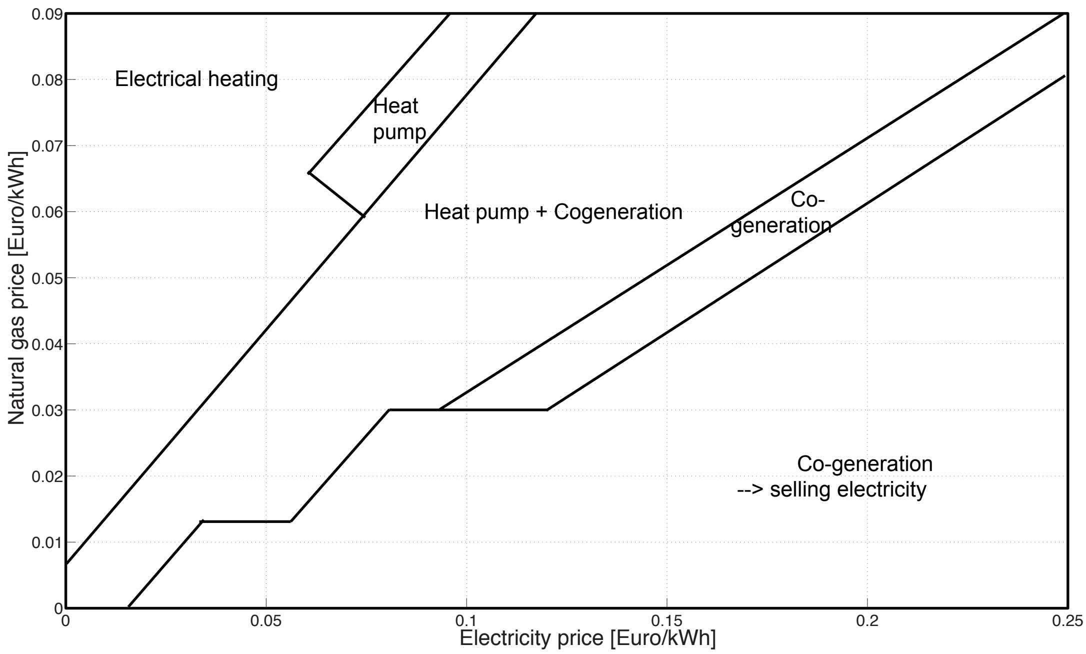


Figure 3.19: Sensitivity analysis

In most cases, the fuel to electricity ratio is in the range where the simultaneous integration of heat pump and co-generation units is interesting (e.g. in France or in Germany). However Germany, having both higher fuel and electricity prices compared to France, has a higher interest in co-generation units.

3.5.1.5 Modeling of heat pumps and mechanical vapour re-compression

There are two heat pumping types, that can be integrated:

- Closed cycle heat pumps use a refrigerant as working fluid and interact with the process only through heat exchangers, which enable the heat transfer between the heat pump and the process. Chapter 5 shows that the integration of intermediate heat transfer network between the process and the heat pumps is possible when industrial constraints require it.
- On the contrary, mechanical vapour re-compressions are open cycle heat pumps and use a given process stream directly as working fluid. The advantage is that the evaporation heat exchanger is not needed and the pressure ratio of the compressor is smaller, since only the ΔT_{min} value has to be accounted on the condensation side. Therefore the investment costs could be lower compared to a closed cycle heat pump of the same size. But on the other side often expensive process modifications are necessary and not all process streams are suitable for a mechanical re-compression. The principle of the mechanical vapor re-compression is shown in Figure 3.20.

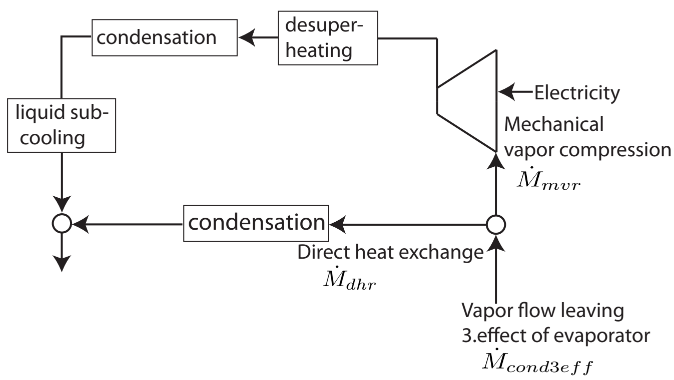


Figure 3.20: MVR integration

It could be possible to optimize the mass flow-rates of MVRs by adding a new Equation (3.34) to the MILP problem in order to create a link between the part that is re-compressed and the part that is used by direct heat exchange.

$$\dot{M}_{cond3eff} = \dot{M}_{mvr} + \dot{M}_{dhr} \tag{3.34}$$

The mechanical vapor re-compression valorizes a part of the vapor flow leaving the last effect of the evaporator ($\dot{M}_{mvr}$). The rest of the waste heat is recovered by direct heat exchange ($\dot{M}_{dhr}$). To do this, process streams are not anymore fixed and become utility streams constrained by Equation (3.34).

The clear distinction between process and utility streams will not be possible anymore. Moreover, the process streams suitable for mechanical vapour re-compression have to be first identified and then integrated as utility streams with the corresponding minimum and maximum value. Therefore, the automatic utility optimization that will be shown in Chapter 4, becomes very complex and finally, for the sake of simplicity, in this thesis mechanical vapour re-compression will be modeled as closed cycle heat pumps (water as refrigerant). The ΔT_{min} values have been adjusted to account only for the ΔT_{min} on the condensation side.

3.5.2 Cheese factory

3.5.2.1 Introduction and minimum energy requirement

The cheese factory has several pasteurization units, the evaporation unit and other process unit operations to transform milk into cheese. The detailed study is given in Appendix B.2.

First, the maximum heat recovery is computed. Exergy losses can be visualized on Figure 3.21, which shows the hot and cold Carnot composite curves. The process grand composite curve, presented in Figure 3.22, is used to define optimal temperature levels for the energy conversion system (utilities).

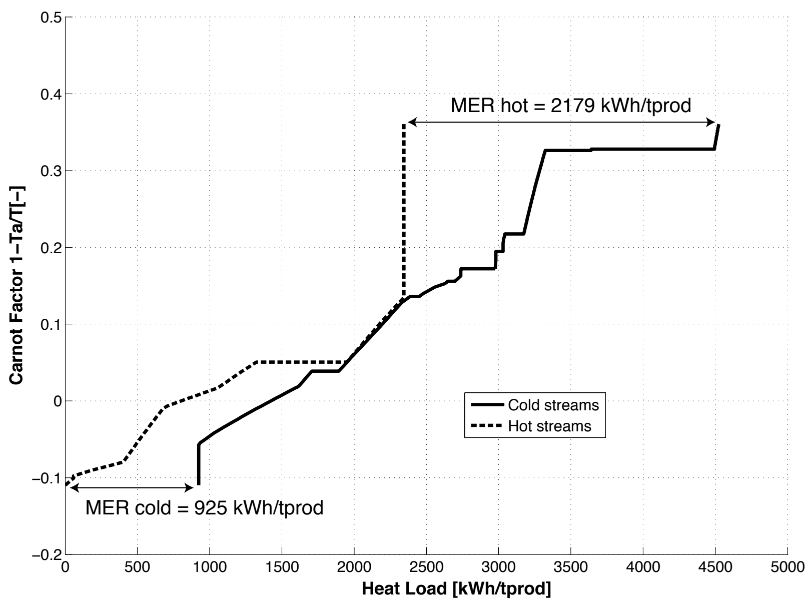


Figure 3.21: Hot and cold composite curves of the cheese factory using the Carnot scale

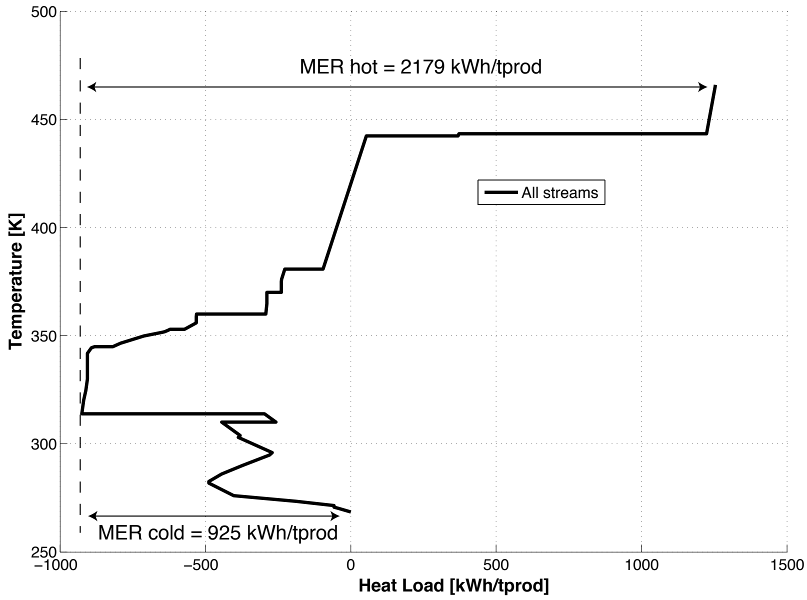


Figure 3.22: Grand composite curve of the cheese factory

3.5.2.2 Process modification with the example of an evaporation unit

To illustrate the potential of process modifications, the basic methodology has been applied to the evaporation unit of the cheese factory. In particular the evaporation unit is suitable for process modifications. The evaporation unit is one of the main energy consumers in the cheese factory and consists in 5 effects and one thermal vapour re-compression. A flow-sheet of this unit is given in Figure 3.23.

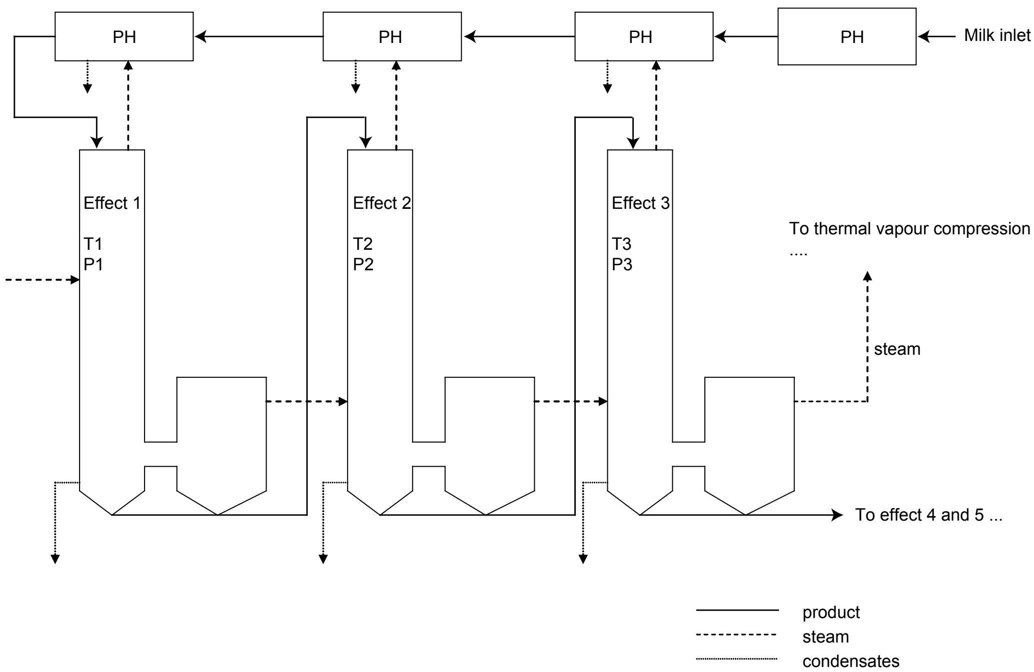


Figure 3.23: Flow-sheet of evaporation unit

Before entering the first effect of the evaporator, the whey is first preheated to reach predefined operating conditions that leads to a first evaporation. Then, the remaining liquid is sent to the second effect at a lower pressure, and the vapour boiled off in the first effect is recovered to provide heat to the second effect. The same operation is repeated in the following effects. In the present configuration, a part of the steam from the third effect is reused in a thermal vapour re-compression (TVR), driven by high pressure steam.

To calculate the MER in the previous section, the evaporation unit has been modeled as a black box in its technology representation, which includes the thermal vapour re-compression. To improve the heat recovery, operating pressures of the process can be modified and mechanical vapour re-compression (MVR) units can replace the thermal vapour re-compression unit. The cases affected by process modifications are modeled in the thermodynamic representation and summarized below. The complete case study is given in Appendix B.2. Here the focus is only on the cases that concern process modifications and heat pump integration (without co-generation integration).

Case 2a The thermal vapour re-compression is replaced by a mechanical vapour re-compression. The layout and the pressure levels of the effects are kept. The steam leaving the third effect at about 61 °C is compressed mechanically to 75 °C. As the ΔT is less than 18 °C, a dynamic compressor is suitable. Figure 3.24 shows the integrated Carnot composite curves.

Case 3a In this case, all effects are connected in parallel and a mechanical vapour re-compression is integrated. The temperature difference for the mechanical vapour re-compression is small (<10 °C). Figure 3.25 shows the integrated Carnot composite curves. The pressure levels of the five effects are adapted, so that all effects evaporates at about 70 °C.

Case 4a The pressure of all effects are modified and 3 mechanical vapour re-compressions are included: First the new temperature levels have to be defined. Assuming that the heat exchange area in the effects will not be changed, the temperature levels of effect 4 and 5 are reduced (see Equation (3.35)). The new temperature levels are 48 °C (effect 4) and 32 °C (effect 5). Theoretically it could be possible to raise waste heat at 32 °C with a heat pump to satisfy a part of the heat demand in effect 1. However, high temperature lifts makes such heat pump integration not optimal. The use of successive mechanical vapour re-compressions between the different effects has therefore been preferred (Figure 3.26). The temperature of effect 5 is quite low. If this operating conditions can not be reached in the unit, it could also be possible to raise the temperature levels of all effects.

$$\dot{Q} = U \cdot A \cdot (T_{vap} - T_{prod}) \tag{3.35}$$

Table 3.5 summarizes the results of the 3 presented cases and compared them to the current situation

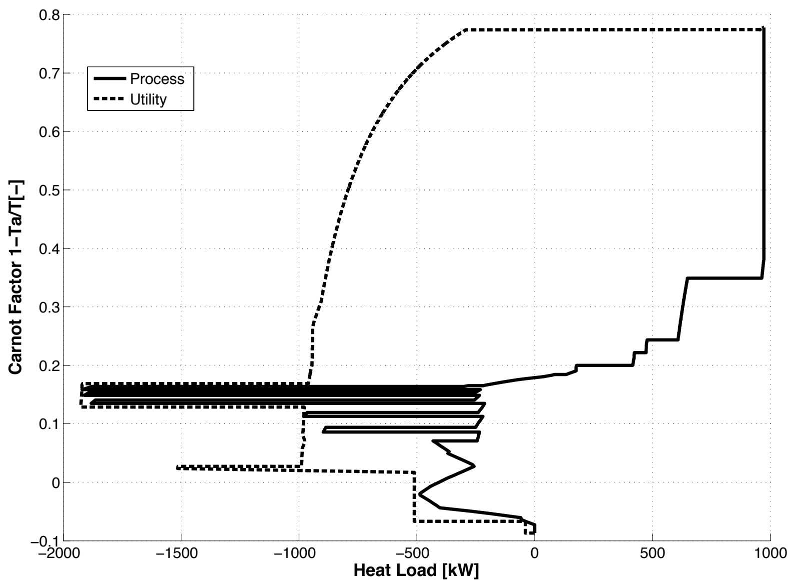


Figure 3.24: Integrated Carnot composite curves of Case 2a

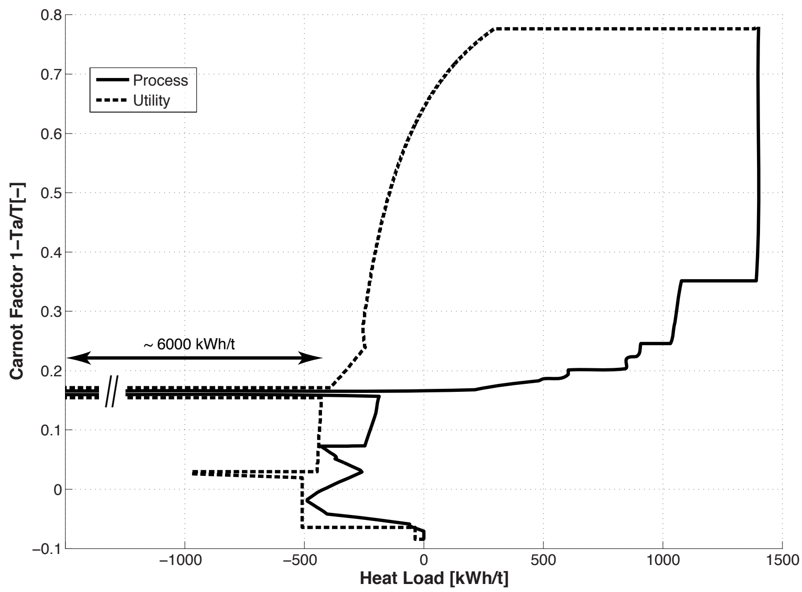


Figure 3.25: Integrated Carnot composite curves of Case 3a

(Case 0). This example shows that process modifications can improve the process efficiency and should not be forgotten when performing process integration. However such an analysis requires a good knowledge of the process and its possible modifications. It has to be done together with the industrial partner, in order to guaranty for example the product quality and safety aspects. Given its high resulting complexity, it is difficult to integrate this step in an automatic optimization strategy. The complete case study and its results are given in B.2 in Table B.11. Regarding the operating and investment costs, case 4b is the most interesting one.

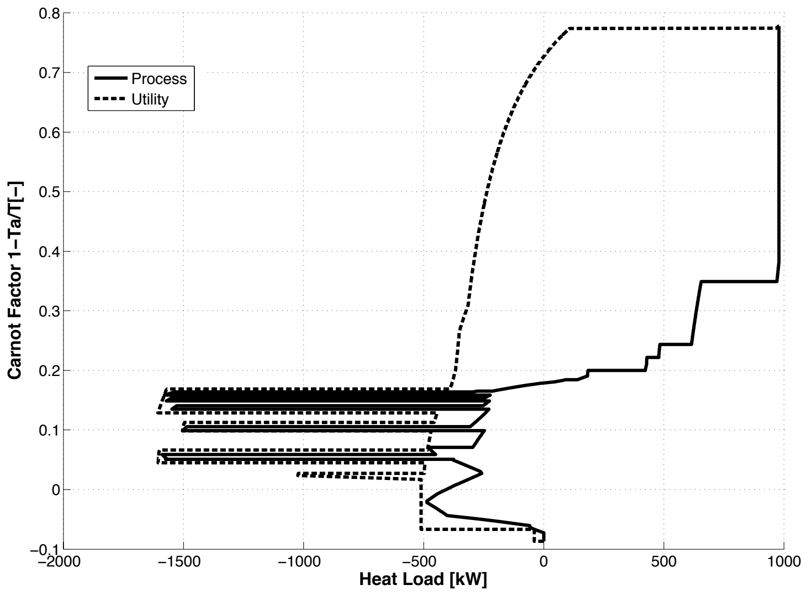


Figure 3.26: Integrated Carnot composite curves of Case 4a

Table 3.5: Results of the cheese factory (Cases: 2a, 3a and 4a)

	Unit	Case 0	Case 2a	Case 3a	Case 4a
OpC	[€/t]	126.0	88.9	86.2	69.4
E_f	[kWh/t]	2895.0	2023.7	1770.5	1394.9
E_{el}	[kWh/t]	194.0	154.8	270.2	237.0
Q_{cw}	[GWh/year]	n.a.	1003.6	464.1	512.9
M_{CO_2}	[kg/t]	602.6	423.0	382.5	303.6
E_p	[GJ/t]	15.3	10.9	11.2	9.1
$InvC$	[k€]	-	516.7	1256.9	1160.3
PB	[year]	-	3.0	5.5	2.1

3.6 Conclusions

The method has been applied to two case studies to illustrate the process integration approach. It has been shown how the operating costs and the energy consumption can be reduced through heat recovery by process integration, exergy analysis or process modifications.

Furthermore, the method is able to answer if there is a heat pump potential in a given process and can roughly evaluate the benefit and the corresponding payback time.

On the one side new insights of heat pump integration, which are summarized below have been shown. But on the other side, some limits of the method have been identified.

3.6.1 New insights of heat pump integration

As main conclusion, first it is important to note that heat pump integration is only optimal when the heat recovery between process streams is optimized. The simultaneous utility integration and the fact that self-sufficient pockets are not suppressed show more heat pump opportunities than conventional pinch analysis.

In the dairy case, it has also been shown that the simultaneous heat pump integration with other utilities and especially with the refrigeration cycle is very important to find optimal solutions. It is shown that the heat pump not only valorizes excess heat from the process but indirectly also a part of the excess heat from the refrigeration cycle. The heat pump can therefore upgrade more than 280 kW of the waste heat indicated at the beginning on the grand composite curve of Figure 3.12. Part of the heat of the last evaporation effect was originally satisfying cold process streams in the self-sufficient pocket below the pinch point. Using the condensation of heat of the refrigeration cycle, those cold process streams can be heated up and thus, more heat is available for the heat pump. The refrigeration cycle activates a so-called utility pinch point, which determines if needed, the amount of an additional cooling utility to evacuate the heat from the condensation. Exergy losses are reduced and the self-sufficient pocket is better exploited than in the case with simple heat recovery.

This example illustrates that simultaneous utility integration is very important to find best heat pump opportunities and that self-sufficient pockets should not be neglected. The rule of optimal placement for heat pump is still valid, but has to be adapted to integrate multiple utility pinch points created by heat pumps and other utility units. As conclusions, it can be stated that the optimal combination of appropriate utilities is crucial in process integration.

3.6.2 Limits and new challenges

Process integration is a useful tool, however further improvements are necessary, since the method encounters several limits.

First, it is always assumed that any heat exchange between hot and cold stream is possible, which in reality is not always feasible: as a consequence, restricted matches due to industrial constraints have to be included in the analysis to make the solutions more realistic. Chapter 5 will present a method defining process sub-systems which do not allow direct heat exchange between them. Indirect heat transfer can be realized by the integration of heat transfer units.

The second limitation is that the method works well for a single time slice or for pseudo multi-period problems (TAM). However to size new heat pump units and other utilities, it is important to solve simultaneously all periods, and to link them by integrating heat transfer networks and storage units. Chapter 6 will present a methodology for heat pump integration in multi-period processes. Heat pump working hours and therefore their profitability can be increased.

Chapter 4

Heat pump technologies and their integration

Based on the process integration approach presented in the previous chapter, this chapter focuses on the systematic integration of industrial heat pumps and presents a method that enables the identification of optimal solutions by multi-objective optimization. Particular attention will be given to the implementation of a heat pump data base, which offers the possibility to integrate different types of heat pumps to any process, in a flexible and systematic way.
Related publications: Case study: Dumbliauskaite et al. (2010), Methodology: Becker et al. (2011b).

4.1 Introduction

As already stated before, pinch analysis is a powerful tool aimed at minimizing the energy consumption of a process, both in terms of hot utility (often related to fuel combustion) and cold utility (often subject to environmental constraints). In this context, heat pump integration plays an important role in the rational use of energy conversion. A correctly placed heat pump upgrades part of the heat available below the pinch point (heat source), making it available above the pinch point (heat sink). Therefore, both heating and cooling requirements are reduced. In practice, an appropriate integration of heat pumps requires the identification of the optimal heat pump(s) and of its operating conditions.

In Chapter 3, a methodology to estimate the potential of heat pump integration based on pinch analysis has been reported. In the frame of an optimization process, it considers heat recovery between process streams together with the integration of energy conversion systems. However the integrated utilities (including heat pumps) are chosen from a predefined and limited list.

The first objective of the work presented in the next step is the development of a systematic methodology to identify the operating conditions of the heat pump through optimization. As already mentioned in the literature review of Section 3.2.1.2, the introduction of the temperature levels as decision variables makes the problem discontinuous and non-linear (Colmenares and Seider, 1987). Thus, to solve this non-linear problem, a multi-objective optimization will be proposed to be a part of the approach.

The second objective is to extend the list of proposed heat pumps by developing a structured heat pump data base in order to provide realistic heat pump models (e.g. volumetric flow-rate operating ranges for different compressor technologies or temperature ranges for refrigerants).

4.2 Method

4.2.1 Energy integration

Process integration aims at reducing energy consumption and its corresponding operating costs. The detailed methodology has been described in Chapter 3. With the help of the case study on a dairy process, the complete approach has been illustrated. The following list summarizes this approach:

1. Process analysis: This begins from the definition of the actual energy consumption of a given process, through the analysis of the process unit operations up to the final definition of the process heat requirements and the corresponding hot and cold streams.

2. Minimum energy requirement (MER): The maximum heat recovery is calculated with the ΔT_{min} assumption and the corresponding minimum hot and cold utility requirements are computed.

3. Integration of utilities: From the grand composite curves, the required temperature levels for hot and cold utilities can be defined. With a certain knowledge and experience, adequate utilities can be chosen. The optimal flow-rates are then calculated by solving a mixed linear programming problem (MILP), in order to minimize the operating costs.

The simple application of this approach leads to one optimal solution, depending strongly on the chosen utilities and optimization parameters defined by the user. For example, for a process with heat pump opportunity, the optimal integration depends on the temperature for the condensation and evaporation chosen by the user.

But it is interesting to optimize the temperature levels and to compare different heat pump solutions depending on their operating conditions. Those solutions will be different in terms of operating costs and investment costs.

4.2.2 Multi-objective optimization

In this section, a multi-objective optimization strategy will be described. Instead of one optimal resulting solution, it provides several optimal solutions. The best solution can then be determined applying financial or other criteria.

- **Input**: A range of validity for each decision variable (in this approach linked to the temperature levels of the heat pump), fixed optimization parameters (e.g. the value of the isentropic efficiency of a given compressor type)
- **Output** Pareto curves with associated results

In engineering, several conflicting objectives are often pursued simultaneously. In the case of two different objectives, it is possible to represent the results on a Pareto curve that represents one objective on each axis.

The applied multi-objective optimization strategy is based on an evolutionary algorithm. This term is coming from its analogy with the Darwin law: "survival of the fittest".

It starts with an initial population of a given number of individuals. Each individual in the system is defined by a set of decision variables. As each of these variables can vary in a certain range, it is possible to generate randomly several individuals. Once the number of initial population size has been found, the real optimization is started.

The optimization evaluates for each individual the objective functions, which allows to compare individuals between each other. Then, the individuals which gave the best performances are kept (i.e. they are the "fittest") and new individuals are created from their decision variables.

The optimization is finished when the maximum number of evaluation is reached.

4.2.3 Master and slave problem

The goal is to identify optimal heat pump technologies with their operating conditions for a given process.

The optimization strategy chosen in this work is based on the decomposition of the optimization problem in master and slave sub-problems (Figure 4.1) (Gassner and Maréchal, 2009). A similar two stage optimization approach has also be applied by Colmenares and Seider (1989b): In their outer problem (master problem), the pressure and temperature levels are optimized, whereas the inner problem (slave problem) optimizes the flow-rates. The inner problem can be linear when the objective function and the constraints are linear.

In the same way, in the presented approach the temperature levels will be optimized in the master problem (non-linear multi-objective optimization) and the utility flow-rates will be optimized in the inner problem (formulated as MILP). Additionally, the interest rate is included as a decision variable of the master problem, in order not to penalize more expensive equipments from the beginning, when estimating the annualized investment costs.

The two objectives are minimizing the operating costs of the industrial process and the investment costs related to new heat pump installations. At the master level, the values of the decision variables are chosen by the evolutionary algorithm and the operating conditions of the heat pumps (condensation and evaporation temperatures) are known, enabling the thermodynamic model to calculate the heat pump cycles and consequently to establish the enthalpy-temperature profiles of heat pumps hot and cold streams, the consumed mechanical power and the annualized investment costs for a nominal size (nominal flow-rate).

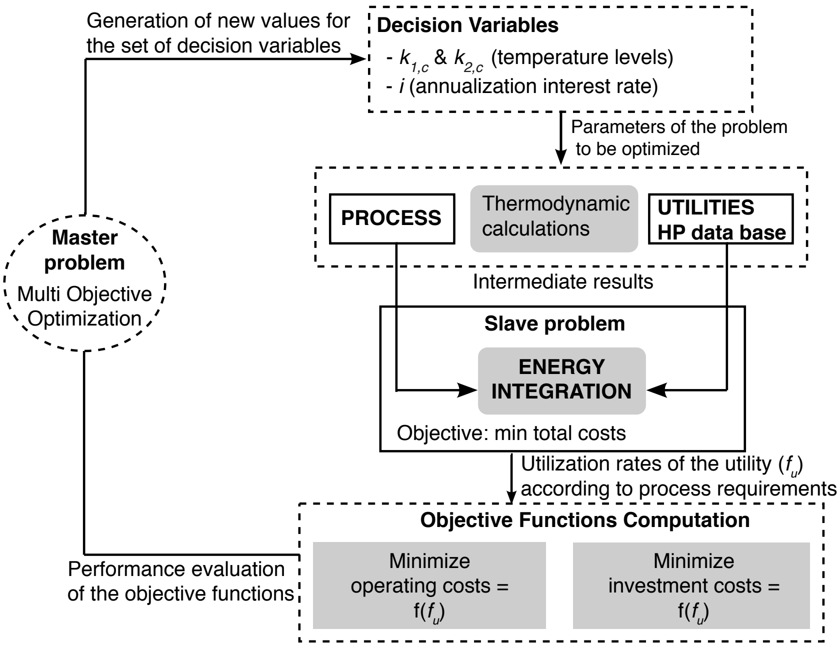


Figure 4.1: Optimization problem

Using the operating conditions of the heat pumping cycles, the slave problem solves the energy integration problem, which aims at minimizing the total cost. It includes the yearly operating costs of hot and cold utilities and the annualized investment costs of heat pump installations. All utility flow-rates are optimized simultaneously, using the mixed integer linear programming (MILP) formulation of the heat cascade (Chapter 3.4.4.1). With the selected technologies and the optimized utility flow-rates, the corresponding operating and investment costs can be computed.

The master problem is solved with an evolutionary algorithm. This allows using complex heuristics in the heat pump selection procedure and process integration, without difficulties related to constraints and non-differentiable equations.

The presented optimization approach results in a set of optimal solutions laying on a Pareto front. Considering that the goal is to help the engineers in their final decision, this method gives the advantage that the final solution can be chosen among the optimal solutions by applying financial, environmental or other criteria.

4.3 Definition of a heat pump data base

The objective of the heat pump data base is to collect different heat pump models, which can then be proposed to the process in a systematic way.

4.3.1 General overview

The heat pump data base consists in several heat pump models, working with different fluids and equipped with different compressor technologies.

As it could be interesting to integrate the same technology several times and at different operating conditions, the heat pump data base provides this possibility.

New heat pump models (e.g. new refrigerants) can easily be added. Although the focus will be given on heat pumps with non-super critical fluids, the heat pump data base can also be extended to integrate modules of heat pumps working with mixtures of fluids or super critical fluids. It is even possible to include a specific heat pump technology with known operating conditions and investment costs.

The input parameters are the evaporation and the condensation temperatures. For a nominal flowrate, the thermodynamical cycle is calculated and the computed values (superheat temperature, heat loads for condensation and evaporation and compressor power) are retrieved and used to define the input for the energy integration problem.

4.3.2 Heat pump modeling

The heat pump modules are modeled in the flow-sheeting software. For each compressor technology and refrigerant a model of a closed compression cycle heat pump is defined. Currently following refrigerants are defined: R-717, R-134a, R245fa, water. The thermodynamic methods have been adapted for each refrigerant. Moreover, the results of the flow-sheeting tool have been compared and validated with the thermodynamic properties given in the Ashrae handbook (Handbook, 2009).

An example of the heat pump modeling (here a heat pump using a centrifugal compressor) is given in Figure 4.2. The operations can be briefly summarized:

- Compression using electricity as driver (COMP_CENTR_1)
- Desuperheating from the outlet temperature of the compressor to its saturation temperature (HX_3_CENTR_1)
- Condensation (HX_2_CENTR_1)
- Subcooling (HX_5_CENTR_1); all cycles are systematically sub-cooled and it could even be possible to include this temperature difference as a decision variable in the optimization, when significant subcooling is considered.
- Expansion (VALVE_CENTR_1)
- Evaporation (HX_1_CENTR_1)
- Superheating (not shown on Figure 4.2, since this step is only considered for refrigerants that will enter in the liquid zone during compression; a typical example is the refrigerant R245fa). In practice however, all refrigerants are slightly superheated to avoid droplets in the compressor.

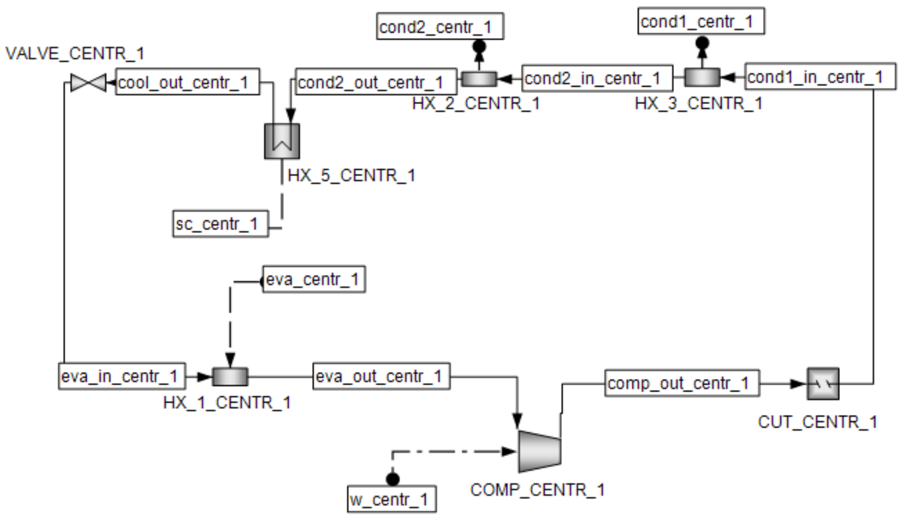


Figure 4.2: Example of heat pump modeling

For each refrigerant, 8 compressor technologies have been studied. In order to be able to have multi-stage heat pumps or parallel cycle, each cycle has been implemented 4 times. Shelton and

Grossmann (1986a) figured out that the mathematical formulation is identical for single and multi-stage refrigeration. To simplify the problem resolution, only single stage heat pumps are modeled. However, the analysis of the solutions and the final design of heat pumps can integrate multi-stage heat pumps, by combining several cycles with the same refrigerant.

4.3.3 Compressors and their operating conditions

There are two main categories of compressors: volumetric compressors, which increase the pressure by reducing the volume; and dynamic compressors, which convert the continuous transfer of the kinetic energy into a pressure rise. Figure 4.3 summarizes the different compressor types considered in this study.

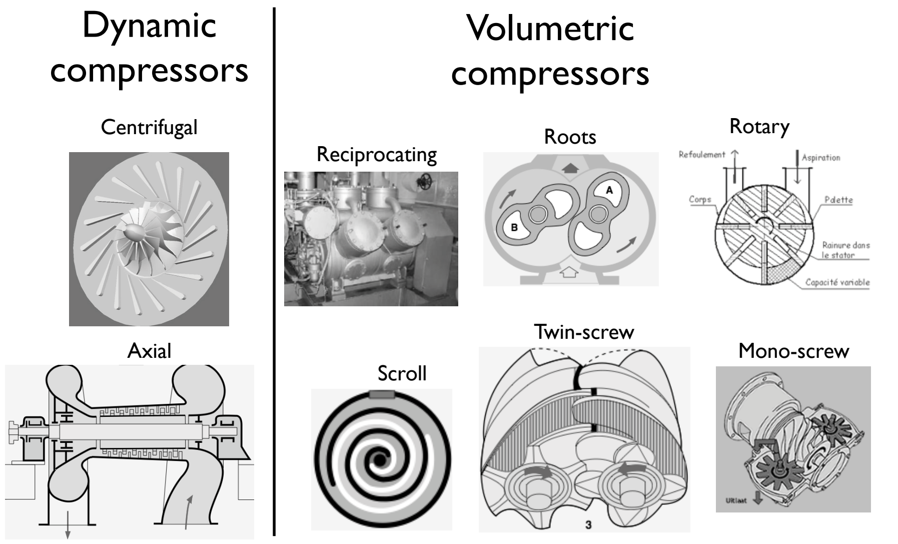


Figure 4.3: Dynamic and volumetric compressors (Favrat, 2006) (Zaid, 2008)

For each compressor, the operating conditions are reported in Table 4.1, where the volumetric flow-rates and pressures ranges and also typical efficiencies are shown (Zaid, 2008) (Favrat, 2006).

4.3.4 Temperature levels and decision variables

The temperature levels of a heat pump are optimized with the help of the adimensional parameters $k_{1,c}$ and $k_{2,c}$. They define the evaporation ($T_{eva,c}$) and condensation ($T_{cond,c}$) levels of the heat pump as a function of the temperature range of a refrigerant f (Equations (4.1) and (4.2)). These parameters correspond to the decision variables of the master problem and are chosen by the evolutionary algorithm. The calculation of the evaporation and condensation temperatures is illustrated in Figure 4.4.

$$T_{cond,c} = T_{cond,max}(f) - k_{1,c} \cdot (T_{cond,max}(f) - T_{eva,min}(f)) \tag{4.1}$$

Table 4.1: Operating conditions of compressors

Compressor	Volume rate [m^3/h]		Pressure ratio per stage [-]		Isentropic efficiency [-]
	MIN	MAX	MIN	MAX	
Centrifugal	2 000	180 000	2.5	9	0.76
Axial	9 000	200 000	1.2	2	0.76
Scroll	600	15 000	3	8	0.6
Mono-screw	200	20 000	3	5	0.75
Twin-screw	300	35 000	3	5	0.75
Rotary	100	6 000	3	4	0.6
Root	0	15 000	1.9	2.1	0.5
Reciprocating	0	10 000	3	8	0.6

$$T_{eva,c} = T_{eva,min}(f) + k_{2,c} \cdot (T_{cond,c} - T_{eva,min}(f)) \tag{4.2}$$

Table 4.2 shows the theoretical temperature ranges of some available refrigerants. However, in

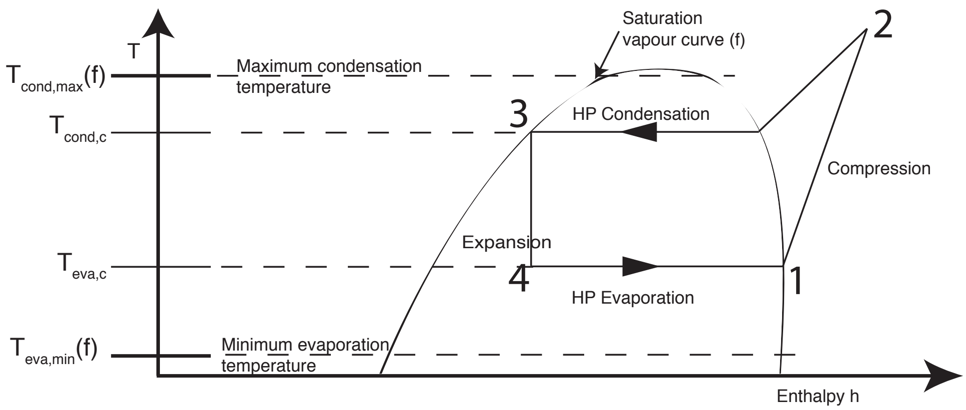


Figure 4.4: Heat pump temperature levels

practice it can be difficult to reach these limits due to technological constraints. k_1 and k_2 are fixed optimization parameters ([0,1[). The lower bound is 0 and the upper bound has to be smaller than 1 in order to guarantee two distinct temperature levels for $T_{cond,c}$ and $T_{eva,c}$. In order to increase the resolution speed, it is also possible to restrain both, the upper and the lower bound of k_1 and k_2.

4.3.5 Extension temperature levels - feasible compressor pressure ratios

This section shows how the pressure ratio of each compressor can be easily taken into account before doing the energy integration. Figure 4.5 demonstrates how the condensation and the evaporation levels are defined in the master problem. As before, first the condensation temperature is

Table 4.2: Theoretical temperature ranges of refrigerants

Fluid	$T_{cond,max}$	$T_{eva,min}$	k1	k2
R717	120 [°C]	-60 [°C]	0-0.99	0-0.99
R134a	90 [°C]	-60 [°C]	0-0.99	0-0.99
R245fa	150 [°C]	-30 [°C]	0-0.99	0-0.99
water	360 [°C]	20 [°C]	0-0.99	0-0.99

calculated according to k_1. Then, by thermodynamic calculation the corresponding pressure level is determined. With the defined minimum and maximum pressure ratio for a given compressor type the new bounds of the evaporation temperature levels ($T'_{eva,max}$ and $T'_{eva,min}$) are calculated as a function of the condensation pressure level. The evaporation temperature is finally calculated by Equation (4.3), which replaces the previous Equation (4.2).

$$T_{eva,c} = T'_{eva,min} + k_{2,c} \cdot (T'_{eva,max} - T'_{eva,min}) \tag{4.3}$$

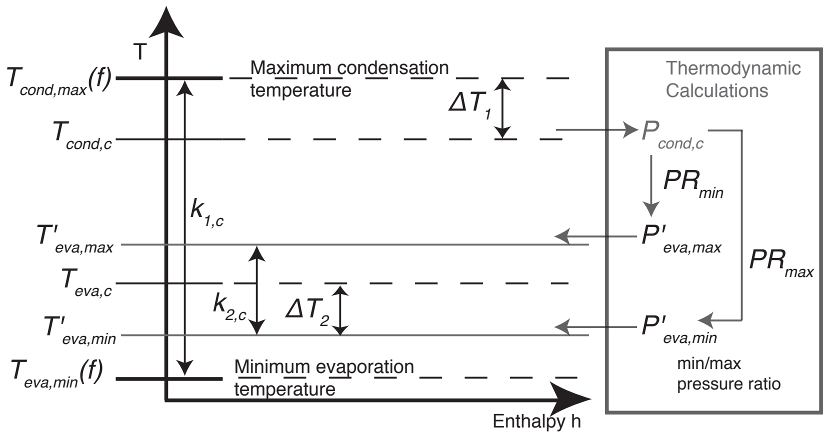


Figure 4.5: Choosing heat pump temperature levels

Two additional special cases have to considered in the approach. First, if only the new calculated evaporation temperature ($T'_{eva,min}$ on Figure 4.5) is lower than minimum evaporation temperature ($T_{eva,min}(f)$), then this value is attributed to $T_{eva,min}(f)$. If both, the new calculated maximum and minimum temperature are lower than $T_{eva,min}(f)$, the corresponding compressor is not suitable and removed from the potential solutions in this evaluation step.

4.4 Optimization

The main goal is to minimize both, process operating costs and the investment costs related to the purchase of new heat pumps. Regarding the heat pump, the parameters that should be optimized are the temperature levels and the size.

4.4.1 Decision variables

The decision variables (DV) of the optimization problem are described below:

- k_1 [0.1, 0.9] and k_2 [0.1, 0.9], which define the evaporation and condensation (levels), as a function of the operating range of the refrigerant
- interest rate i [0,0.2], which calculates the annual investment costs $InvC_a$ (Equation (4.4)). This parameter ensures that expensive technologies are not penalized from the beginning.

$$InvC_a = InvC \cdot \frac{i(1+i)^n}{(1+i)^n - 1} \tag{4.4}$$

4.4.2 Objective functions

The two conflicting objective functions of the master problem are the minimization of both the operating and the investment costs.

After having solved the slave problem, the effective requirements of the process are known and it is possible to compute the (yearly) operating costs OpC [kEuro/year].

$$OpC = d \cdot (\sum_{f=1}^{nf} (c_f^+ \sum_{u=1}^{nu} \boldsymbol{f_u} \dot{E}_{f,u}^+) + c_{el}^+ \dot{\boldsymbol{E}}_{\boldsymbol{el}}^+ - c_{el}^- \dot{\boldsymbol{E}}_{\boldsymbol{el}}^- + \sum_{u=1}^{nu} \boldsymbol{f_u} c_u) \tag{4.5}$$

In Equation (3.9) $\dot{E}_{f,u}^+$ is the nominal energy provided to unit u by the fuel (i.e. natural gas) purchased at the price c_f^+, $\dot{E}_{el}$ is the purchased ($^+$) or sold ($^-$) electrical power and c_{el} is the electricity price ($^+/^-$).

The investment costs $InvC$ are estimated according to available correlations, as a function of the power absorbed by the compressor. Two types of heat pumps can be distinguished. Standard heat pumps are easily available on the heat pump market with reasonable investment costs. A formulation from selected quotation EDF (2011) is given in Equation (4.6). f is the installation factor and $\dot{E}_{hp}$ is the electrical power (kW) needed by the compressor.

$$InvC_{hp} = f \cdot (-0.1814 \cdot \dot{E}_{hp} + 375) \cdot \dot{E}_{hp} \qquad [Euro] \tag{4.6}$$

The second type of heat pump is not directly on the market but technological feasible. The total investment costs related to this heat pumps are thus higher. The corresponding estimation (Equation

(4.7)) has already been mentioned before in Section 3.4.5.

$$InvC_{hp} = f \cdot 1500 \cdot 160^{0.1} \cdot \dot{E}_{hp}^{0.9} \qquad [Euro] \tag{4.7}$$

The domain validity of both equations is from 100 kW$_{el}$ up to 500 kW$_{el}$ electrical power (EDF, 2011). Since no investment correlations are available for smaller heat pumps, the same correlations are used, but in this case it has to be mentioned that the investment could be underestimated.

4.4.3 Solving procedure

First, the initial population of individuals is generated by the evolutionary algorithm. Once the values of the objective functions are computed, the optimizer evaluates them and generates new values for the set of decision variables, which aims at minimizing both objective functions. After a certain number of evaluations (specified by the user) the algorithm stops and the optimal solutions could be plotted on a Pareto curve to visualize the trade-off between the conflicting objective functions.

The solution quality increases with the number of initial individuals and the number of evaluations, whereas the computation time decreases.

4.4.3.1 Summary of the optimization framework

This section gives a small summary on how the heat pump data base is integrated in the optimization approach. The steps are described below for one evaluation.

1. The decision variables k_1 and k_2 evaluated in the master problem are sent to the heat pump data base.
2. The temperature levels are calculated as described in Figure 4.2 or in Figure 4.5 when the pressure ratio ranges are taken into account.
3. The thermodynamic cycle of each heat pump is solved according to the calculated temperature levels (step 2) and a nominal flow-rate.
4. The heat and power streams for the energy integration are established for a nominal volumetric flow-rate and sent to the slave problem.
5. The MILP problem optimizes the flow-rates and the real volumetric flow-rates of the heat pumps are adjusted by a multiplication factor. Volumetric flow ranges for a given compressor type are considered, by limiting the multiplication factor with a lower and an upper bound.

6. The objective functions are evaluated by the optimizer of the master problem and new values for the decision variables are generated.

4.5 Numerical examples

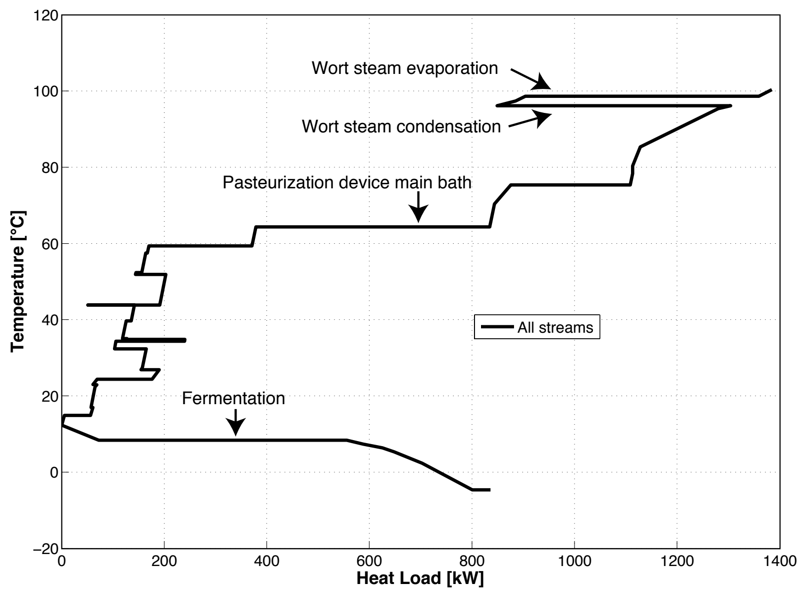
4.5.1 Brewery

4.5.1.1 Problem statement

The brewery process transforms water and malt into beer after several stages of heating and cooling. The operating conditions are fixed by process practice. Refrigeration and auxiliary systems, such as the cleaning in place, are also considered for the process integration. More details on this case study can be found in Appendix B.3.

From the definition of the process requirements, the maximum heat recovery is calculated and the resulting heat cascade is obtained. By analyzing the grand composite curve of Figure 4.6 heat pump opportunities can be identified. Therefore, this example proposes two refrigerants (R717 and R134a), which seem to be adapted to the process requirements.

Figure 4.6: Brewery Process Grand Composite Curve



As the shape of the grand composite curves is known, and to increase the optimization speed, the following temperature limits have been considered.

- R717: min -20 °C, max 40 °C

- R134a: min 0 °C, max 80 °C

The ranges for the compressors are the one previously defined in Table 4.1. The optimization parameters k_1 and k_2 can be chosen between 0.1 and 0.9.

4.5.1.2 Optimization results

In order to illustrate the progress of the evolutionary algorithm, the Pareto fronts have been plotted successively after a given number of evaluations. These graphs are obtained with an initial population of 100 and after 1643 evaluations (Figure 4.7) and 1700 evaluations (Figure4.8). The final considered Pareto front is obtained after 3000 evaluations in Figure 4.9.

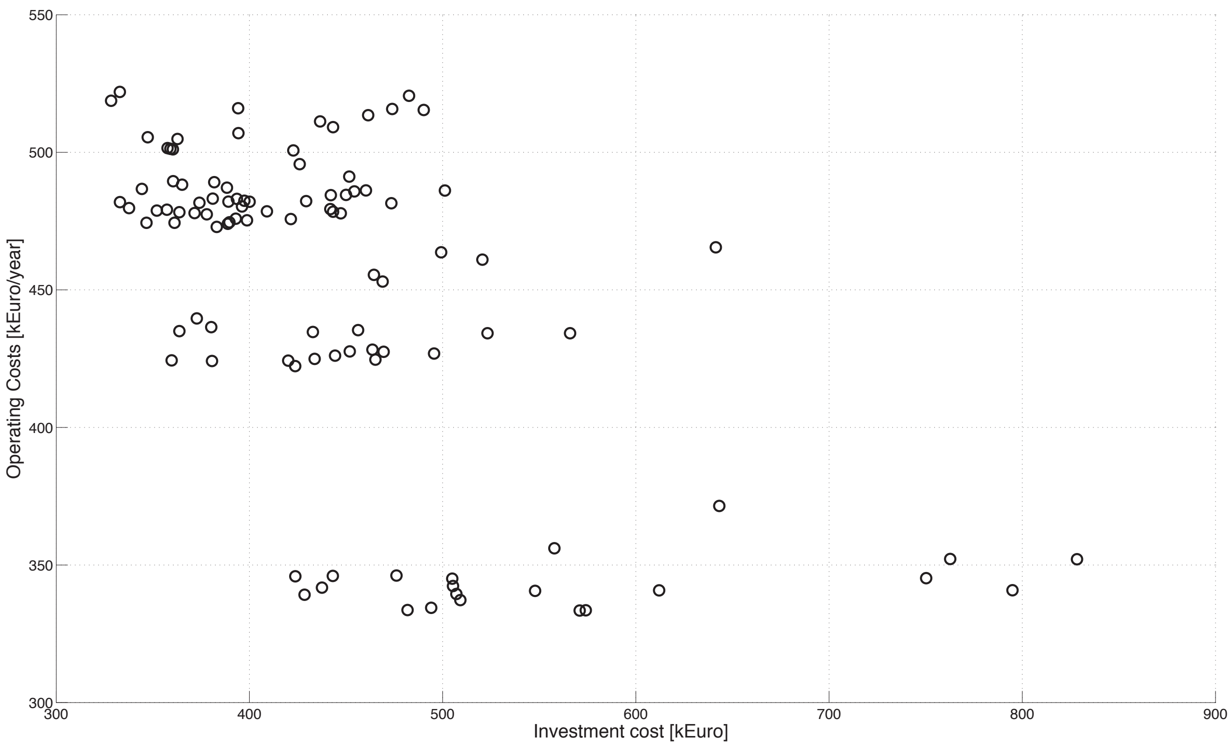


Figure 4.7: Pareto front after 1643 evaluations

The evolution of the quality of solutions can be seen through Figures 4.7 to 4.9. At the beginning lot of individuals (solution points) are "alive". By multi-objective optimization the worse individuals are removed and the good ones are mutated to find better results. Thus, it can be seen that both operating and investment costs are reduced.

To illustrate the results, 3 interesting points are chosen from the last Pareto front in Figure 4.9. Table 4.3 presents the results of these selected points and the reference situation (ref). In order to estimate the investment costs, the solutions are compared to the current situation as a reference case with one optimal integrated refrigeration cycle using the refrigerant R717 and its current operating

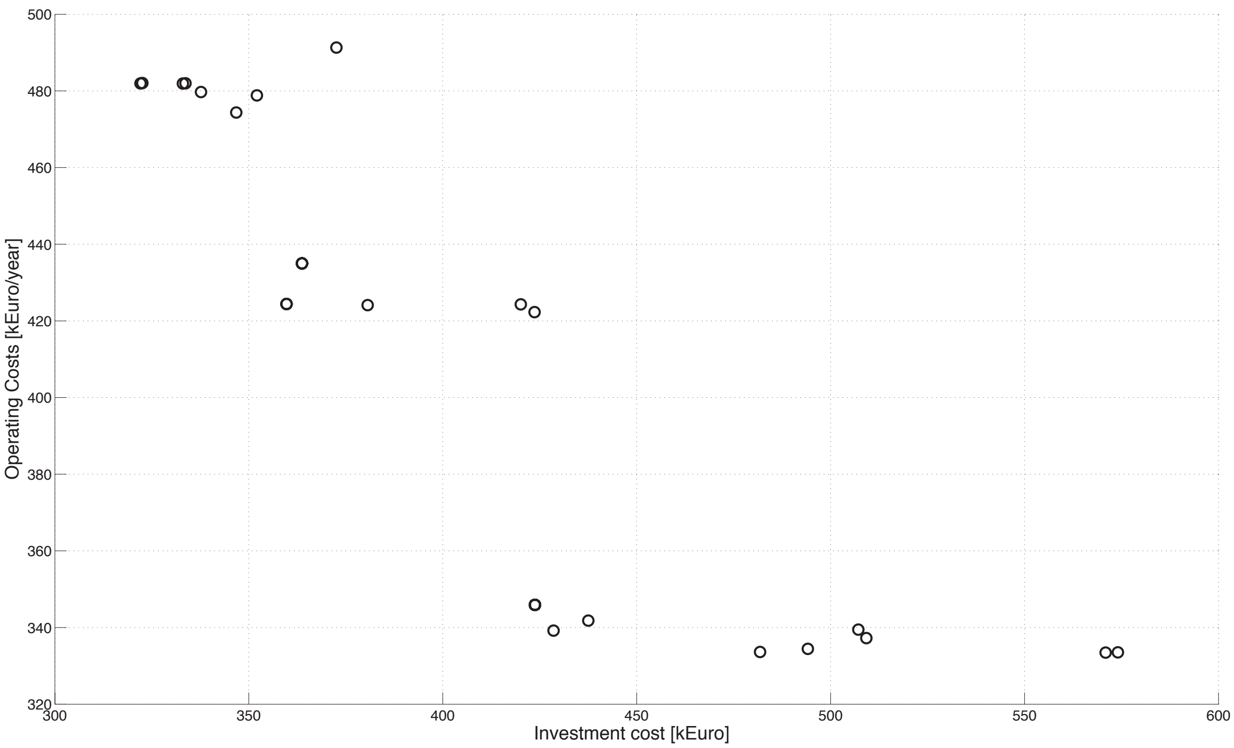


Figure 4.8: Pareto front after 1700 evaluations

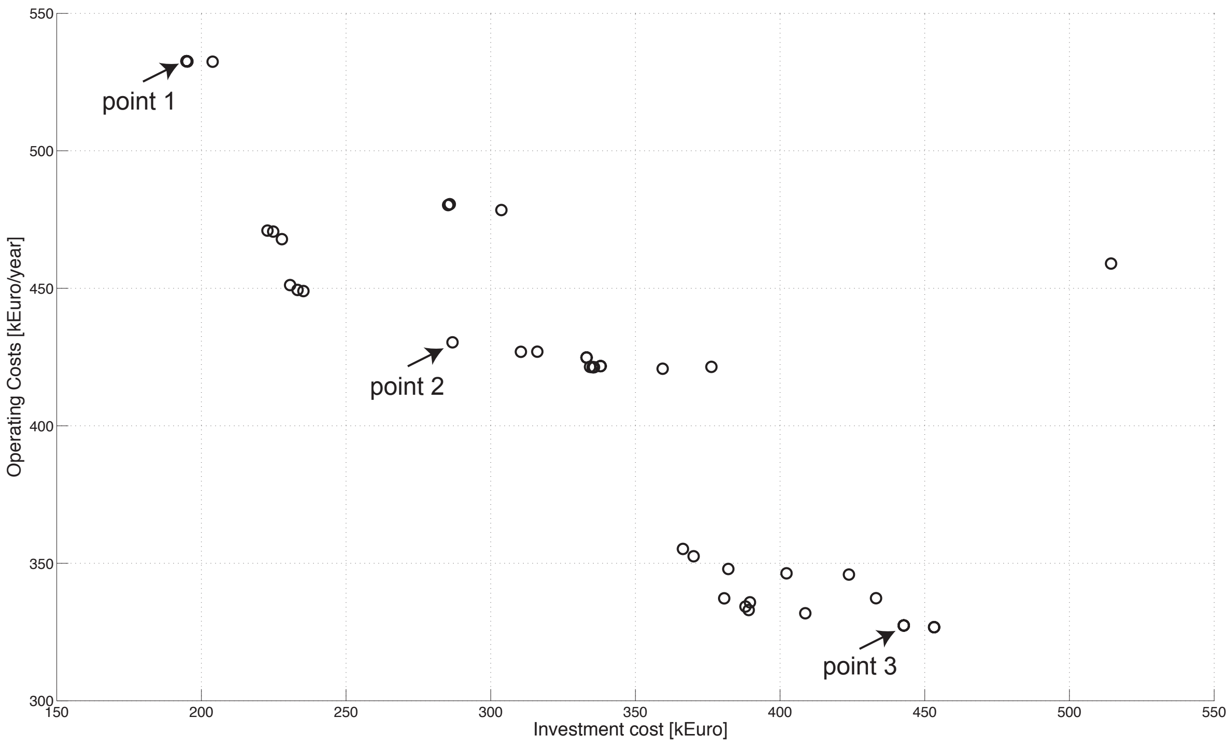


Figure 4.9: Pareto front after 3000 evaluations

conditions (condensation at -20 °C and evaporation at 40 °C).

Table 4.3: Brewery results: reference situation and 3 selected optimization points

	InvC [kEuro]	OpC [kEuro/year]	Fuel [kW]	Cooling water [kW]	Electricity [kW]	HP units [-]	PR [year]
ref	(701.6)	534.1	1172	842	336	1	-
point1	194.8	532.6	1301	874	251	5	130
point2	286.8	430.4	1023	593	221	6	2.8
point3	442.7	327.4	573	265	298	6	2.1

It is important to notice that all heat pump cycles including those for refrigeration have an investment cost in the optimization. Therefore point 1 seems to be less promising, since the investment corresponds to the complete replacement of the refrigeration. On the contrary the pay-back rates for points 2 & 3 are promising.

For comparison the integrated Carnot composite curves for the reference case is given in Figure 4.10. The theoretical investment costs of the current refrigeration cycle is given in Table 4.3.

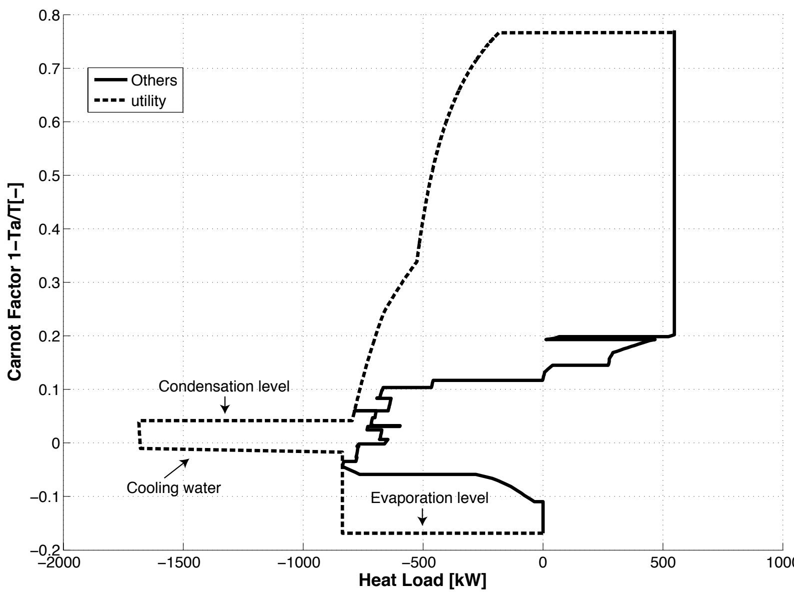


Figure 4.10: Integrated Carnot composite curves of the reference case of brewery

Point 1 corresponds to the minimum investment costs and consists in 5 heat pump cycles used for refrigeration (Figure 4.11), as required to satisfy the cooling demand of the process. The evaporation levels are optimized and adapted to the process cooling requirements. In reality, it should be analyzed if it could be possible to adapt the temperature levels of the current refrigeration system (retrofit). In order to avoid this kind of optimization problem it could also be possible to integrate a heat pump module corresponding to the current refrigeration system, with no assigned investment costs.

Table 4.4 shows the details of the selected heat pump units. The optimizer chooses 3 ammonia heat

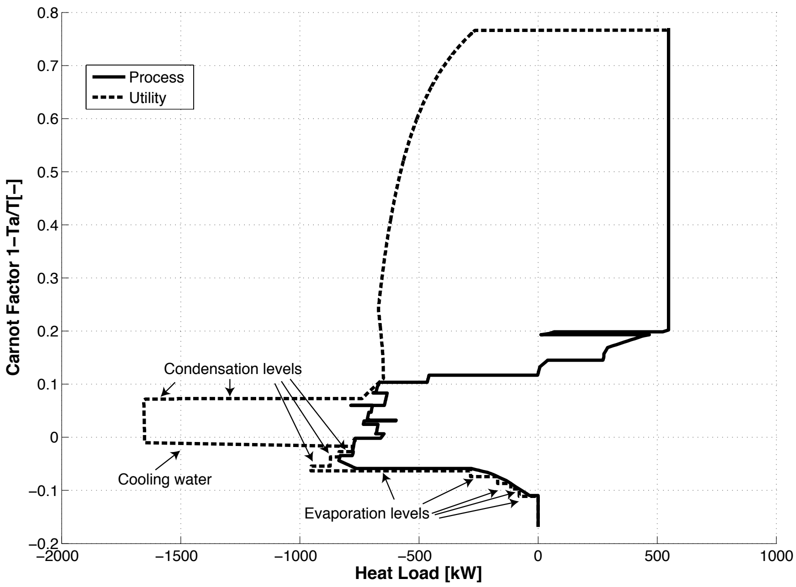


Figure 4.11: Integrated Carnot composite curves of point 1 of brewery

pumps using root compressor. This type of compressor is mainly chosen because of its low pressure ratio and moreover there is no minimal volumetric flow-rate and thus even small compressors are available. For the other two heat pumps, also mainly for refrigeration, the refrigerant R134a is proposed and mono-screw compressors with a higher isentropic efficiency can be used, since the volumetric flow-rate is high enough. It can be noticed that their condensation level is at around 50 °C, except for a small part of the process which is heated up with the desuperheating after compression, the process does not require this high temperature level. However, mono-screw compressors need a minimum pressure ratio of 3 which leads to this high condensation level.

Table 4.4: Detailed heat pump characteristics point 1 of brewery

Fluid	Compressor	Power [kW]	T_{eva} [°C]	T_{cond} [°C]	InvC [k€]	Stand
R717	ROOT	8.8	-0.9	19.2	26.5	0
R717	ROOT	5.6	-3.4	16.4	17.6	0
R717	ROOT	11.7	-6.8	11.6	34.3	0
R134a	MSCREW	190.0	5.3	50.5	97.0	1
R134a	MSCREW	35.0	2.5	50.1	19.3	1

In point2, one supplementary heat pump is integrated (Figure 4.12, Table 4.5). The latter upgrades heat, reducing consequently the operating costs with only limited supplementary investment costs. The highest condensation level achieves almost 65 °C and reduces therefore the boiler consumption. Reciprocating compressors are integrated. Compared to the root compressors, they are also available for small volumetric flow-rates, but they have a better isentropic efficiency and require a higher minimum pressure ratio.

Finally, Point3 corresponds to the minimum operating costs (Figure4.13, Table 4.6). The highest condensation level goes up to 66.5 °C.

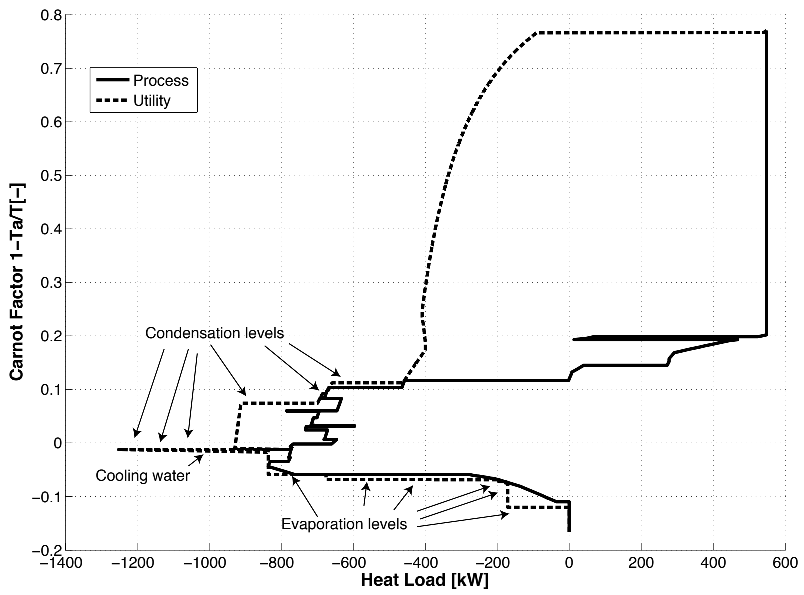


Figure 4.12: Integrated Carnot composite curves of point 2 of brewery

Table 4.5: Detailed heat pump characteristics point 2 of brewery

Fluid	Compressor	Power [kW]	T_{eva} [°C]	T_{cond} [°C]	InvC [k€]	Stand
R717	MSCREW	31.8	-8.9	23.5	84.1	0
R717	ROOT	3.0	2.9	23.4	10.1	0
R717	ROOT	45.5	4.0	23.5	116.1	0
R134a	MSCREW	54.3	3.9	51.0	29.8	1
R134a	RECIPROC	83.2	6.4	64.7	44.9	1
R134a	RECIPROC	3.1	2.3	57.3	1.7	1

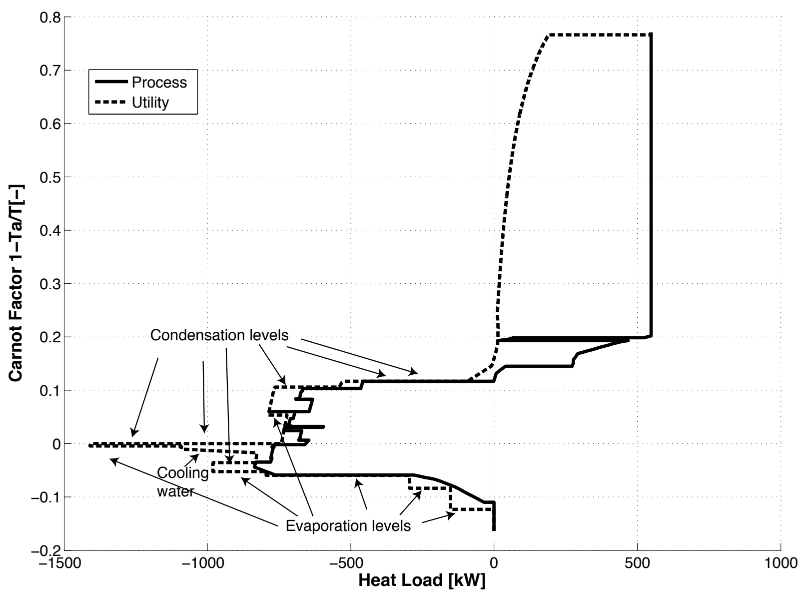


Figure 4.13: Integrated Carnot composite curves of point 3 of brewery

Table 4.6: Detailed heat pump characteristics point 3 of brewery

Fluid	Compressor	Power [kW]	T_{eva} [°C]	T_{cond} [°C]	InvC [k€]	Stand
R717	MSCREW	32.7	-9.6	27.0	86.3	0
R717	ROOT	80.8	6.3	27.0	194.6	0
R134a	MSCREW	88.7	21.7	66.5	47.8	1
R134a	MSCREW	12.8	0.0	16.8	37.1	0
R134a	TSCREW	68.7	8.2	62.3	37.4	1
R134a	ROOT	13.8	39.9	66.5	39.7	0

4.5.2 Dairy

The dairy process (Appendix B.1) has already been used to show the combined heat pump integration for heating and refrigeration. As for the brewery process, the proposed multi-objective optimization approach has been applied. Figure 4.14 shows the obtained Pareto front.

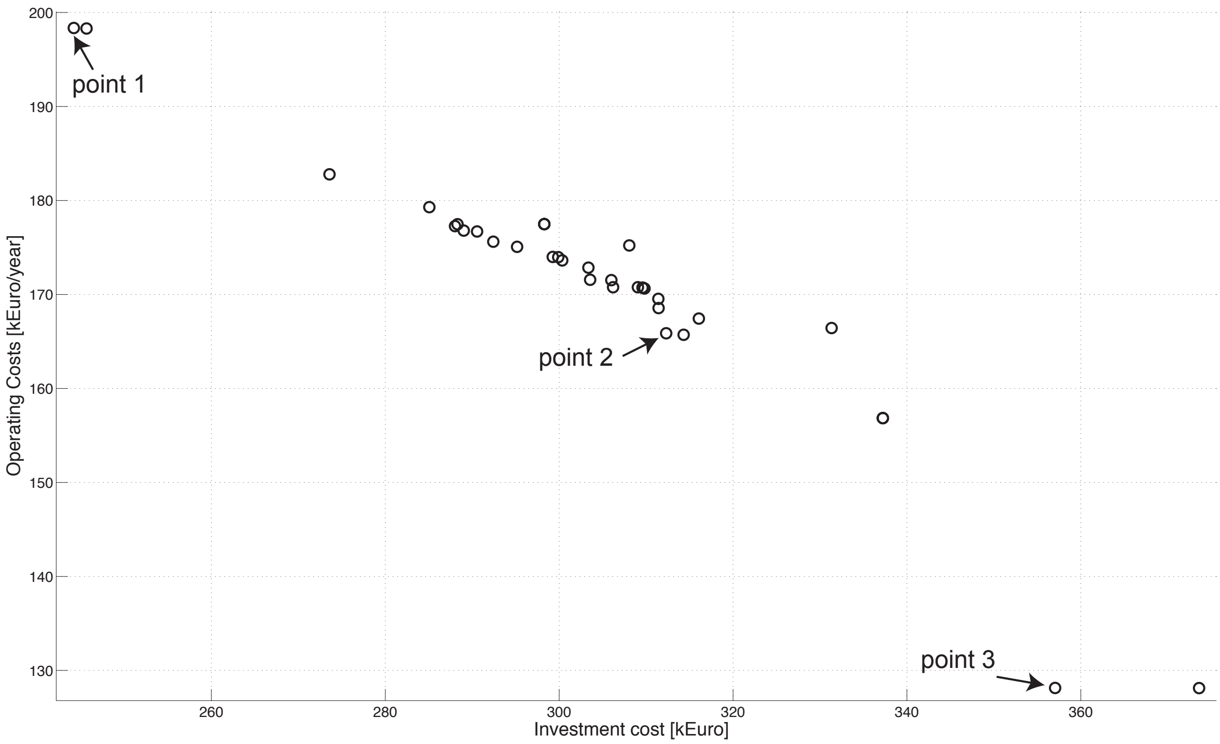


Figure 4.14: Pareto front after 1350 evaluations

Table 4.7 gives the global results and Tables 4.8 to 4.10 give the detailed heat pump integration results for the 3 selected points.

Figures 4.15 to 4.17 show the corresponding integrated composite curves.

In point 1, it is interesting that the optimizer choses evaporation temperature levels below the cooling water temperature. Therefore, the condensation heat is directly used to heat up process

Table 4.7: Dairy results

	InvC [kEuro]	OpC [kEuro/year]	Fuel [kW]	Cooling water [kW]	Electricity [kW]	HP units [-]	PR [year]
ref		199.3	1662	862	162		-
point1	244.2	198.3	1760	882	94	4	244.2
point2	312.3	165.9	1404	589	122	5	9.4
point3	357.0	128.1	1000	241	147	3	5.0

Table 4.8: Detailed heat pump characteristics point 1 of dairy

Fluid	Compressor	Power [kW]	T_{eva} [°C]	T_{cond} [°C]	InvC [k€]	Stand
R717	MSCREW	64.5	-9.8	22.8	159.0	0
R717	ROOT	3.8	3.4	22.8	12.5	0
R717	ROOT	23.1	0.1	19.4	63.2	0
R717	ROOT	2.8	1.2	21.3	9.5	0

Table 4.9: Detailed heat pump characteristics point 2 of dairy

Fluid	Compressor	Power [kW]	T_{eva} [°C]	T_{cond} [°C]	InvC [k€]	Stand
R717	MSCREW	79.3	-7.8	24.9	191.5	0
R717	ROOT	16.8	-4.2	14.3	47.4	0
water	ROOT	0.1	54.3	69.4	0.7	0
water	ROOT	7.9	56.3	71.7	24.1	0
water	ROOT	17.3	55.0	69.3	48.6	0

Table 4.10: Detailed heat pump characteristics point 3 of dairy

Fluid	Compressor	Power [kW]	T_{eva} [°C]	T_{cond} [°C]	InvC [k€]	Stand
R717	MSCREW	104.5	-1.1	33.8	245.4	0
R717	ROOT	1.1	-9.6	8.3	4.2	0
water	AXIAL	41.8	58.2	72.5	107.5	0

streams at quite low temperature. The cooling water enters above in the self-sufficient pocket to cool down process streams at higher temperature.

Point 2 corresponds quite good to the attended results from the conventional pinch analysis. A refrigeration system at two levels and two heat pumps which upgrade the waste heat without entering the self-sufficient pocket are integrated.

As already seen in the previous chapter, it is possible to enter the self-sufficient zone by adapting the temperature levels of the refrigeration and the heat pump cycle. This corresponds exactly to the integration of point 3. It is also interesting to see that the size of the axial compressor is slightly higher than needed. This is due to the minimum volumetric flow-rate constraint of this type of compressor.

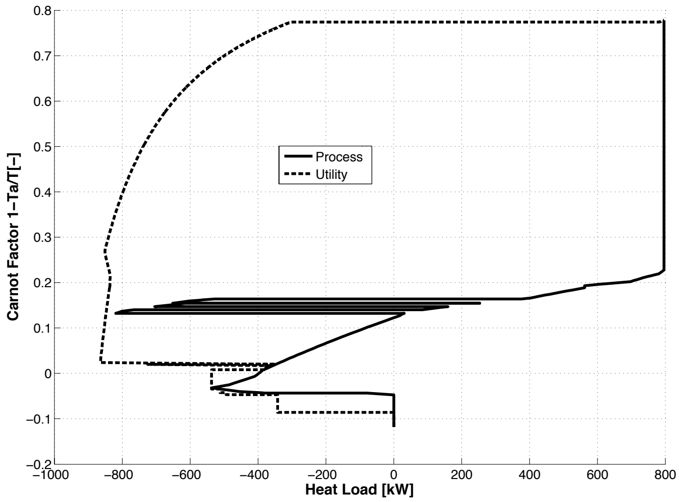


Figure 4.15: Integrated Carnot composite curves of point 1 of dairy

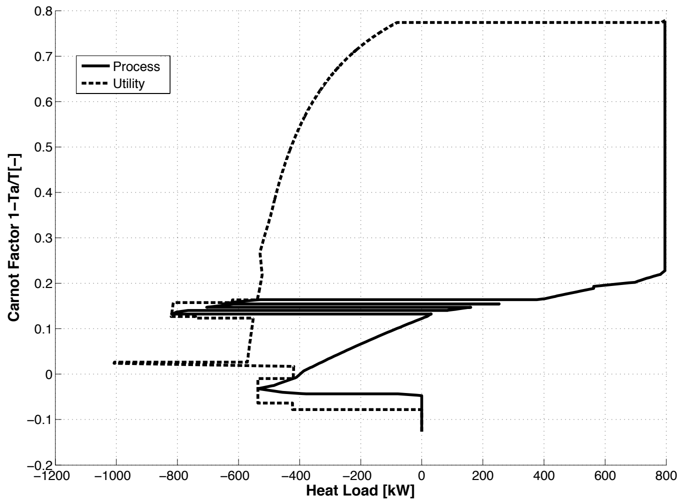


Figure 4.16: Integrated Carnot composite curves of point 2 of dairy

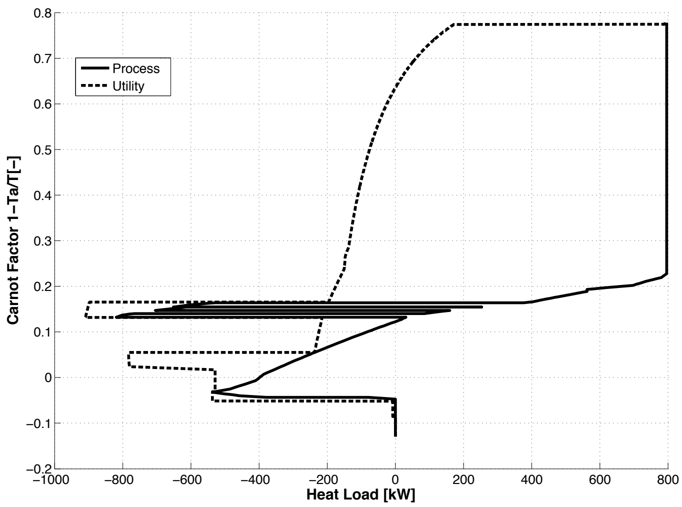


Figure 4.17: Integrated Carnot composite curves of point 3 of dairy

4.6 Conclusions and perspectives

Chapter 3 showed how appropriate temperature level for heat pump integration can be chosen manually by analyzing the shape of the grand composite curve. The approach presented above can do this automatically using a heat pump data base and a systematic optimization strategy. It brings benefits both to the non-experts, who can solve the problem without deep knowledge of the method, and to the experts, who can identify new innovative combinations of solutions.

The results consist in several optimal solutions among which the final solution can be chosen by applying financial analysis for example.

The approach used in the implementation of the heat pump data base allows the optimal integration of single stage heat pumps and even the combination of different cycles (e.g. refrigeration and heat pump as shown with the example of a dairy in Section 3.5.1). The final design of heat pumps can also integrate multi-stage heat pumps. The data base approach allows to use specific models for each type of heat pump. It is implemented in such a way that any compressor technology or heat pump cycle configurations working with pure or mixture of refrigerants could be integrated.

On the other side, following disadvantages including proposition for amelioration of the method can be stated:

- The calculation time is quite high and depends on the computer efficiency. (For example the brewery example took about 1week of calculation time to reach 3000 evaluations). However the used Matlab code has not been programmed in a very efficient way, since it is only a prototype to demonstrate the feasibility of the methods. A reprogramming by IT specialist will most probably result in considerable calculation time savings.
- Convergence problems during the thermodynamic calculations of the heat pump cycle are frequent and due to bad initializations in the flow-sheeting software. Especially for a wide range of operating conditions, the thermodynamic model shows its limitation. One possible solution could be to create a data base of already calculated and converged solutions and to initialize automatically before calculation to improve the problem resolution.
- The results are strongly influenced on the input data and especially on the investment cost definitions. Therefore, a good estimation of those costs and of their corresponding validity domain is crucial to get realistic results in terms of investment costs and to be able to estimate the payback time.

Chapter 5

Heat exchange restrictions

This chapter extends the basic methodology to include heat exchange restrictions due to industrial constraints. The method accounts for restricted matches between sub-systems, but in order to increase the heat recovery, intermediate heat transfer units can be considered for the indirect heat transfer. To identify them, the new concept "envelope composite curves" is developed.
Related publication: Methodology: Becker and Maréchal (2012a).

5.1 Introduction

5.1.1 State of the art

It has been shown that pinch analysis is a promising tool to optimize the energy efficiency of industrial processes. To realize the maximum heat recovery and the optimal integration of utilities to supply process heating and cooling requirements, first the heat load distribution based on process and optimal utility streams has to be calculated. One major difficulty is the assumption that any hot stream can exchange heat with any cold stream of the system. In reality, heat exchanges become difficult or even impossible, due to constraints such as the distance between streams or product quality and/or safety reasons, or due to system dynamics such as non-simultaneous operations.

Heat exchange restrictions can easily be added to the transportation problem approach (Cerda et al., 1983). More in detail, Cerda and Westerberg (1983) studied heat exchanger networks with restricted matches and propose an algorithm which imposes constraints disallowing in part or in total the matching of stream pairs. Their approach concerns only the process streams and no intermediate heat transfer units are considered to reduce the energy penalty related to the heat exchange restrictions.

Forbidden matches between certain pairs of process streams have also been considered by Papoulias and Grossmann (1983b). They proposed a mathematical formulation to identify the heat load distribution that minimizes the energy penalty of restricted matches without proposing any solutions for adding heat transfer fluids or integrating utility systems.

The total site approach, presented by Dhole and Linnhoff (1992) and later by Klemeš et al. (1997), implicitly accounts for restricted matches between different processes (sub-systems) of the production site before designing the heat exchanger network. For each sub-system the grand composite curve is built. The hot and cold composite curves of the global system are then graphically constructed without considering self-sufficient pockets, since the heat recovery of the hot and cold streams inside the pockets is possible. Thus, heat exchange restrictions exist between the sub-systems, which can only exchange heat via the steam system.

Also Hui and Ahmad (1994) studied total site integration with indirect heat transfer between process plants through steam utilization from the steam network. In this work exergy analysis is used and the self-sufficient zones are not always suppressed, in order to increase the efficiency of the process. More directly, Rodera and Bagajewicz (1999) pointed out that skipping the self sufficient pocket can reduce significantly the opportunities for heat recovery and they use a transshipment model which calculates the heat to be transferred between two process plants. An extension to several plants has been presented later by the same authors (Bagajewicz and Rodera, 2000) (Bagajewicz and Rodera, 2002). Bagajewicz and Rodera (2001) proposed a single heat belt as heat transfer system, which exchanges heat between process plants by an intermediate fluid. Only for special cases (3 process plants) this problem can be solved with a MILP formulation.

Combining the total site proposed by Dhole and Linnhoff (1992) and the approach of Bagajewicz and Rodera (2000), Bandyopadhyay et al. (2010) introduced the site level grand composite curves as a graphical resolution approach, which includes the self-sufficient pockets for indirect heat transfer.

Indirect heat transfer between plants and an extension to industrial zones containing several process plants have been presented by Stijepovic and Linke (2011). Mainly the global utility system is optimized and only waste heat can be transferred between process plants.

Maréchal and Kalitventzeff (1999) proposed a MILP strategy, which integrates forbidden heat exchange connections as constraints in the targeting phase, contrary to the approaches presented by Cerda and Westerberg (1983) or Papoulias and Grossmann (1983b). It also allows the integration of heat transfer fluids. The penalty in terms of utility and operating costs are considered.

5.1.2 Synthesis

This thesis proposes an extension of the MILP strategy (Maréchal and Kalitventzeff, 1999), which can restrict the heat exchange for a given connection between one hot and one cold stream. The proposed methodology will restrict the heat exchange between sub-systems. Direct heat exchange between sub-systems is therefore not allowed, but heat can be transferred indirectly through the heat transfer system.

Depending on a given process or system, the members of sub-systems should be defined to represent the following situations: direct heat exchange can become impossible due to safety or product quality reasons, long distances or non-simultaneous process operations. Heat exchange restrictions can be necessary between two process units, but also between utility and process units (e.g. a boiler which transfers the heat indirectly through a steam network to a process unit).

The main objective is to extend the basic heat cascade formulation of Chapter 3 to include heat exchange restrictions between sub-systems.

A second goal is to develop a mathematical formulation, which enables to choose optimal heat transfer units. Based on the sub-system definition, a new MILP formulation will be proposed to calculate the envelope composite curves consisting in fictive hot and cold streams, which embed optimal heat transfer technology.

5.2 Methodology

The new methodology, proposed here, takes into account heat exchange restrictions at the targeting stage by dividing industrial plants into sub-systems. By definition, heat recovery and heat exchanges between hot and cold streams inside a sub-system are possible but no direct heat exchange with other sub-systems is allowed (Figure 5.1). The only way to satisfy heat demands of sub-systems is to exchange heat with streams of a unit belonging to the heat transfer system. Two types of units can be distinguished: Heat transfer units (HTU) and common units (CU). HTUs are defined when heat has to be transferred between two sub-systems (e.g. hot water heat recovery loop).

Utility units (e.g. steam boiler or cooling water) can be defined as CUs in the heat transfer system when they can exchange heat without restrictions with all sub-systems. If not, it is also possible to define them in a related sub-system. In this case, suitable HTUs (e.g. a steam network for transferring heat from a boiler to the process) have to be defined to ensure the indirect heat transfer.

The flow-rates of the heat transfer fluids are optimized in order to minimize the energy penalty of restricted matches between sub-systems. Particular attention will be given to the choice of the optimal heat transfer technology by using a new mathematical formulation to draw the envelope composite curves for indirect heat exchange. The problem is solved in several steps which are summarized in Table 5.1.

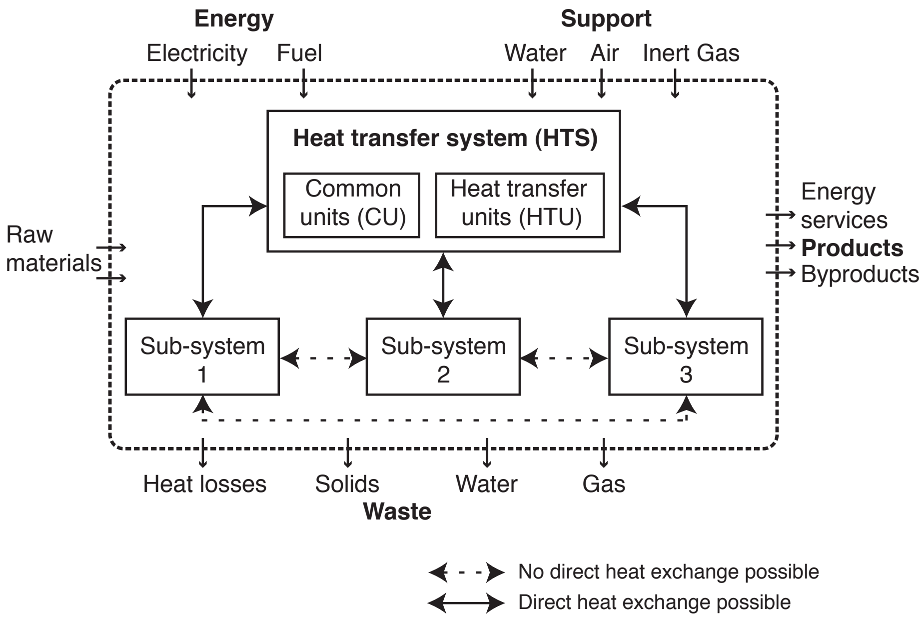


Figure 5.1: Definition of subsystems

Table 5.1: Problem algorithm

Step	Name	Goal	MILP problem
1	No restrictions	Find best case and corresponding multiplication factors and operating costs	Section 3.4.4.1
2	Restrictions	Visualize energy penalty including heat exchange restrictions	Section 5.3.1
3	Envelope	Visualize envelope composite curves for defining optimal HTUs (input: multiplication factors or operating costs from step 1)	Section 5.3.2
4	Integrated HTUs	Choose and integrate HTUs with the help of the previous step	Section 5.3.1
4a	Optimization (optional)	Multi-objective optimization for choosing among several possibilities	Section 5.3.3
5	Heat load distribution	Compute heat load distribution of final solution	Section 3.4.4.3

In step 1, a MILP problem without restricted matches is solved. This defines the optimal flowrates in the energy conversion systems (utility system) and the minimum operating costs (Section 3.4.4.1).

The energy penalty is then calculated in step 2, by solving a MILP problem including restricted matches between sub-systems but with no possibility to integrate new heat transfer technologies for the indirect heat exchange (Section 5.3.1).

In step 3, the envelope composite curves are computed by using a MILP problem including industrial constraints and fictive hot and cold streams for the heat transfer system (Section 5.3.2). It represents the necessary enthalpy-temperature profiles for optimal heat transfer systems to eliminate the energy penalty.

The identified HTUs are then added in the list of hot and cold streams and a final MILP problem including restricted matches and chosen optimal heat transfer units can be resolved (step 4). Their flow-rates are calculated by solving again the MILP problem of Section 5.3.1.

For large scale problems, it is possible to perform an optional multi-objective optimization (step 4a) in order to choose between different HTUs (Section 5.3.3).

As a last step (step 5), the heat load distribution problem (HLD), proposed by Maréchal and Kalitventzeff (1989), is then adapted to incorporate the definition of sub-systems and restricted matches (Section 3.4.4.3). The resolution of the HLD problem becomes much easier and is the basis for the heat exchanger network design. The major advantages of the presented method are:

- The process is divided into sub-systems (more realistic than just heat restriction constraints between two streams); heat exchange inside sub-systems is favored.
- On the contrary to the total site approach, self-sufficient pockets are not suppressed. This allows the maximization of the combined heat and power production.
- Simultaneous optimization of the utility units and the heat transfer system defines the complete list of streams including utility streams for the heat load distribution.
- Optimal heat transfer technologies can be identified and optimized.
- The combinatorial nature of the HEN design is reduced. Thus, the design of the heat exchanger network becomes easier and more flexible and implicitly includes topological constraints.

5.3 Heat cascade formulations

The general MILP formulation has been presented in the previous chapter. In the next sections, this formulation will be adapted to include restricted matches and to calculate the envelope composite curves.

5.3.1 MILP formulation with restricted matches between sub-systems

Like for the conventional heat cascade calculation, the objective is to minimize the operating costs (Equation (3.9)). When the industrial plant is divided into sub-systems, the basic heat cascade (Equations (3.14) and (3.15)) is replaced by Equations (5.1) to (5.7) in order to take into account heat exchange restrictions. It is important to remark that units and their streams can either be part of a sub-system or belong to the heat transfer system (common units or heat transfer units).

For each sub-system s, the heat cascade is given by Equations (5.1) to (5.3). When a sub-system has a deficit or a surplus of heat in the temperature interval k, the heat is supplied from the heat transfer system ($\dot{\boldsymbol{Q}}^{-}_{\boldsymbol{hts,s,k}}$) or respectively removed by the heat transfer system ($\dot{\boldsymbol{Q}}^{+}_{\boldsymbol{hts,s,k}}$). $\dot{\boldsymbol{R}}_{\boldsymbol{s,k}}$ is the cascaded heat to the lower temperature interval k in sub-system s.

$$\sum_{h_{s,k}=1}^{ns_{h,s,k}} \boldsymbol{f_u} \dot{Q}_{h,s,k,u} - \sum_{c_{s,k}=1}^{ns_{c,s,k}} \boldsymbol{f_u} \dot{Q}_{c,s,k,u} + \dot{\boldsymbol{Q}}^{-}_{\boldsymbol{hts,s,k}} - \dot{\boldsymbol{Q}}^{+}_{\boldsymbol{hts,s,k}} + \dot{\boldsymbol{R}}_{\boldsymbol{s,k+1}} - \dot{\boldsymbol{R}}_{\boldsymbol{s,k}} = 0 \quad \forall k = 1..., nk \quad \forall s = 1..., nsub \tag{5.1}$$

$$\dot{\boldsymbol{R}}_{\boldsymbol{s,1}} = 0 \quad \dot{\boldsymbol{R}}_{\boldsymbol{s,nk+1}} = 0 \qquad \dot{\boldsymbol{R}}_{\boldsymbol{s,k}} \geq 0 \qquad \forall k = 2..., nk \quad \forall s = 1..., nsub \tag{5.2}$$

$$\dot{\boldsymbol{Q}}^{+}_{\boldsymbol{hts,s,k}} \geq 0 \quad \dot{\boldsymbol{Q}}^{-}_{\boldsymbol{hts,s,k}} \geq 0 \qquad \forall k = 1..., nk \quad \forall s = 1..., nsub \tag{5.3}$$

The heat cascade for the heat transfer system hts, which contains all process or utility units not belonging to a sub-system, is given by Equations (5.4) and (5.5).

$$\sum_{h_{hts,k}=1}^{ns_{h,hts,k}} \boldsymbol{f_u} \dot{Q}_{h,hts,k,u} - \sum_{c_{hts,k}=1}^{ns_{c,hts,k}} \boldsymbol{f_u} \dot{Q}_{c,hts,k,u} - \sum_{s=1}^{nsub_k} \dot{\boldsymbol{Q}}^{-}_{\boldsymbol{hts,s,k}} + \sum_{s=1}^{nsub_k} \dot{\boldsymbol{Q}}^{+}_{\boldsymbol{hts,s,k}} + \dot{\boldsymbol{R}}_{\boldsymbol{hts,k+1}} - \dot{\boldsymbol{R}}_{\boldsymbol{hts,k}} = 0 \quad \forall k = 1..., nk \tag{5.4}$$

$$\dot{\boldsymbol{R}}_{\boldsymbol{hts,1}} = 0 \quad \dot{\boldsymbol{R}}_{\boldsymbol{hts,nk+1}} = 0 \qquad \dot{\boldsymbol{R}}_{\boldsymbol{hts,k}} \geq 0 \qquad \forall k = 2, ..., nk \tag{5.5}$$

A graphical representation of the heat cascade calculation for sub-systems and the heat transfer system is given in Figure 5.2. To ensure that the heat is cascaded correctly, a second set of equations is necessary. Equations (5.6) and (5.7) express respectively the heat balance of the hot and the cold streams in the heat transfer system. The flow-rates of the heat transfer units have to be optimized in order to satisfy the remaining heat demand of all sub-systems.

$$\sum_{h_{hts,k}=1}^{ns_{h,hts,k}} \boldsymbol{f_u} \dot{Q}_{h,hts,k,u} + \dot{\boldsymbol{R}}_{\boldsymbol{hts,k+1}} - \dot{\boldsymbol{R}}_{\boldsymbol{hts,k}} - \sum_{s=1}^{nsub_k} \dot{\boldsymbol{Q}}^{-}_{\boldsymbol{hts,s,k}} \geq 0 \qquad \forall k = 1..., nk \tag{5.6}$$

$$-\sum_{c_{hts,k}=1}^{ns_{c,hts,k}} \boldsymbol{f_u} \dot{Q}_{c,hts,k,u} + \dot{\boldsymbol{R}}_{\boldsymbol{hts,k+1}} - \dot{\boldsymbol{R}}_{\boldsymbol{hts,k}} + \sum_{s=1}^{nsub_k} \dot{\boldsymbol{Q}}^{+}_{\boldsymbol{hts,s,k}} \leq 0 \qquad \forall k = 1..., n_k \tag{5.7}$$

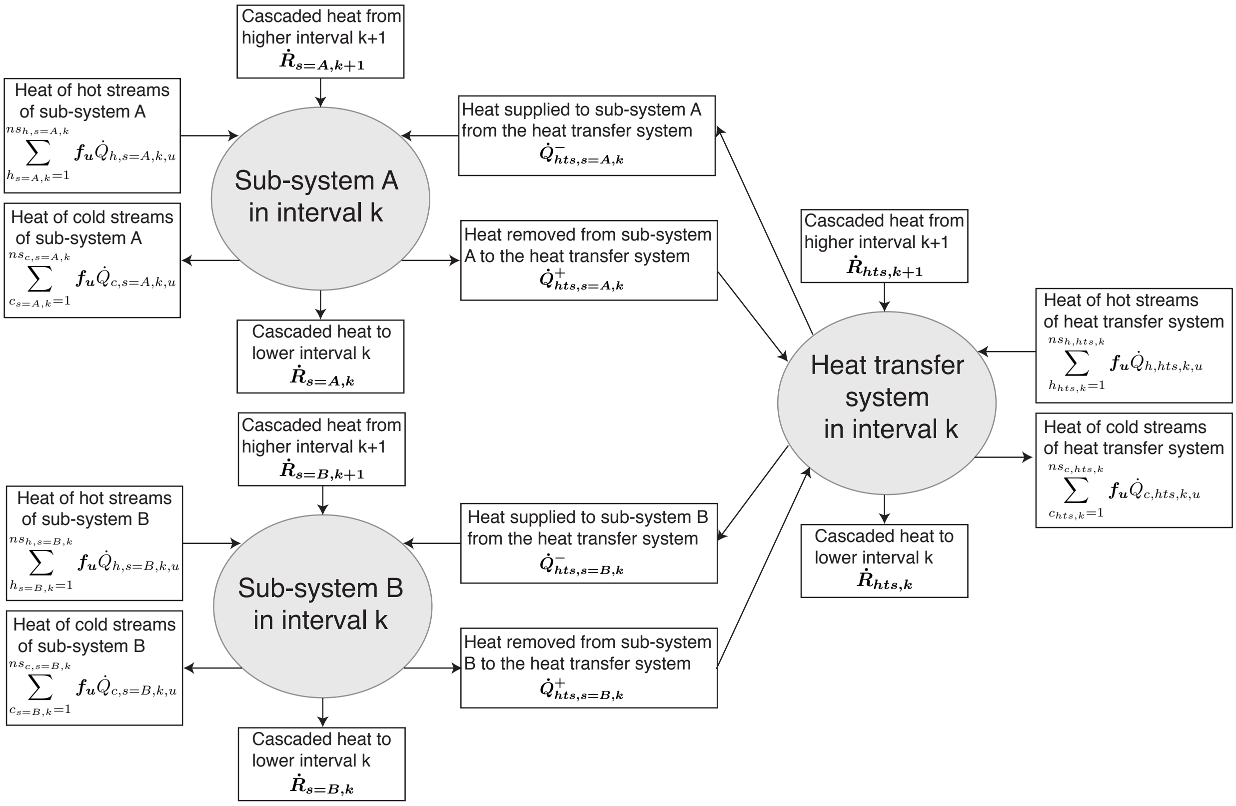


Figure 5.2: Graphical heat cascade representation for MILP formulation with restricted matches of sub-systems

The total cascaded heat from an upper interval k is expressed by Equation (5.8).

$$\dot{\boldsymbol{R}}_{\boldsymbol{k}} = \dot{\boldsymbol{R}}_{\boldsymbol{hts,k}} + \sum_{s=1}^{nsub_k} \dot{\boldsymbol{R}}_{\boldsymbol{s,k}} \qquad \forall k = 1..., nk+1 \tag{5.8}$$

The complete mathematical formulation consists in Equations (3.9) to (3.13) and the modified heat cascade Equations (5.1) to (5.8).

When no heat transfer unit is considered, the MILP problem presented here allows to calculate the cost of the energy penalty. For this, at least one common hot and cold utility have to be defined in the heat transfer system in order to satisfy the heat cascade equations. It is also possible to integrate indirect heat transfer units, based on a list of units defined with their enthalpy temperature profiles and the corresponding technologies. Pumping costs, proportional to their optimized flow-rates have to be included, in order to size them correctly when solving the MILP formulation presented in this section.

However, this formulation does not provide information on optimal temperature levels of heat transfer units. For example, when several sub-systems are defined for a given process, it is difficult to define the necessary temperature levels for hot water loops. In the next section, a MILP formulation is presented, which draws the envelope composite curves in order to choose the optimal temperature levels of the intermediate heat transfer units.

5.3.2 Envelope composite curves - choice of intermediate heat transfer networks

The goal of the next formulation is to calculate the envelope composite curves, which will embed the intermediate heat transfer units. The envelope composite curves represent the conditions that the streams of the heat transfer units have to satisfy, when the impact of the heat exchange restriction should be eliminated. For each temperature interval k, one fictive hot stream $\boldsymbol{\dot{Q}_{h,env,k}}$ and one fictive cold stream $\boldsymbol{\dot{Q}_{c,env,k}}$ are added to the heat cascade of the heat transfer system and Equations (5.3) - (5.5) are replaced by Equations (5.9) - (5.11).

$$\sum_{h_{hts,k}=1}^{ns_{h,hts,k}} \boldsymbol{f_u} \dot{Q}_{h,hts,k,u} - \sum_{c_{hts,k}=1}^{ns_{c,hts,k}} \boldsymbol{f_u} \dot{Q}_{c,hts,k,u} + \boldsymbol{\dot{Q}_{h,env,k}} - \boldsymbol{\dot{Q}_{c,env,k}} - \sum_{s=1}^{nsub_k} \boldsymbol{\dot{Q}^{-}_{hts,s,k}} + \sum_{s=1}^{nsub_k} \boldsymbol{\dot{Q}^{+}_{hts,s,k}} + \boldsymbol{\dot{R}_{hts,k+1}} - \boldsymbol{\dot{R}_{hts,k}} = 0 \qquad \forall k = 1, ..., n_k \tag{5.9}$$

A graphical representation is given in Figure 5.3. The fictive hot and cold streams form the envelope composite curves. To cascade heat correctly following constraints have to be added to the

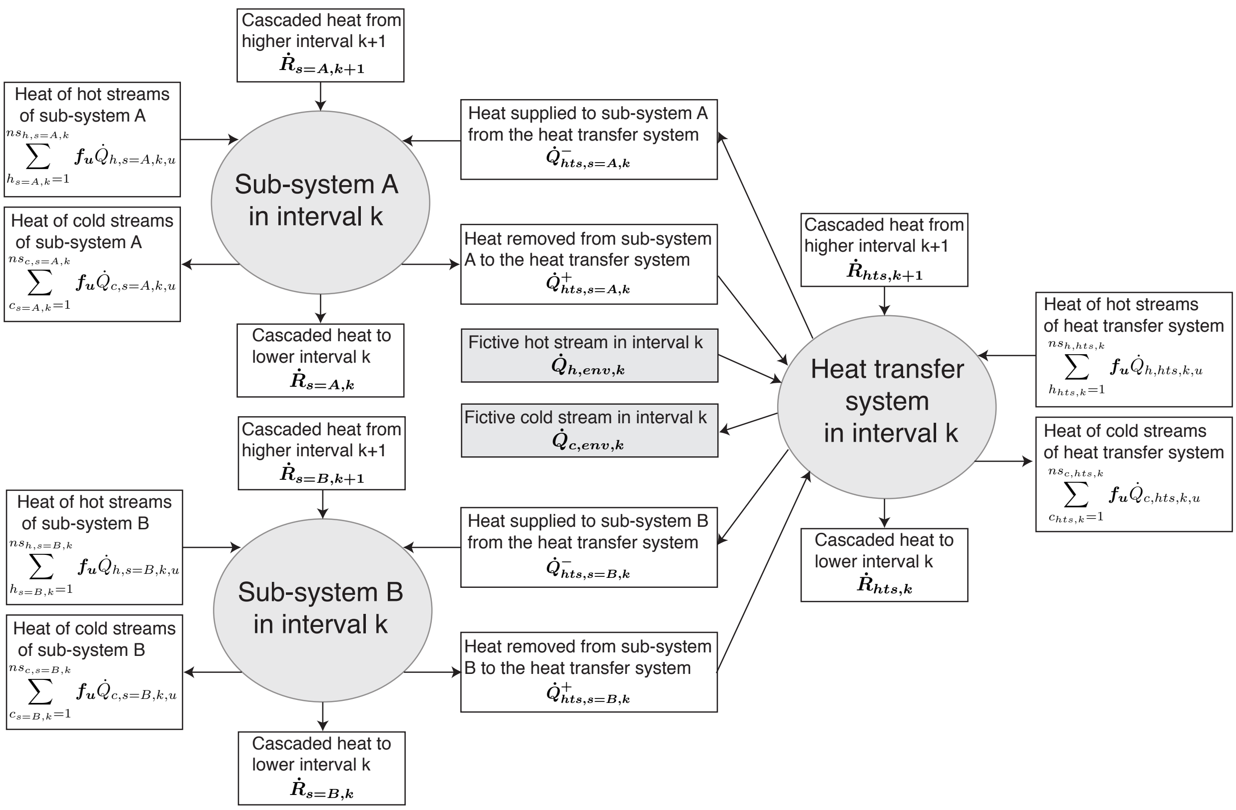


Figure 5.3: Graphical heat cascade representation for MILP formulation for envelope composite curves

mathematical formulation.

$$\dot{\boldsymbol{R}}_{\boldsymbol{hts,1}} = 0, \dot{\boldsymbol{R}}_{\boldsymbol{hts,nk+1}} = 0 \qquad \dot{\boldsymbol{R}}_{\boldsymbol{hts,k}} \geq 0 \qquad \forall k = 2, ..., nk \tag{5.10}$$

$$\begin{aligned} \dot{\boldsymbol{Q}}^{+}_{\boldsymbol{hts,s,k}} \geq 0 \quad \dot{\mathbf{Q}}^{-}_{\mathbf{hts,s,k}} \geq 0 \quad \dot{\boldsymbol{Q}}_{\boldsymbol{h,env,k}} \geq 0 \quad \dot{\boldsymbol{Q}}_{\boldsymbol{c,env,k}} \geq 0 \\ \forall k = 1..., nk \quad \forall s = 1..., nsub \end{aligned} \tag{5.11}$$

The cold envelope composite curve takes the excess heat from the sub-systems, whereas the hot envelope composite curve gives back the same amount of heat to the sub-systems requiring heat. The cold envelope curve is therefore a set of fictive cold streams and can be interpreted as the maximum enthalpy temperature profile for the cold streams of the heat transfer units. In the same way, the hot envelope curve defines the minimum enthalpy temperature profile for the hot streams of the heat transfer units. An example of envelope composite curves is given on Figure 5.4.

A supplementary constraint (Equation (5.12)) is necessary, to ensure that the heat absorbed by the cold streams is equal to the heat delivered by the hot streams. When calculating the envelope, it is assumed that there are no heat losses in the intermediate heat transfer networks.

$$\sum_{k=1}^{nk} \dot{\boldsymbol{Q}}_{\boldsymbol{h,env,k}} = \sum_{k=1}^{nk} \dot{\boldsymbol{Q}}_{\boldsymbol{c,env,k}} \tag{5.12}$$

In order to ensure that the cold envelope composite curve is hotter than the hot envelope composite curve, Equation (5.13) is added to the formulation.

$$\sum_{r=k}^{nk} \dot{\boldsymbol{Q}}_{\boldsymbol{h,env,k}} - \sum_{r=k}^{nk} \dot{\boldsymbol{Q}}_{\boldsymbol{c,env,k}} \leq 0 \qquad \forall k = 2, ..., nk \tag{5.13}$$

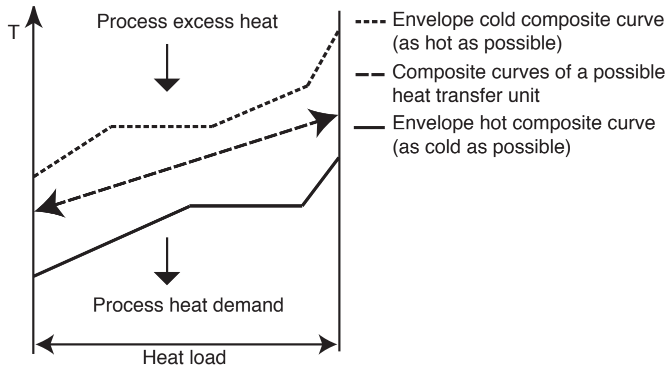


Figure 5.4: Example of graphical representation of the envelope composite curves

The fictive hot and cold heat loads ($\dot{\boldsymbol{Q}}_{\boldsymbol{h,env,k}}$,$\dot{\boldsymbol{Q}}_{\boldsymbol{c,env,k}}$) are added as decision variables in the MILP formulation to calculate the envelope composite curves. The problem is resolved considering all process and utility streams including the fictive hot and cold streams for the envelope composite curves.

However, auxiliary constraints have to be added, to avoid completely the energy penalty of restricted matches. For this, the MILP problem without constraints (Section 3.4.4.1) is first solved in order to obtain the utility flow-rates that minimize the yearly operating costs. Then, two options are possible:

1. The first alternative is to fix the utility flow-rates to the values, which correspond to the calculated flow-rates of the case without constraints (Equation (5.14)). δ is a numerical precision parameter for the optimization.

$$f_{u_o} - \delta \leq \boldsymbol{f_u} \leq f_{u_o} + \delta \tag{5.14}$$

   The degree of freedom analysis however shows that at least one utility flow-rate can not be fixed since it will be calculated to close the system balance.

2. The second alternative is to limit the optimized value for the yearly operating costs. This value can then be used as a maximum bound constraint of the envelope problem formulation (Equation (5.15)).

$$d \cdot (\sum_{f=1}^{nf}(c_f^+ \sum_{u=1}^{nu} \boldsymbol{f_u} \boldsymbol{\dot{E}^+_{f,u}}) + c_{el}^+ \boldsymbol{\dot{E}^+_{el}} - c_{el}^- \boldsymbol{\dot{E}^-_{el}} + \sum_{u=1}^{nu} \boldsymbol{f_u} c_u) \leq OpC_o + \delta \tag{5.15}$$

The results of both approaches are similar for the tested case studies. For the example presented below the first option has been chosen to solve the problem.

Furthermore the original objective function has to be modified like it is shown in Equation (5.16).

$$\begin{aligned} F_{obj} = min(\kappa \cdot d \cdot (\sum_{f=1}^{nf}(c_f^+ \sum_{u=1}^{nu} \boldsymbol{f_u} \dot{E}^+_{f,u}) + c_{el}^+ \boldsymbol{\dot{E}^+_{el}} - c_{el}^- \boldsymbol{\dot{E}^-_{el}} + \sum_{u=1}^{nu} \boldsymbol{f_u} c_u) \\ + \sum_{k=1}^{nk} \boldsymbol{\dot{Q}_{h,env,k}} + \sum_{k=1}^{nk} \boldsymbol{\dot{R}_k}) \end{aligned} \tag{5.16}$$

The new objective function is composed of three terms that aim at

- minimizing the yearly operating costs of the energy conversion system and therefore maximizing the heat recovery (κ is a weighting factor)
- minimizing the heat load transferred by the fictive hot and cold streams, representing the heat transfer envelope
- minimizing the cascaded heat; This makes the fictive cold streams as hot as possible and the fictive hot streams as cold as possible.

The last two terms are important to be able to draw the envelope composite curves, however the operating costs must also be included in the objective function, in order to calculate the heat cascade. Since either some of the multiplication factors or the operating costs are bounded, the first term becomes less important and therefore the weighting factor κ has to be chosen rather small but significantly higher than 0 (e.g. in our case $10e^{-7}$).

The graph of hot and cold envelope composite curves allows to define the acceptable temperature enthalpy profiles of heat transfer units in the corrected temperature domain. An example of envelope composite curves is given in Figure 5.9. In order to minimize the energy penalty due to restricted matches, the hot and cold composite curves of the heat transfer system have to be embedded between the hot and cold envelope composite curves. The envelope composite curves will feature as many pinch points as the process without heat exchange restrictions including the one generated by the utility streams. Each section (between two pinch points) of the envelope curves can therefore be analyzed separately. Consequently, the heat transfer fluids have to be defined in a way that the independence of these sections is maintained. Once the optimal temperature profiles are known, appropriate indirect heat transfer units can be chosen and integrated with the MILP formulation presented in Section 5.3.1. To sum up, the complete mathematical formulation for the envelope composite curves consists in the new objective function (Equation (5.16)), Equations (3.10) to (3.13), the heat cascade equations of sub-systems (Equations (5.1) and (5.2)) and the new heat cascade for the heat transfer system (Equations (5.9) to (5.13)). Additional equations are necessary. For the first alternative (fixing the flow-rates of utilities) Equation (5.14) is added while for the second alternative (fixing the operating costs) Equation (5.15) has to be added.

5.3.3 Multi-objective optimization - choice of intermediate heat transfer networks

When several heat transfer units are possible, or when the temperature levels of the heat transfer units are not precisely identified, a non-linear programming approach can be interesting to choose among several heat transfer units and to define the most appropriate one. This can be done by a multi-objective optimization approach, based on an evolutionary algorithm (Molyneaux et al., 2010). The chosen strategy is adapted from the decomposition of the optimization problem in master and slave sub-problems as presented in (Gassner and Maréchal, 2009). Figure 5.5 shows the optimization algorithm, which is similar to the approach presented in Section 4.2.3 for the systematic heat pump integration. The decisions variables of the master problem are the temperature conditions of the heat transfer units (e.g. networks). Their ranges can be deduced from the analysis of the envelope composite curves. In our example four networks have been modeled for being integrated, however the approach is generic and can be extended to a higher number of networks. The two objectives are minimizing the operating cost (evaluated by the cost for the natural gas, electricity or cooling water) and minimizing the investment cost (evaluated by Equation (5.18)).

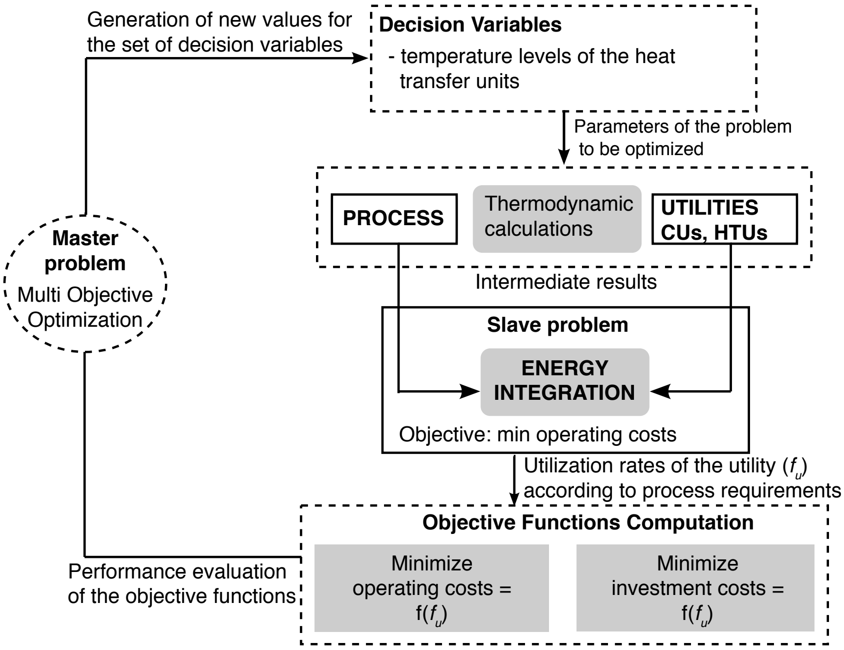


Figure 5.5: Multi-objective optimization algorithm

The pumping costs of the heat transfer units are included in the operating costs, in order to ensure that the flow-rates and operating temperatures of heat transfer units are optimal and to distinguish between different networks options. As presented in 5.3.1, the slave problem contains the MILP problem which minimizes the operating costs (OpC) defined by Equation (5.17).

$$OpC = F_{obj} = min(d \cdot (\sum_{f=1}^{nf}(c_f^+ \sum_{u=1}^{nu} \boldsymbol{f_u} \dot{E}_{f,u}^+) + c_{el}^+ \dot{\boldsymbol{E}}_{\boldsymbol{el}}^+ - c_{el}^- \dot{\boldsymbol{E}}_{\boldsymbol{el}}^- + \sum_{u=1}^{nu} \boldsymbol{f_u} c_u)) \tag{5.17}$$

The investment costs (InvC) are estimated according to available correlations shown in Section 3.4.5. Once the flow-rates are defined in the slave problem, the vertical heat exchange area (A) and the minimum number of heat exchange connections (N_{min}) are calculated. Then, the mean area (A_{mean}) is computed and the investment costs of the heat exchangers are estimated with Equation (5.18). Following values have been considered: $InvC_{ref} = 6300$ [€] is the reference cost for the reference area $A_{ref} = 1$ [m^2] and $\gamma = 0.65$.

$$InvC = N_{min} \cdot InvC_{ref} \cdot \frac{A_{mean}}{A_{ref}}^{\gamma} \qquad [Euro] \tag{5.18}$$

As a result after a given number iterations, the Pareto front shows optimal solutions in terms of operating and investment costs. The final solution can be chosen among optimal solutions. One major disadvantage is that the multi-objective optimization can be time consuming.

Instead of using a multi-objective optimization approach, the problem could also be solved by a single objective function based on the annualized profit or the net present value. But by varying the operating conditions in the heat transfer units, different pinch points can be activated and the problem becomes non-linear and discontinuous. Hence, the problem will be more difficult to solve using mathematical programming methods. An alternative is the chosen approach using an evolutionary algorithm. It can be quite inefficient for solving a single objective function, but it is interesting for multi-objective problems. Moreover it allows generating a set of solutions instead of one single solution. A better and in most case more realistic final solution can be selected from the Pareto front by applying other criteria (e.g. financial, environmental).

5.3.4 Extension to multi-level sub-systems

The previous introduced concept of restricted matches between sub-systems can be extended to multi-level sub-systems (Figure 5.6). The definition of sub-systems inside sub-systems become possible. For each level a heat transfer system is necessary. It consists in units which can exchange heat with all sub-systems of the corresponding level. The global problem is represented by the last sub-system which contains all sub-systems and the global heat transfer system with no heat exchange restrictions. For each sub-system s the heat cascade is given by Equations (5.19) to (5.21).

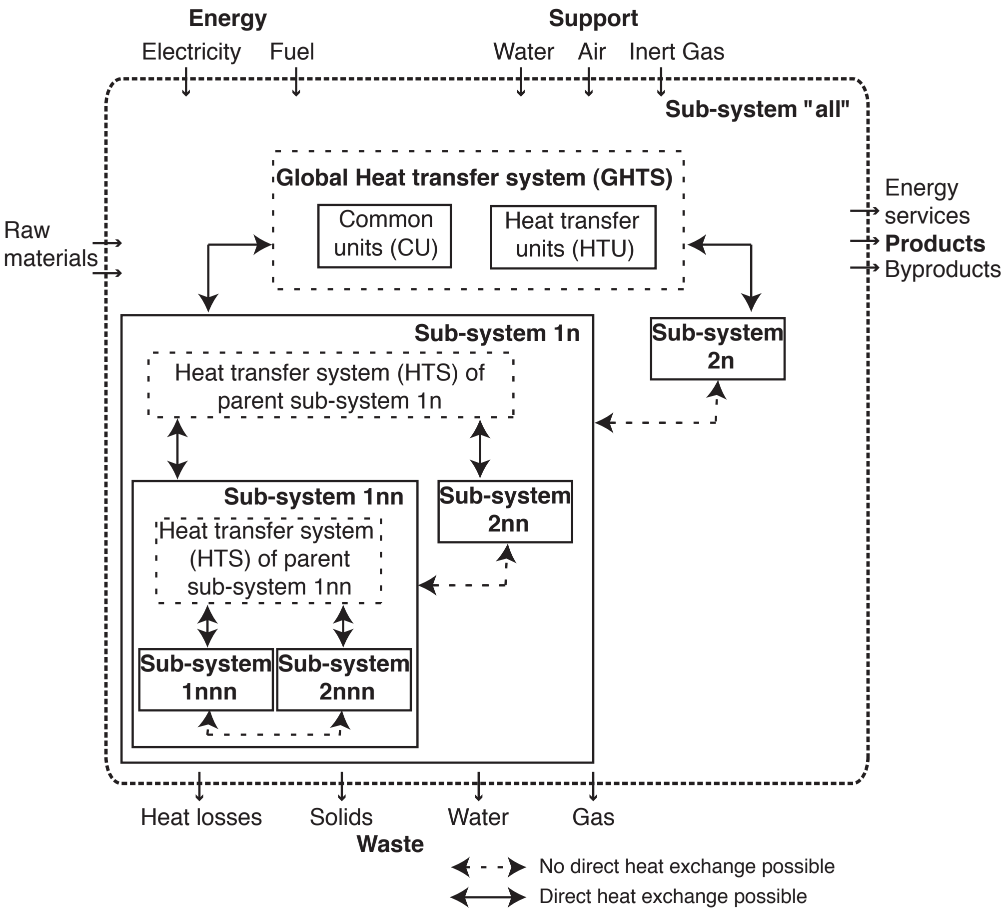


Figure 5.6: Definition of sub-systems with several levels

$$\sum_{h_{s,k}=1}^{ns_{h,s,k}} \boldsymbol{f_u} \dot{Q}_{h,s,k,u} - \sum_{c_{s,k}=1}^{ns_{c,s,k}} \boldsymbol{f_u} \dot{Q}_{c,s,k,u} + \dot{\boldsymbol{Q}}^{-}_{\boldsymbol{hts(s),s,k}} - \dot{\boldsymbol{Q}}^{+}_{\boldsymbol{hts(s),s,k}} + \dot{\boldsymbol{R}}_{\boldsymbol{s,k+1}} - \dot{\boldsymbol{R}}_{\boldsymbol{s,k}} = 0 \quad \forall k = 1..., nk \quad \forall s = 1..., nsub \tag{5.19}$$

$$\dot{\boldsymbol{R}}_{\boldsymbol{s,1}} = 0 \quad \dot{\boldsymbol{R}}_{\boldsymbol{s,nk+1}} = 0 \qquad \dot{\boldsymbol{R}}_{\boldsymbol{s,k}} \geq 0 \qquad \forall k = 2..., nk \quad \forall s = 1..., nsub \tag{5.20}$$

$$\dot{\boldsymbol{Q}}^{+}_{\boldsymbol{hts(s),s,k}} \geq 0 \quad \dot{\boldsymbol{Q}}^{-}_{\boldsymbol{hts(s),s,k}} \geq 0 \qquad \forall k = 1..., nk \quad \forall s = 1..., nsub \tag{5.21}$$

$\dot{Q}_{h,s,k,u}$ is the nominal heat load of hot stream h in sub-system s and interval k and belonging to unit u. The real heat load is calculated with the multiplication factor $\boldsymbol{f_u}$. When a sub-system has a deficit or a surplus of heat in the temperature interval k, the heat is supplied from the heat transfer system of its sub-system ($\dot{\boldsymbol{Q}}^{-}_{\boldsymbol{hts(s),s,k}}$) or respectively removed by the same heat transfer system ($\dot{\boldsymbol{Q}}^{+}_{\boldsymbol{hts(s),s,k}}$). $\dot{\boldsymbol{R}}_{\boldsymbol{s,k}}$ is the cascaded heat to the lower temperature interval k in sub-system s.

The heat cascade for the heat transfer system hts for each parent sub-system is given by Equations (5.22) to (5.23). $\dot{\boldsymbol{Q}}^{-}_{\boldsymbol{hts+1,k}}$ is the heat supplied from the heat transfer system of the level above the sub-system s and $\dot{\boldsymbol{Q}}^{+}_{\boldsymbol{hts+1,k}}$ is the heat transferred from the sub-system s to the heat transfer system of the higher level.

$$\sum_{h_{hts,k}=1}^{ns_{h,hts,k}} \boldsymbol{f_u} \dot{Q}_{h,hts,k,u} - \sum_{c_{hts,k}=1}^{ns_{c,hts,k}} \boldsymbol{f_u} \dot{Q}_{c,hts,k,u} + \dot{\boldsymbol{Q}}^{-}_{\boldsymbol{hts+1,k}} - \dot{\boldsymbol{Q}}^{+}_{\boldsymbol{hts+1,k}} - \sum_{s=1}^{nsub(hts)_k} \dot{\boldsymbol{Q}}^{-}_{\boldsymbol{hts(s),s,k}} + \sum_{s=1}^{nsub(hts)_k} \dot{\boldsymbol{Q}}^{+}_{\boldsymbol{hts(s),s,k}} + \dot{\boldsymbol{R}}_{\boldsymbol{hts,k+1}} - \dot{\boldsymbol{R}}_{\boldsymbol{hts,k}} = 0 \quad \forall k = 1..., nk \quad \forall hts = 1..., nps \tag{5.22}$$

$$\dot{\boldsymbol{R}}_{\boldsymbol{hts,1}} = 0 \quad \dot{\boldsymbol{R}}_{\boldsymbol{hts,nk+1}} = 0 \quad \dot{\boldsymbol{R}}_{\boldsymbol{hts,k}} \geq 0 \quad \forall k = 2, ..., nk \quad \forall hts = 1..., nps \tag{5.23}$$

Like before, to ensure that heat is cascaded correctly, a second set of Equations for the global heat transfer system is necessary. Equations (5.24) and (5.25) express respectively the heat balance of the hot and cold streams in the heat transfer system. The flow-rates of the heat transfer units have to be optimized in order to satisfy the remaining heat demand of all sub-systems.

$$\sum_{h_{hts,k}=1}^{ns_{h,hts,k}} \boldsymbol{f_u} \dot{Q}_{h,hts,k,u} + \dot{\boldsymbol{Q}}^{-}_{\boldsymbol{hts+1,k}} + \dot{\boldsymbol{R}}_{\boldsymbol{hts,k+1}} - \dot{\boldsymbol{R}}_{\boldsymbol{hts,k}} - \sum_{s=1}^{nsub(hts)_k} \dot{\boldsymbol{Q}}^{-}_{\boldsymbol{hts(s),s,k}} \geq 0 \quad \forall k = 1..., nk \quad \forall hts = 1..., nps \tag{5.24}$$

$$- \sum_{c_{hts,k}=1}^{ns_{c,hts,k}} \boldsymbol{f_u} \dot{Q}_{c,hts,k,u} - \dot{\boldsymbol{Q}}^{+}_{\boldsymbol{hts+1,k}} + \dot{\boldsymbol{R}}_{\boldsymbol{hts,k+1}} - \dot{\boldsymbol{R}}_{\boldsymbol{hts,k}}$$
$$+ \sum_{s=1}^{nsub(hts)_k} \dot{\boldsymbol{Q}}^{+}_{\boldsymbol{hts(s),s,k}} \leq 0 \qquad \forall k = 1..., n_k \quad \forall hts = 1..., nps \tag{5.25}$$

5.4 Numerical example - drying process in the paper industry

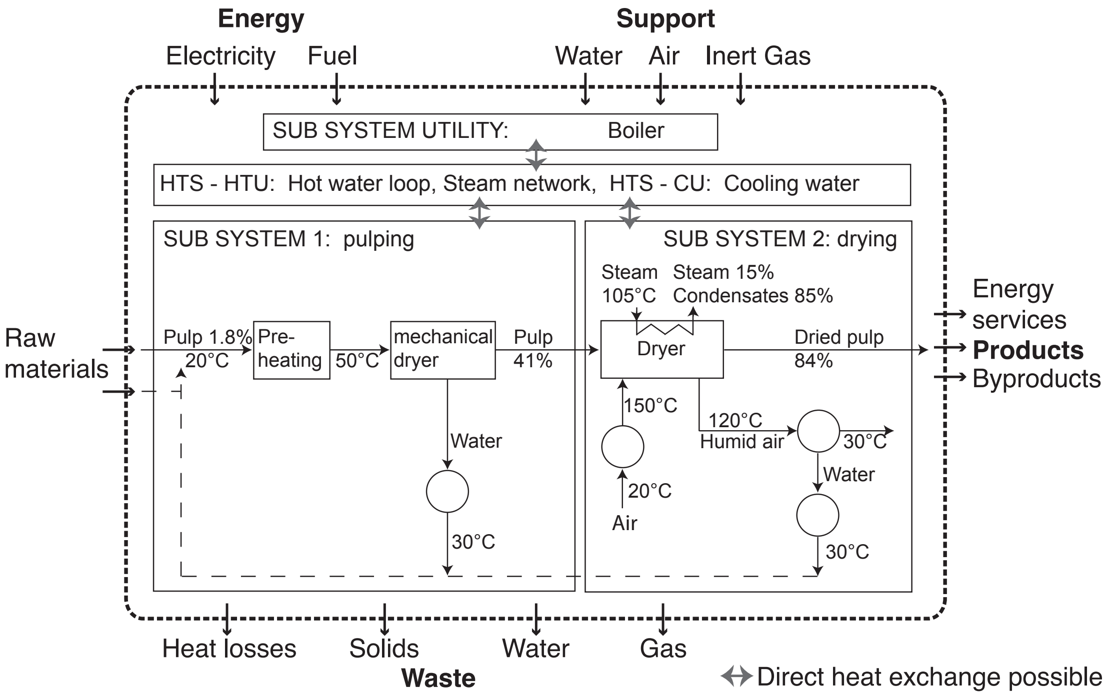


Figure 5.7: Representation of the process

In order to illustrate the application of the method, a dryer process of a paper production plant will be studied (Figure 5.7).

The humid pulp is first preheated in the pulping unit before entering the dryer unit. Currently steam, produced by a natural gas fired boiler, is defined as a hot utility. It is used for drying paper mill rolls and for producing hot air, which is mainly used to evacuate the evaporated water from the pulp. Possible heat recovery is introduced by a humid air stream (hot stream which has to be cooled down to the final temperature of 30°C). The list of process streams is given in Table 5.2. The pulping unit (sub-system 1), drying unit (sub-system 2) and later the boiler (sub-system utility) are considered as different sub-systems. Heat can not be exchanged directly between these sub-systems. Thus, in absence of any heat transfer unit, the heat demand of sub-system 1 can only be satisfied by an external hot utility, even if the excess heat of sub-system 2 is sufficient to satisfy the demand (Figure 5.7).

Table 5.2: Process streams with $\Delta T_{min}/2$ values: 2°C (liquids), 0.5°C (gases)

Unit	Name	Tin [°C]	Tout [°C]	Heat load [kW]	Remarks
pulping	ph_c1	20	50	11262	Preheating
	ph_h1	50	30	7297	Water cooling
drying	st_c1	95	105	6057	Steam demand
	st_h3	105	105	892	Condensation of 15% steam
	st_h2	105	95	112	Cooling of condensates
	air_c1	20	150	664	Air heating
	air_h1	100	30	5278	Humid air cooling

In the following, each step of the methodology is presented for the example. Except the multi-objective optimization, all problems are MILP problems. The problem size is reported in the summary of results (Table 5.8).

Located in France, the fuel price (natural gas) and the electricity price (purchase price) are considered to be 0.0392 €/kWh and 0.0620 €/kWh$_{el}$ respectively. The exported electricity can be sold for 0.0496 €/kWh$_{el}$. The cost of cooling water is small, since a near located river can be used as a cold source.

5.4.1 Step 1: Minimum utility cost without restricted matches

The MILP formulation presented in Section 3.4.4.1 is used to integrate the process and utility units. With the steam boiler and cooling water two utility units are proposed to the process. This corresponds to two integer variables in the MILP problem. The characteristics of the steam boiler and cooling water are briefly described below. The boiler produces steam by taking heat from the flue gases at a temperature higher than 1000 °C and cooled down to the stack temperature (120 °C). Also air preheating from the ambient (20 °C) to the stack temperature is included. In order to better show the exergy losses, the boiler has been modeled with the fumes. On the other hand, the cooling water enters the system at 7 °C. Considering the streams and their ΔT_{min} values of Table 5.2, the maximum heat recovery leads to a natural gas consumption of 6073 kW and a cooling water consumption of 1668 kW. It corresponds without restricted matches to the minimum operating costs (utility costs) of 2.4 M€/year.

5.4.2 Step 2: Calculation and visualization of the energy penalty due to restricted matches

The energy penalty due to restricted matches can be evaluated using the MILP formulation presented in Section 5.3.1. For this calculation, the two utility units (steam boiler and cooling water) are defined as common units (CUs) in the heat transfer system. They are allowed to exchange heat with the process units (sub-system 1 & 2). The penalty, which consumes by the same amount (about 3850 kW) more of both the hot and the cold utility, can also be visualized when comparing the integrated composite curve of the utility system in both configurations (Figure 5.8).

The results are compared in Table 5.7.

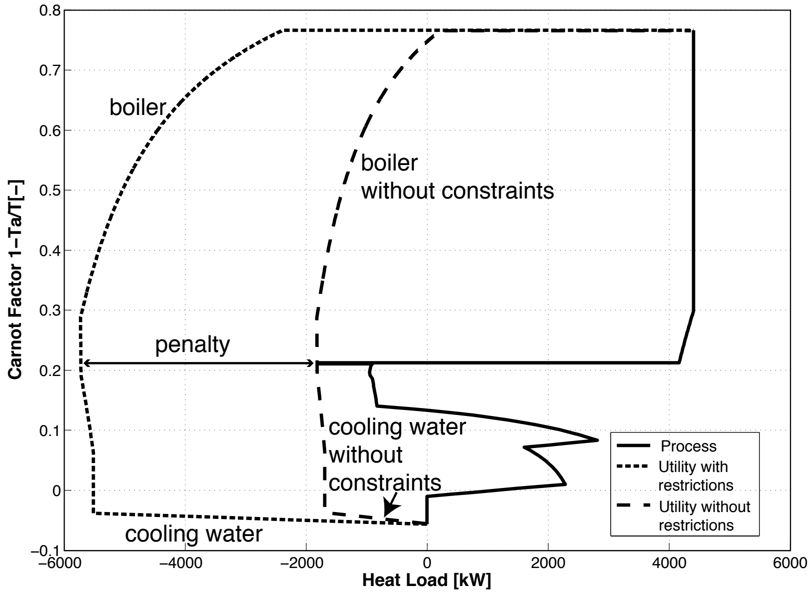


Figure 5.8: Penalty of the system

5.4.3 Step 3: Hot and cold envelope composite curves and choice of intermediate heat transfer units

In the example, the pulping and drying sub-systems cannot exchange heat directly. The flue gases of the boiler are also defined as a sub-system which cannot exchange heat directly with the process.

It is possible to find optimal heat transfer units by analyzing the required temperature levels of the process demand. However this is not evident when more sub-systems are defined. In such cases, the envelope composite curves help to identify optimal intermediate heat transfer units.

Considering the multiplication factor of the boiler (calculated in step 1) and the heat exchange constraints between sub-systems a second problem can be solved, using the MILP formulation presented in Section 5.3.2. With this, the envelope composite curves can be visualized (Figure 5.9). According to the pinch point locations, two separate heat transfer units are necessary: one to transfer the heat from the boiler to the process demand above the pinch point (105 °C) and another one to transfer heat between sub-system 1 and 2 to make heat recovery possible below the pinch point.

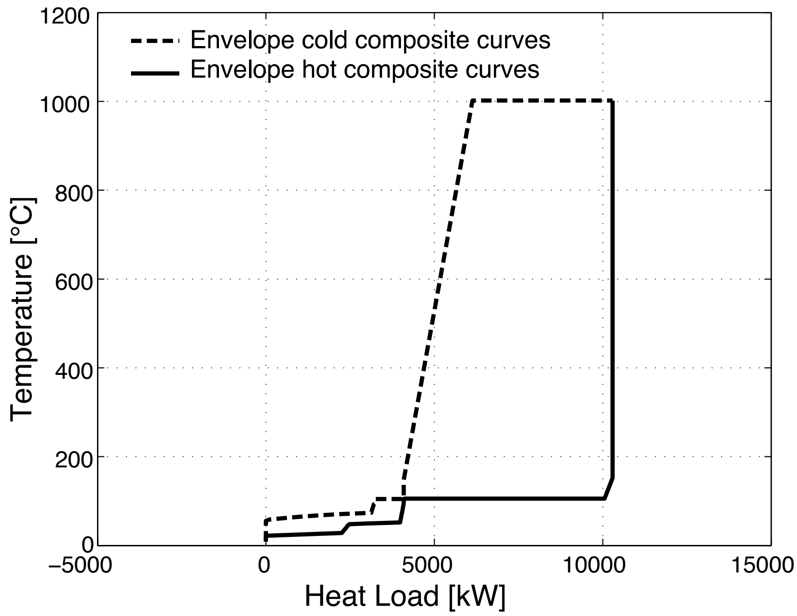


Figure 5.9: Envelope composite curves for intermediate heat transfer units

5.4.4 Step 4: Restricted matches and integration of intermediate heat transfer units

The definition of the heat transfer system is based on the temperature levels of the envelope composite curves (Figure 5.9) but also technological constraints have to be considered. To transfer heat from the boiler to the process, a simple steam network could transfer heat by producing steam in the boiler and returning condensates after heat exchange with the process. It is also possible to consider combined heat and power production in the steam network by using high pressure steam from the boiler in steam turbines to produce lower pressure steam delivering heat to the process at lower temperatures. Because of a better exergy efficiency the second option is preferred and integrated in the following. The steam network acts as heat transfer unit between the boiler (steam production at 80 bar, 295 °C) and the process demand (steam utilization at 7 bar, 165 °C and 2 bar, 120 °C). Mainly, the process heat demand requires a temperature of 105°C, but a small part requires a higher temperature level up to 150 °C. On the envelope composite curves it can also be seen that two steam extraction pressures are needed (Figure 5.10).

It is important to remark that even without constraints, the energy integration will choose the steam network when it is useful to use this electricity in the process or when the selling price is attractive.

For the second network below the pinch point, an intermediate hot water loop can be integrated. In this case, water is heated up from 25°C to 80°C with streams from the drying unit and this heat is given back to the pulping unit by cooling down the water from 80°C to 25 °C. The pumping costs are included, in order to size correctly the recovery loop.

The summary of selected temperature levels for the heat transfer units is given in Figure 5.10. With the new utility units for indirect heat transfer, the targeting problem (MILP formulation

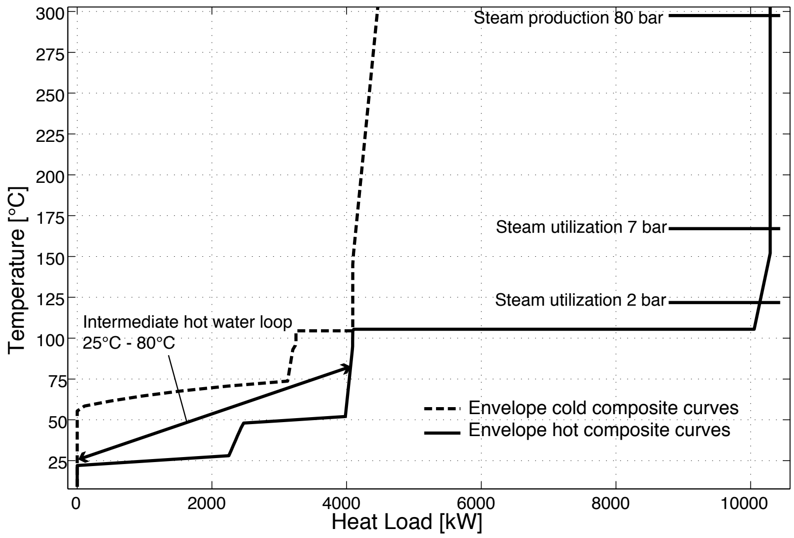


Figure 5.10: Choice of temperature levels of intermediate heat transfer units

from Section 5.3.1) is solved. The multiplication factors of the utility units (steam boiler, cooling water, steam network and intermediate heat recovery loop) are calculated to minimize the cost of the energy conversion system, while satisfying the restricted matches constraints. As the utility streams depend on the combined heat and power production, the multiplication factor may differ from the one calculated without constraints. For example, the utilization rate of the boiler will be higher when a steam network, producing electricity, is integrated.

The integrated utility composite curves, including a steam network at higher temperature and an intermediate hot water loop below the pinch point, are shown in Figure 5.11. The overall system is optimized and the energy penalty due to heat exchange restrictions is minimized.

5.4.5 Step 4a (optional): Multi-objective optimization and choice of intermediate heat transfer units

The application of the method presented in Section 5.3.3 to the example above leads to the Pareto of Figure 5.12. Approximative solutions are obtained after 3000 evaluations. Considering the results of the envelope composite curves, the decision variables have been chosen in order to accept the minimal and maximal temperature level of the intermediate heat recovery loop (25 °C and 100 °C respectively). The variation in the operating cost is small, because the only difference comes from the pumping cost which is proportional to the mass flow-rate. For a higher temperature difference, the pumping cost becomes smaller. But on the other side the investment costs become higher because the necessary heat exchanger area increases.

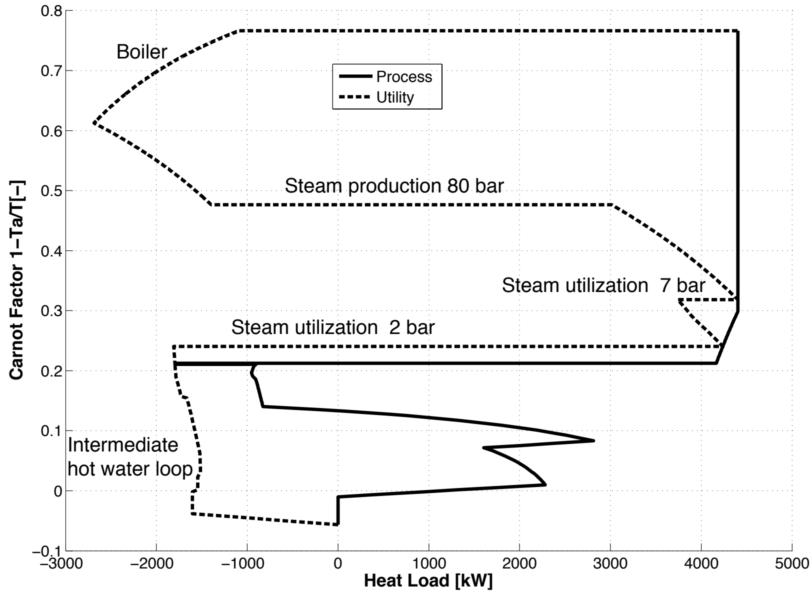


Figure 5.11: Integrated utility composite curves

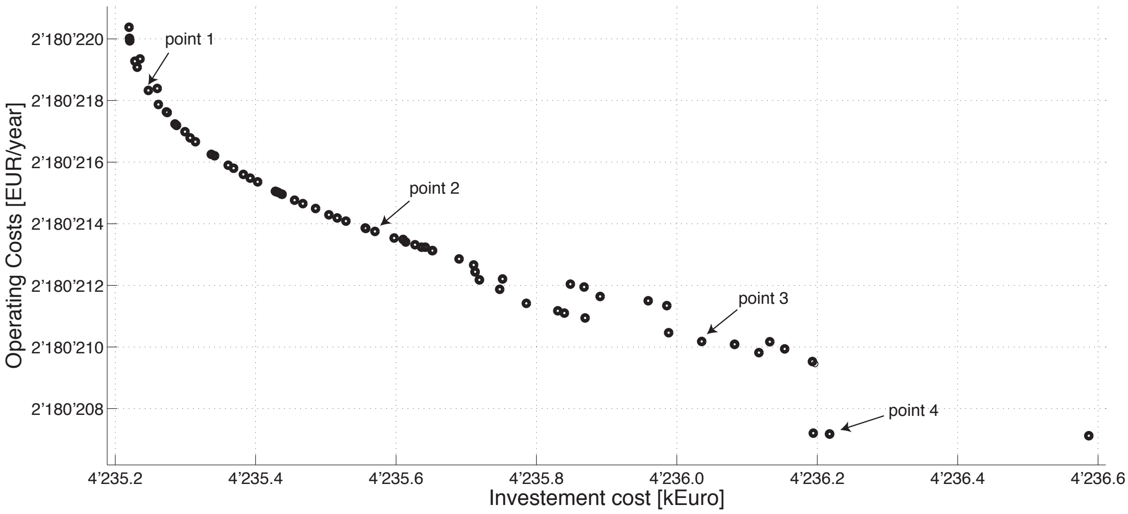


Figure 5.12: Pareto front to select intermediate heat transfer networks

Table 5.3 shows the results of 4 points on the Pareto curve and compares it with the manual (man) chosen intermediate loop (T_{low} = 25 °C, T_{up} = 80 °C), by studying the envelope composite curves.

For this application example, the multi-objective optimization approach cannot provide much additional information, when comparing with the results of the envelope approach. However for more complex case studies, the multi-objective optimization can add interesting information, especially when several networks are possible. But on the other side it is quite time consuming.

In the next step, the intermediate heat recovery network is integrated for the manual solution.

Table 5.3: Results of multi-objective optimization

Point	OpC [€/year]	InvC [k€]	T_{low} [°C]	T_{up} [°C]
1	2 180 218	4,2353	49.3	69.5
2	2 180 214	4,2356	46.3	73.4
3	2 180 210	4,2361	40.0	78.0
4	2 180 207	4,2362	25.4	81.6
man	2 180 207	4,2363	25.0	80.0

Table 5.4: Heat load distribution for zone 1 (119 °C - 1000 °C)

Hot stream	Cold stream	Heat load [kW]
D2_C_Ds	drying_air_c1	166.7
boiler_boi_h1	boiler_boi_c1	85.0
boiler_boi_h1	C_H1_Cs	5409.3
boiler_boi_h2	C_H1_Cs	2786.4
D2_C_Ds	C_H1_Cs	555.9

5.4.6 Step 5: Heat load distribution

Finally, the heat load distribution (3.4.4.3) is computed for the case with integrated steam network and a hot water loop from 25 °C to 80 °C. The complete heat load distribution for this example is shown in Figure 5.13. 3 different zones between pinch points (utility pinch point at 9 °C, process pinch point at at 105 °C, utility pinch point at 119 °C and utility pinch point at 1002 °C) can be distinguished. The heat load distribution for these 3 zones is also summarized in Tables 5.4 to 5.6. They show the exchanged heat amount between a hot and a cold stream. The results can be later used to design the heat exchanger network.

Table 5.5: Heat load distribution for zone 2 (105 °C - 119 °C)

Hot stream	Cold stream	Heat load [kW]
D1_C_Ds	drying_air_c1	75.9
D1_C_Ds	boiler_boi_c1	29.1
D2_C_Ds	boiler_boi_c1	20.1
D1_C_Ds	drying_st_c1	5956.8
boiler_boi_h2	C_H1_Cs	46.9
D1_C_Ds	C_H1_Cs	156.4

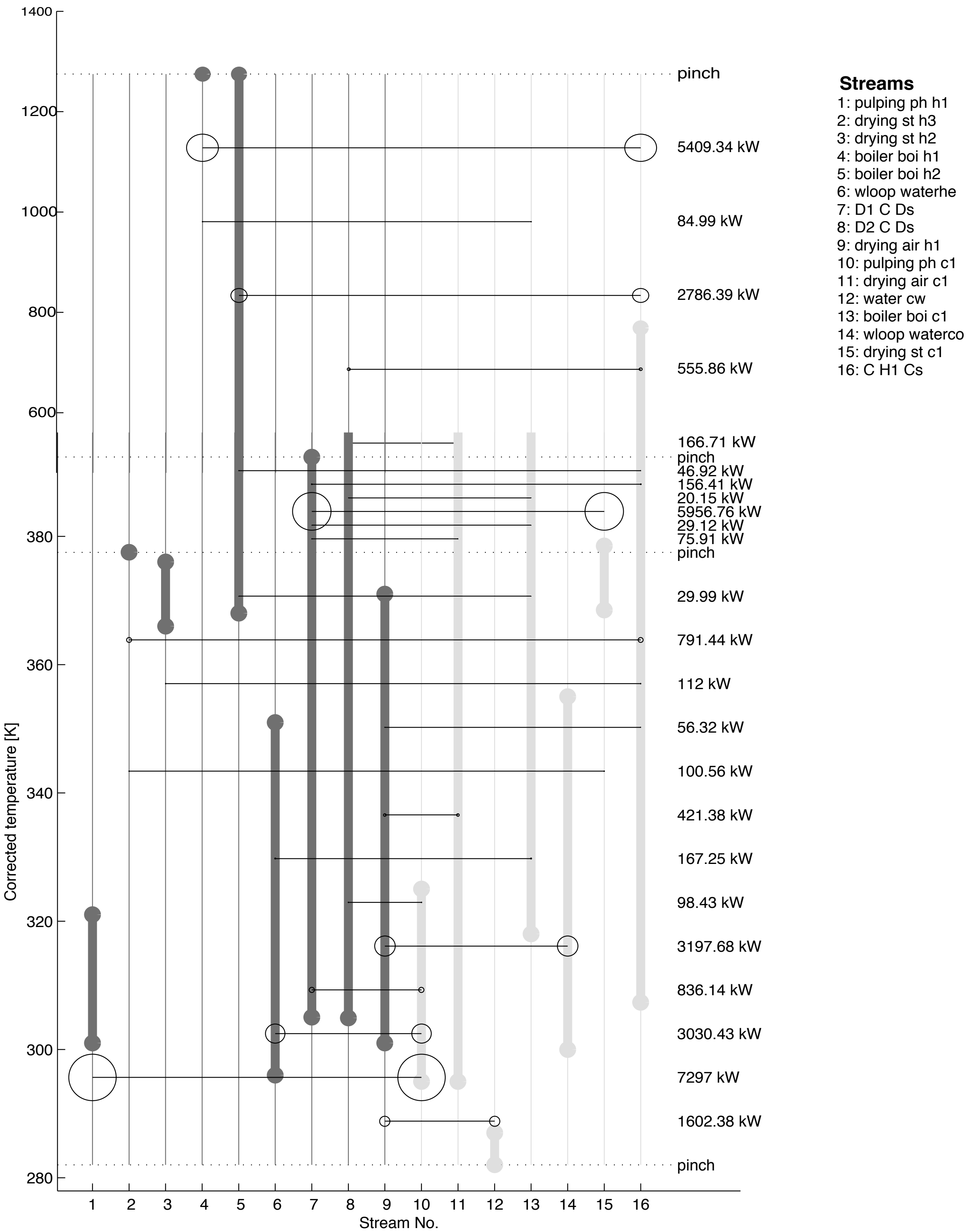


Figure 5.13: Heat load distribution

Table 5.6: Heat load distribution for zone 3 (9 °C - 105 °C)

Hot stream	Cold stream	Heat load [kW]
pulping_ph_h1	pulping_ph_c1	7297.0
wloop_waterhe	pulping_ph_c1	3030.4
D1_C_Ds	pulping_ph_c1	836.1
D2_C_Ds	pulping_ph_c1	98.4
drying_air_h1	drying_air_c1	421.4
drying_air_h1	water_cw	1602.4
boiler_boi_h2	boiler_boi_c1	30.0
wloop_waterhe	boiler_boi_c1	167.3
drying_air_h1	wloop_waterco	3197.7
drying_st_h3	drying_st_c1	100.6
drying_st_h3	C_H1_Cs	791.4
drying_st_h2	C_H1_Cs	112.0
drying_air_h1	C_H1_Cs	56.3

5.4.7 Summary of results & discussion

The summary of all results is given in Table 5.7. For better comparison the net heat delivered to the process by the boiler is reported. The corresponding natural gas consumption, which takes into account the boiler efficiency is calculated, but not included in Table 5.7 . Compared to the case without constraints, the same amount of energy has to be added to the boiler and the cooling water when constraints are integrated. In the case of integrating intermediate heat transfer units (hot water loop and steam network), the boiler consumption increases but at the same time electricity is produced. Table 5.8 compares the number of constraints and variables and the computation time

Table 5.7: Results

	Unit	No constraints	With constraints	Constraints and heat transfer system
Operating Costs	[k€/year]	2353.6	3844.8	2180.2
Heating consumption	[kW]	6073	9920	8026
Cooling water	[kW]	1668	5516	1602
Electricity	[kW]			-2019

for the MILP problems. Introducing restricted matches increases the number of constraints and variables and also the computation time.

For this example the complete piping and investment costs have not been included. Pumping costs, proportional to flow-rates, are included in the electricity consumption of intermediate heat transfer networks (here recovery loop and steam network). However the complete piping costs can not easily

Table 5.8: Problem size

MILP Problem	Constraints	Variables	Computation time
No constraints	133	125	0.016 sec
With constraints	499	385	0.031 sec
Envelope composite curves	834	676	0.047 sec
Constraints & heat transfer system	764	674	0.047 sec
Heat load distribution	1679	1399	0.203 sec

be integrated in the proposed targeting method, as they depend not only on the flow-rates but also on the distance between interconnected units.

The investment costs for pipes could be estimated after having applied the targeting method and calculated the heat load distribution and the heat exchanger network. The heat exchange connections will be known and the distance between hot and cold streams could be determined.

To include the investment cost in the objective function of the optimization, the heat load distribution with all possible heat exchange connections has to be established in order to be able to consider the interconnections as cost. For this, Weber et al. (2006) have developed a method for the system design. Thus, the next step of our analysis could be the application of this method to the pulp and paper example.

For the presented example, not considering the investment costs of steam network (CHP unit) gives good solutions in a first step, since the boiler and process units have been defined in sub-systems, so that intermediate heat transfer units are necessary and only the steam network is proposed to the process.

The presented method is optimizing the operating costs, but the annualized investment of the steam network and other utility units could easily be added to the formulation by defining fixed and proportional costs, as demonstrated in Maréchal and Kalitventzeff (2003).

A second possibility is to perform a multi-objective optimization as it is presented in Sections 5.3.3 and 5.4.5 to include the investment costs for new utility units. Optimal solutions representing the trade-off between operating and investment costs are located on the resulting Pareto curve.

5.5 Other useful applications

Restricted matches can be useful for the utility integration (e.g. a heat pump which cannot exchange heat directly with the process and/or other utilities). The approach can also be used to separate units which do not work simultaneously (e.g. batch problems).

As presented in Section 5.3.4, industrial processes can be defined in several levels of sub-systems.

5.5.1 Problem statement of cheese factory with restricted matches - pseudo multi-period

To illustrate multiple levels of restrictions, the example of a cheese factory (Appendix B.2) with heat exchange restrictions and non-simultaneous processes is now presented.

To make batch processes continuous and easier to solve, Linnhoff et al. (1988) introduced the time average approach. Thus, the energy consumption will be expressed by a mean value related to the energy consumption of 1 ton of cheese (kWh/t).

Heat exchanges become more difficult for multi-period processes as they occur indirectly through heat transfer units and storage tanks. For this, the units working at different periods are defined as independent sub-systems. The heat transfer between those sub-systems can only be realized by intermediate heat transfer networks. For the cheese factory, the following process units are not always working simultaneously and are therefore defined in sub-systems with no possibility of direct heat exchange: evapo, pasto 1-5 and proc6.

Furthermore, the $\Delta T_{min}/2$ has to be adapted to include the fact that the instantaneous heat load will be higher than the mean heat load. Thus, a higher heat exchange area and investment cost for the same heat amount will be required. The instantaneous heat load ($\dot{Q}^*_{h/c}$) of a hot or cold stream belonging to unit u can be expressed by Equation (3.5). $\dot{Q}_{h,c}$ is the mean heat load over all periods, d is the total operating time and d_u is the effective operating time of unit u. The complete list of streams is given in appendix in Table B.10.

5.5.2 Process integration with multi-level sub-systems

First, the maximum heat recovery is computed. Exergy losses can be visualized on Figure B.9a, which shows the hot and cold Carnot composite curves. The process grand composite curve, presented in Figure B.9b, is used to define optimal temperature levels for the energy conversion system (utilities).

Considering the self-sufficient pockets of Figure B.9b, it can be deduced that the heat recovery between process streams concerns process streams below the pinch point. As process streams without heat exchange restrictions are mainly involved, the heat recovery penalty of restricted matches due to non-simultaneously process operations is rather small. The penalty can be evaluated by comparing the case with heat exchange restrictions and no indirect heat recovery to the case without any heat exchange restrictions. The penalty can also be illustrated on the integrated composite curves (5.14), which corresponds to Case 1a (France) of Table B.11. The utility pinch

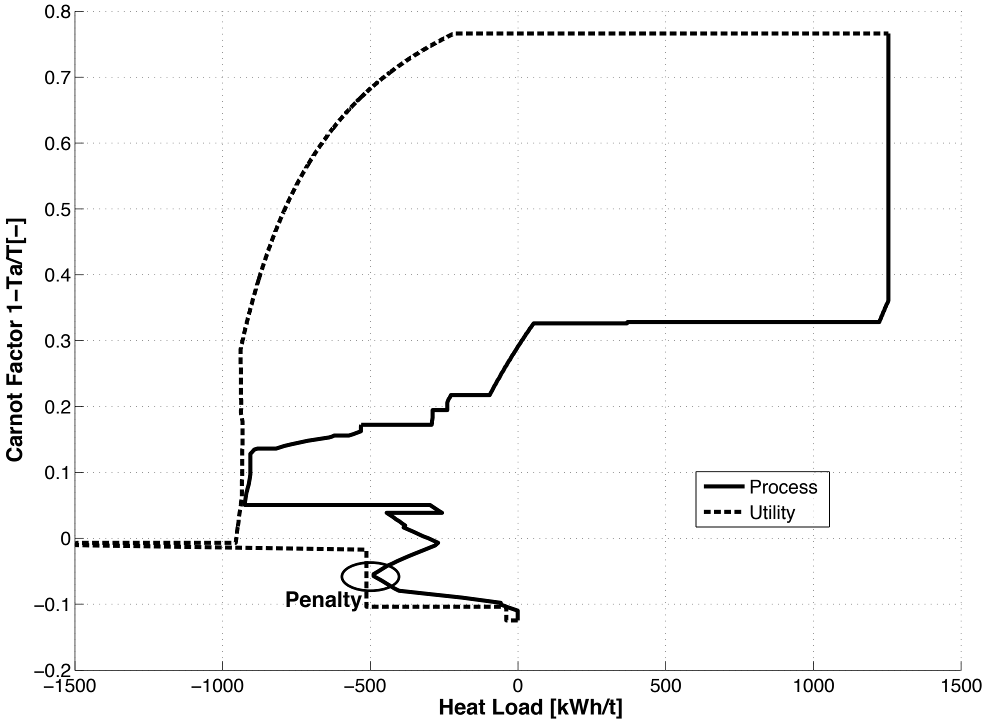


Figure 5.14: Penalty on integrated utility curves of the cheese process

point (marked with the circle on Figure 5.14) is not activated. However, as this penalty is very small, it can also be deduced that the use of intermediate heat transfer networks could only recover a limited amount of heat. The exact penalty due to non-simultaneous operations can be evaluated by drawing the corresponding envelope composite curves (Becker and Maréchal, 2012a). The envelope is given in Figure 5.15. In fact, the penalty of 27 kWh/ t corresponding to 1% of the hot utility

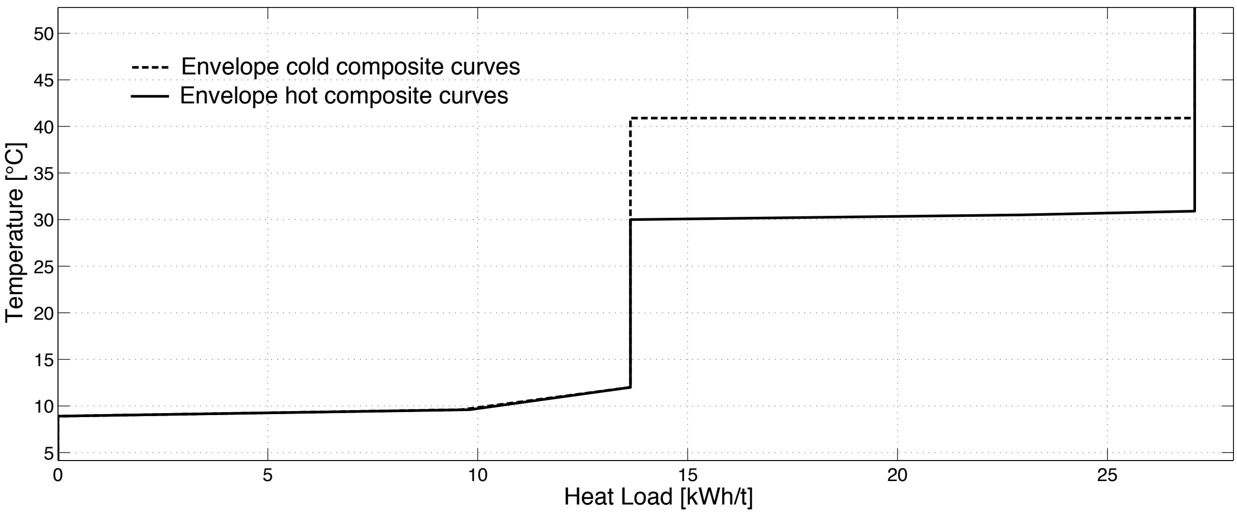


Figure 5.15: Envelope composite curves of the cheese process

and 3% of cold utility, is quite small and can be neglected in the following. Moreover, only half of this energy can be recovered with realistic ΔT_{min} values. This means that no supplementary heat transfer units will be integrated to reduce this penalty. Thus, in other terms, the storage potential of this case study is rather low.

Regarding the necessary temperature levels for heat supply on Figure B.9b, the following utility types are proposed to optimize the energy conversion of the process: currently implemented utilities (conventional steam boiler, refrigeration cycle and cooling water) as well as new ones (closed cycle heat pump and co-generation engine).

The direct heat exchange between a potential heat pump or co-generation engine and the process is not allowed. In the higher level two sub-systems (in this case the process and the heat pump) cannot exchange heat directly. Moreover, heat cannot be exchanged directly between different periods and hence the process operation units of different periods are defined in independent sub-systems (e.g. evaporation unit, pasteurization units) of the sub-system process. Therefore, the evaporation unit can not exchange heat directly with the pasteurization units nor with the heat pump, but heat exchanges with other process units are allowed. To satisfy the required temperature levels, a closed cycle heat pump using the refrigerant R245fa has been chosen. By analyzing the shape of the grand composite curve, appropriate operating conditions for the heat pump and the intermediate heat transfer units can be estimated. The utilities, the heat pump and the heat transfer units are considered simultaneously in the MILP model to define the corresponding optimal flow-rates that minimize the operating costs.

The closed cycle heat pump and the co-generation engine are integrated through an intermediate heat transfer network as they cannot exchange heat directly with the process.

The potential of a closed cycle heat pump without direct heat exchange is summarized in Appendix B.2. Up to 30% of operating cost savings become possible. The temperature difference for the heat sink and heat source of the heat pump is relatively high. Nevertheless the overall operating costs and especially the fuel consumption can be reduced. The saving potential could be increased by minimizing the minimum temperature difference in the heat exchangers between the heat pump and the intermediate heat transfer fluids.

The corresponding investment costs have to be considered. The economic analysis from Table B.11 shows clearly that the integration of a co-generation engine is not profitable in France. Considering for example, case 1c where only a co-generation unit is proposed to the process, it can be seen that the operating costs in Germany would significantly be reduced. Thus, the payback time of 3 years is obtained. On the contrary, in France, the co-generation engine reduces the operating costs only by a small amount. Regarding the high payback time of 11 years, it is improbable that a company would invest in a co-generation unit.

Considering in a second step case 1d where a heat pump and a co-generation unit are proposed to the process. In Germany, both a heat pump and a co-generation engine are integrated and satisfy a part of the hot utility. The engine also satisfies the electricity consumption of the process. As the temperature levels of the co-generation engine is similar to the condensation level of the heat pump, both utilities are in direct competition. On the other side, in France, only a heat pump is integrated. Compared to the German case, this heat pump is bigger and satisfies more of the hot utility.

5.6 Conclusions

A method for targeting the optimal integration of energy conversion systems is proposed. Including heat exchange restrictions, the heat recovery and the combined heat and power production are maximized for an industrial process. The method considers total site integration and defines sub-systems. Between them, heat exchange can not be realized without using heat transfer units. The problem is formulated as a MILP problem that calculates simultaneously the flow-rates of the utility system and the heat transfer units by minimizing the operating costs of the energy conversion system.

The main advantages of the proposed method are summarized below: Compared to conventional pinch analysis based on graphical methods, which assumes that any heat exchange connection is feasible, the problem is formulated as a MILP problem with heat exchange restrictions. Heat transfer units (e.g. steam network but also heat transfer technologies such as heat recovery loops) can be considered and are integrated simultaneously with the energy conversion system (utility units) and the process units. Another advantage of the proposed targeting method is that the self-sufficient pockets are not suppressed and therefore the combined heat and power production is not penalized. The proposed method can be used for plant wide integration (or total site integration) where each plant is defined as a sub-system, but it can also be used inside a plant (e.g. heat exchange restrictions between process operation units because of safety reasons or non-simultaneous process operations).

Furthermore a new MILP formulation is proposed to draw the envelope composite curves. It helps to characterize the heat transfer units, which will minimize the energy penalty due to restricted matches.

The targeting problem, including optimal heat transfer units, defines the complete list of streams to be considered in the heat load distribution problem. The heat exchange constraints can be included and therefore the number of integer variable in the heat load distribution is considerably reduced.

Although the method presented above is illustrated by a simple example with three sub-systems, it enables to solve complex examples with multiple sub-systems (e.g. process units with different locations or other industrial site problems). The method has been applied successfully to two more complex examples from the industry (Brewery process (Dumbliauskaite et al., 2010) with 4 process sub-systems and 55 streams and a cheese factory (Becker et al., 2011c) with 7 process sub-systems and 60 streams).

The sub-system concept is also possible for calculating the integration of utility systems, for example when the produced heat in a boiler cannot exchange directly with process streams and a steam network makes the heat exchange possible.

The use of restricted matches reveals to be a great importance in process integration as demonstrated by (Pouransari et al., 2011). By understanding restricted matches and the required heat transfer units, the proposed tool allows to generate process integration solutions.

Also for large scale integration of energy systems like cities, restricted matches become important. For example each district or location can be considered as a separate sub-system and to connect them, the district heating / cooling systems can be defined as heat transfer units.

The method can also be used to identify the required heat transfer units when solving batch process integration. Applying the time average method, the non-simultaneous operations can be considered in different sub-systems for which the direct heat exchange is not possible. This would be the first step before realizing the detailed integration as presented by Krummenacher et al. (2010).

The proposed formulation is based on the decomposition of the global system into sub-systems. The extension for defining sub-systems inside sub-systems has also been shown. In the case, where only one specific heat exchange is forbidden, the involved hot and cold streams could be defined in two separate sub-systems. But if the decomposition into sub-systems is not possible, the proposed formulation is not applicable anymore and other methods like the one presented by Maréchal and Kalitventzeff (1999) have to be used.

The approach is useful since it does not require the same amount of information than would be necessary to solve a real time dependent heat integration problem, which includes the process schedule. One drawback is that the heat storage and the corresponding equations cannot be included. For this a multi-period and multi-time slice approach has been developed (see Chapter 6).

Chapter 6

Multi-period and batch problems

This chapter focuses on non-continuous process operations. The methodology from the previous chapter is extended to take into account heat exchange restrictions and multi-period or batch problems. Especially the integration of heat storage is considered.
Related publications: Case study: Becker et al. (2011d), Methodology: Becker and Maréchal (2012b).

6.1 Introduction

Process integration was originally developed for continuous processes. But many processes have non-continuous operations and therefore the approach has to be adapted for multi-period problems.

Linnhoff et al. (1988) distinguished 2 different models: Time average model (TAM) and time slice model (TSM). The time average model uses average values over all periods. This approach is correct when it is assumed that all batch operations can take place at any time and in any order or, as it is shown in Chapter 3, where it is assumed that heat storage through water tanks is available. On the other side, the time slice model cuts the process into several periods where the process operations are realized simultaneously. The previous single period approach can be applied for each time slice, without having any link between them. The authors (Linnhoff et al., 1988) proposed a combination of applying both models to find opportunities for saving energy in batch production lines and they suggested rescheduling solutions without proposing a systematic approach.

The problem complexity increases and often the heat integration is realized by using time averaging approaches (e.g. pseudo multi-period of Section 3.4.3.3). The discontinuous processes are formulated as continuous problems assuming that storage tanks are available for heat recovery. The main disadvantages of the time average approach are that the sizing of equipments becomes more difficult and that the investment cost calculation is almost impossible at the targeting stage.

Since the integration of storage tanks is not included their optimal size can not be determined. To overcome these limits, the present chapter will present a mathematical formulation that takes into account time dependences and that enables to treat real multi-period problems.

In the following, batch and multi-period problems will be considered. The distinction between them is not always clear as, for both, the process operations are not simultaneous and depend on the time. Here, multi-period problems will refer to different scenarios or process states. Utility units have to be sized in a way that the process heat demand is satisfied in all operating scenarios, but no time dependence will be considered. Typical examples of multi-period problems are processes with different operating conditions (e.g. seasonal variations). Compared to multi-period problems, batch problems are time dependent and have a shorter operation time (order of minutes or hours). The process operations consists in a list of successive tasks with a given start and a given end time.

As usual, first the literature review is given. Then the problem formulation is developed, and applied on a real case study.

6.2 State of the art

6.2.1 Approaches based on TAM and further developments

Stoltze et al. (1995) proposed a combinatorial method to include the heat storage in the heat exchanger network design. The number of storage tanks and its temperature levels are chosen by analyzing the composite curves. By choosing only a limited number of storage tanks, the authors stated that the computational work is not necessarily higher than for the same problem without storage.

Heat storage system and intermediate fluids provide a high flexibility of the process (Sadr-Kazemi and Polley, 1996). They proposed an iterative searching method based on the composite curves and considered heat quantities (e.g. kWh) to define the temperature levels of the storage tanks. Moreover, they discussed the optimal layout and the number of tanks. Although the focus is on heat storage, they pointed out that good scheduling can decrease the size of storage tanks.

Also using the composite curves resulting from the TAM, Krummenacher and Favrat (2001) proposed a heuristic targeting method to identify the minimum number of heat storage units for the heat recovery. Later, Krummenacher et al. (2010) presented an approach using a genetic algorithm to design the heat exchanger network (including storage units) for batch problems.

According to Kemp and Macdonald (1987), the TAM or pseudo-continuous approach is too simple and gives over optimistic results, except when heat storage is available, as the TAM results in the theoretical energy efficiency which needs a maximal heat storage to be achieved. In order to identify energy storage for batch problems the authors proposed the new ”time cascade” by analogy to the the graphical heat cascade.

6.2.2 Mathematical programming

First, Grossmann and Santibanez (1980) presented a MILP formulation for a given number of periods with different prices and demand fluctuations. The problem not only optimizes the total annual costs but also gives the optimal scheduling solution.

Using a heuristic approach, Vaselenak et al. (1986) considered heat integration in batch processes by optimizing the heat integration between hot and cold storage tanks. For a restricted number of temperature levels, they combined the heuristic approach with MILP. Unfortunately, the problem can only be solved for a limited number of storage tanks. Moreover, they assumed that heat is available at any time and the time scheduling of operation tasks is not taken into account.

Using the heat cascade, Maréchal and Kalitventzeff (2003) developed a methodology based on the time slice approach for multi-period problems. The periods are only linked with the common utility streams at variable usage level, where the maximum usage level defines the related investment costs. There is no possibility to integrate heat storage between periods.

To integrate indirect energy storage of batch problems in the heat exchanger network, Chen and Ciou (2008) used a MINLP formulation. The problem can become soon very complex, since the storage tanks are included in the heat exchanger network superstructure.

6.2.3 Total site approach

Using the TSM and total site approach, Varbanov and Klemeš (2011) integrated heat storage to manage the heat supply from renewables. A systematic approach allows heat storage between process operations but the self-sufficient pockets are not considered, which may miss opportunities for combined heat and power integration.

6.2.4 Scheduling

In the literature, batch problems are often solved by optimizing only the scheduling without taking into account the possibility of heat recovery through heat storage (e.g. (Adonyi et al., 2003), (Halim and Srinivasan, 2009)). Scheduling is an important issue as appropriate scheduling of tasks can reduce the size of storage tanks or even enable to avoid them. However, since the process operations are linked, rescheduling is not always feasible and it is almost impossible for complex problems to reschedule the tasks in a way that heat recovery can always be realized by direct heat exchange.

6.2.5 Synthesis of the state of the art

Heat storage and intermediate heat transfer networks enables the heat recovery in batch and discontinuous processes. Moreover, the heat transfer networks (indirect heat transfer with or without storage) will guarantee the independence of process units (e.g. disruption of one unit) and increase the flexibility and the operability of the processes. It becomes possible to practically realize large scale process integration problems.

As shown from the literature review, heat storage is very important to improve the energy efficiency of a non-continuous process. The time slice model seems to be more realistic for integrating multi-period and storage problems, since instant heat loads can be used. Therefore, the following development will focus on modifying the heat integration methods in order to take into account multi-period problems with heat storage. Scheduling problems are not considered, however it is possible to adapt the proposed method to solve simultaneously storage and scheduling problems.

There is a need to develop a methodology which is able to solve multi-scenarios (without storage) and multi- time slices (with storage between time slices). Therefore a multi-period multi-time slice approach is chosen.

6.3 Methodology for multi-period multi-time slice problems

6.3.1 Definitions

The methodology is based on two important definitions that are summarized below:

- Period: Each period refers to an independent scenario and there is no heat storage possibility between periods.
- Time slice: A time slice describes the process for a given duration, where the operations are simultaneously. When all time slices are considered together, storage units can be integrated to improve the heat recovery between time slices.

The method proposed here combines the multi-period and the multi-time slice approach. Thus, it becomes possible to solve simultaneously several periods and times slices.

For this, each independent period p consists in one or several different time slices. The concept is also shown in Figure 6.1. A period is characterized by its operating duration d_p and the number of repeated cycles cy_p over a given time interval.

The time slices can be defined for each period. As already mentioned above, heat storage between these time slices will be possible. Each time slice t is characterized with its operating time $d_{p,t}$, where p is related to the corresponding period of time slice t.

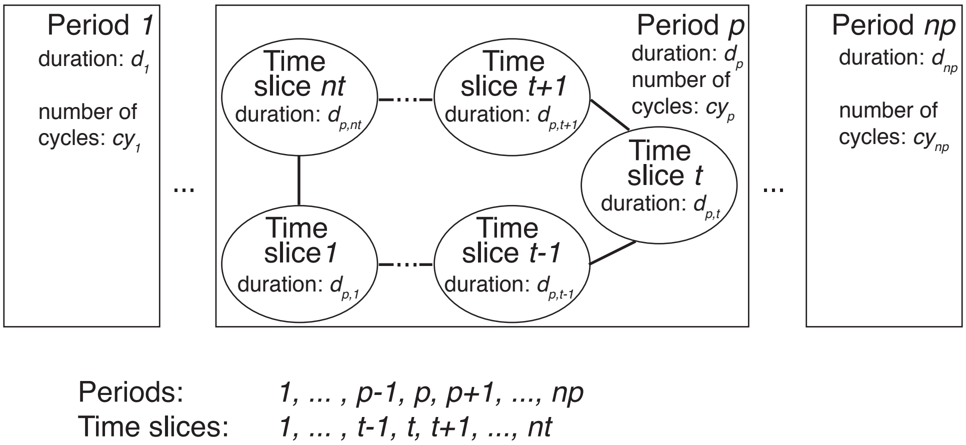


Figure 6.1: Definition of multi-period and multi-time

In the multi-time slice approach, the time slices are considered as being cyclic, which means that the state at the beginning of the first time slice is identical to the one at the end of the last time slice. Therefore, it is assumed that the amount of heat storage at the last time slice of a given period corresponds to the initial state of heat storage at the first time slice of the same period.

Each period duration d_p corresponds to the sum of all time slice durations $d_{p,t}$ (Equation (6.1)). And the total duration of a process is defined in Equation (6.2) as a function of the period durations and their number of cycles.

$$d_p = \sum_{t=1}^{nt} d_{p,t} \tag{6.1}$$

$$d = \sum_{p=1}^{np} cy_p \cdot d_p \tag{6.2}$$

With this approach, non-continuous processes can easily be modeled, by dividing them for example into a certain number of typical days and each day is divided into necessary time slices depending on the process schedule. The approach can also be useful for other application like the optimization of large industrial productions sites or urban systems.

The work focuses on the integration of storage tanks (e.g. defining the size of storage tanks and the corresponding operating conditions in each of the time slices) and the corresponding multi-time slice approach.

The multi-time slice problem is formulated as a MILP problem that will be solved independently for each of the periods. The global synthesis is done after having solved all MILP problems. As demonstrated in Chapter 7, the multi-period problem can then be solved using an automatic iterative procedure.

6.3.2 MILP formulation for multi-time slice problems

A MILP problem that targets simultaneously the heat recovery and the integration of energy conversion systems like heat pumps and other utilities in multi-time slice problems, is developed. In each time slice, the heat cascade constraints are considered together with the flows of the utility streams and the storage tanks. The storage equations are added and create a link between the different time slices. Utility and storage units are optimized to satisfy the process demand in all time slices simultaneously. The operating time of each time slice is considered.

The MILP formulation, including the approach with restricted matches from Chapter 5 and extended to the time dimension is given below. As before, bold letters are used to highlight the decision variables.

The objective is to minimize the operating costs (Equation (6.3)) of period p.

$$OpC_p = min(cy_p \cdot \sum_{t=1}^{nt} d_{p,t} \cdot (\sum_{f=1}^{nf}(c^+_{f,p,t} \sum_{u=1}^{nu} \boldsymbol{f_{u,p,t}} \dot{E}^+_{f,u,p,t}) + c^+_{el,p,t} \boldsymbol{\dot{E}^+_{el,p,t}} - c^-_{el,p,t} \boldsymbol{\dot{E}^-_{el,p,t}} + \sum_{u=1}^{nu} \boldsymbol{f_{u,p,t}} c_u)) \tag{6.3}$$

cy_p is the number of cycles of period p and $d_{p,t}$ is the operating time of time slice t in period p.

Equations (6.4) to (6.6) define, for each time slice, the overall electricity import ($\boldsymbol{\dot{E}^+_{el,p,t}}$) and the produced electricity ($\boldsymbol{\dot{E}^-_{el,p,t}}$) in the process. The concept is similar to the one of the continuous problem.

$$\sum_{u=1}^{nu} \boldsymbol{f_{u,p,t}} \dot{E}^+_{el,u,p,t} + \boldsymbol{\dot{E}^+_{el,p,t}} - \sum_{u=1}^{nu} \boldsymbol{f_{u,p,t}} \dot{E}^-_{el,u,p,t} \geq 0 \qquad \forall t = 1..., nt \tag{6.4}$$

$$\sum_{u=1}^{nu} \boldsymbol{f_{u,p,t}} \dot{E}^+_{el,u,p,t} + \boldsymbol{\dot{E}^+_{el,p,t}} - \boldsymbol{\dot{E}^-_{el,p,t}} - \sum_{u=1}^{nu} \boldsymbol{f_{u,p,t}} \dot{E}^-_{el,u,p,t} = 0 \qquad \forall t = 1..., nt \tag{6.5}$$

$$\boldsymbol{\dot{E}^+_{el,p,t}} \geq 0 \qquad \boldsymbol{\dot{E}^-_{el,p,t}} \geq 0 \qquad \forall t = 1..., nt \tag{6.6}$$

$c^+_{el,p,t}$ is the electricity purchase cost and $c^-_{el,p,t}$ its selling price for time slice t in period p. It becomes possible to adapt the electricity price to the market price (e.g. peek hours or night prices). By analogy, $c^+_{f,p,t}$ is the fuel price. $\dot{E}^+_{f,u,p,t}$ is the nominal energy delivered to unit u by the fuel (e.g. natural gas) and $\dot{E}_{el,u,p,t}$ is the nominal electricity demand$^{(+)}$ or excess$^{(-)}$ of unit u for time slice t

in period p. c_u is the nominal operating cost per hour of unit u (excluding the fuel and electricity costs).

The flow-rates of streams belonging to units are proportional to the multiplication factor $\boldsymbol{f_{u,p,t}}$ in the corresponding time slice. When the stream is a process stream that exists during the time slice t,$\boldsymbol{f_{u,p,t}} = 1$; for the utility streams the multiplication factor is variable. This factor is limited by a minimum and maximum value and corresponds to the usage level for a given time slice. The associated integer variable $\boldsymbol{y_{u,t}}$ defines if the utility unit u is used by the process ($\boldsymbol{y_{u,t}} = 1$) or not ($\boldsymbol{y_{u,t}} = 0$) in time slice t. Thus, globally integrated utilities can be activated in a time slice and deactivated in another time slice.

$$\boldsymbol{y_{u,p,t}} \cdot f_u^{min} \leq \boldsymbol{f_{u,p,t}} \leq \boldsymbol{y_{u,p,t}} \cdot f_u^{max} \qquad \forall t = 1..., nt \tag{6.7}$$

The concept of restricted matches between sub-systems and multi-level sub-systems can be extended to the multi-time slice calculation. For each sub-system s the heat cascade is given by Equations (6.8) to (6.10).

$$\begin{aligned} \sum_{h_{s,k}=1}^{ns_{h,s,k}} \boldsymbol{f_{u,p,t}} \dot{Q}_{h,s,k,u,p,t} - \sum_{c_{s,k}=1}^{ns_{c,s,k}} \boldsymbol{f_{u,p,t}} \dot{Q}_{c,s,k,u,p,t} + \boldsymbol{\dot{Q}^{-}_{hts(s),s,k,p,t}} - \boldsymbol{\dot{Q}^{+}_{hts(s),s,k,p,t}} \\ + \boldsymbol{\dot{R}_{s,k+1,p,t}} - \boldsymbol{\dot{R}_{s,k,p,t}} = 0 \quad \forall k = 1..., nk \quad \forall s = 1..., nsub \quad \forall t = 1..., nt \end{aligned} \tag{6.8}$$

$$\begin{gathered} \boldsymbol{\dot{R}_{s,1,p,t}} = 0 \quad \boldsymbol{\dot{R}_{s,nk+1,p,t}} = 0 \quad \boldsymbol{\dot{R}_{s,k,p,t}} \geq 0 \\ \forall k = 2..., nk \quad \forall s = 1..., nsub \quad \forall t = 1..., nt \end{gathered} \tag{6.9}$$

$$\boldsymbol{\dot{Q}^{+}_{hts(s),s,k,p,t}} \geq 0 \quad \boldsymbol{\dot{Q}^{-}_{hts(s),s,k,p,t}} \geq 0 \qquad \forall k = 1..., nk \quad \forall s = 1..., nsub \quad \forall t = 1..., nt \tag{6.10}$$

$\dot{Q}_{h,s,k,u,p,t}$ is the nominal heat load of hot stream h in sub-system s and temperature interval k, belonging to unit u and in time slice t. In this formulation, process units could have different heat loads in different time slices. The real heat load is calculated with the multiplication factor $\boldsymbol{f_{u,p,t}}$. When a sub-system has a deficit or a surplus of heat in the temperature interval k and in time slice t, the heat is supplied from the heat transfer system of the same sub-system ($\boldsymbol{\dot{Q}^{-}_{hts(s),s,k,p,t}}$) or respectively removed by the same heat transfer system ($\boldsymbol{\dot{Q}^{+}_{hts(s),s,k,p,t}}$). $\boldsymbol{\dot{R}_{s,k,p,t}}$ is the cascaded heat to the lower temperature interval k in sub-system s. The positive entering ($^+$) and sorting ($^-$) convention is related to the higher level heat transfer system as reference.

The heat cascade for the heat transfer system hts of each parent sub-system is given by Equations (6.11) to (6.12). $\boldsymbol{\dot{Q}^{-}_{hts+1,k,p,t}}$ is the heat supplied from the heat transfer system a level above and $\boldsymbol{\dot{Q}^{+}_{hts+1,k,p,t}}$ is the heat transferred from heat transfer system hts to the heat transfer system level $hts + 1$ of the higher level.

The heat transfer system at a higher level is part of a sub-system that contains several other sub-system with their corresponding heat transfer systems (e.g Figure 5.6 of Chapter 5).

$$\begin{aligned}&\sum_{h_{hts,k}=1}^{ns_{h,hts,k}} \boldsymbol{f_{u,p,t}} \dot{Q}_{h,hts,k,u,p,t} - \sum_{c_{hts,k}=1}^{ns_{c,hts,k}} \boldsymbol{f_{u,p,t}} \dot{Q}_{c,hts,k,u,p,t} + \boldsymbol{\dot{Q}^{-}_{hts+1,k,p,t}} \\ &- \boldsymbol{\dot{Q}^{+}_{hts+1,k,p,t}} - \sum_{s=1}^{nsub(hts)_k} \boldsymbol{\dot{Q}^{-}_{hts(s),s,k,p,t}} + \sum_{s=1}^{nsub(hts)_k} \boldsymbol{\dot{Q}^{+}_{hts(s),s,k,p,t}} \\ &+ \boldsymbol{\dot{R}_{hts,k+1,p,t}} - \boldsymbol{\dot{R}_{hts,k,p,t}} = 0 \quad \forall k = 1..., nk \quad \forall hts = 1..., nps \quad \forall t = 1..., nt\end{aligned} \tag{6.11}$$

$$\begin{aligned}&\boldsymbol{\dot{R}_{hts,1,p,t}} = 0 \quad \boldsymbol{\dot{R}_{hts,nk+1,p,t}} = 0 \quad \boldsymbol{\dot{R}_{hts,k,p,t}} \geq 0 \\ &\forall k = 2, ..., nk \quad \forall hts = 1..., nps \quad \forall t = 1..., nt\end{aligned} \tag{6.12}$$

This second set of Equations (6.13) and (6.14) ensures that the heat is cascaded correctly.

$$\begin{aligned}&\sum_{h_{hts,k}=1}^{ns_{h,hts,k}} \boldsymbol{f_{u,p,t}} \dot{Q}_{h,hts,k,u,p,t} + \boldsymbol{\dot{Q}^{-}_{hts+1,k,p,t}} + \boldsymbol{\dot{R}_{hts,k+1,p,t}} - \boldsymbol{\dot{R}_{hts,k,p,t}} \\ &- \sum_{s=1}^{nsub(hts)_k} \boldsymbol{\dot{Q}^{-}_{hts(s),s,k,p,t}} \geq 0 \quad \forall k = 1..., nk \quad \forall hts = 1..., nps \quad \forall t = 1..., nt\end{aligned} \tag{6.13}$$

$$\begin{aligned}&- \sum_{c_{hts,k}=1}^{ns_{c,hts,k}} \boldsymbol{f_{u,t}} \dot{Q}_{c,hts,k,u,p,t} - \boldsymbol{\dot{Q}^{+}_{hts+1,k,p,t}} + \boldsymbol{\dot{R}_{hts,k+1,p,t}} - \boldsymbol{\dot{R}_{hts,k,p,t}} \\ &+ \sum_{s=1}^{nsub(hts)_k} \boldsymbol{\dot{Q}^{+}_{hts(s),s,k,p,t}} \leq 0 \qquad \forall k = 1..., n_k \quad \forall hts = 1..., nps \quad \forall t = 1..., nt\end{aligned} \tag{6.14}$$

Basically, the above equations are only extended to include the new time dimension.

6.3.3 Integration of storage tanks

In order to be able to integrate storage units, each time slice needs a link with its previous and next time slice. This section describes the necessary equations to include storage into the MILP formulation of the heat cascade. The following assumptions are considered to establish the model:

- Only liquid (sensible heat) storage will be considered. The heat load is therefore calculated considering a volume, a temperature difference and a specific heat load. The approach can however be extended to consider latent heat storage.
- The storage system is discretized into a finite number of temperature levels that are modeled as interconnected storage tanks. Each virtual tank corresponds to the stored amount of

liquid at a specific temperature level. Figure 6.2 shows an example. The storage heat loads are modeled by considering the heat exchange needed (cold stream $c_l(t)$) to transfer a given amount of liquid from one temperature level to the next temperature level. Similarly, a hot stream ($h_l(t)$) that corresponds to the transfer of the liquid from temperature level to the next lower temperature level is defined.

- A bound for the maximum volume in the storage system (M_{tot}) and the temperature discretization of the tanks are defined by the user.

- The approach can include an estimation of heat losses. They are proportional to the temperature and to the volume stored: The heat exchange area of the storage tank is calculated based on the predefined maximum volume capacity.

- The investment costs of tanks are evaluated as a function of their maximum needed storage capacity

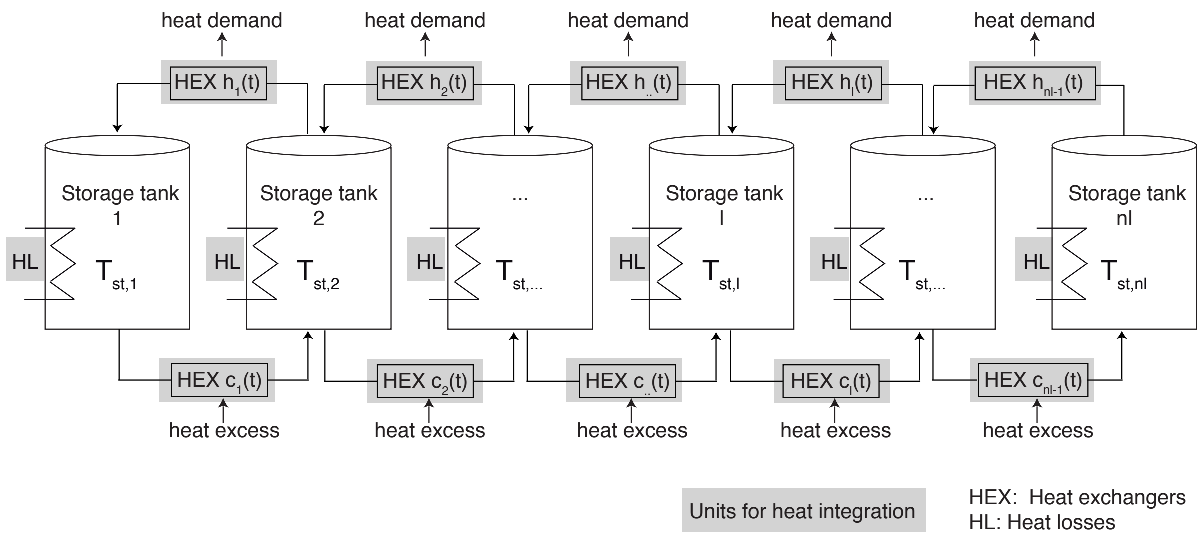


Figure 6.2: Definition of the storage model

6.3.3.1 Mass balances of the storage model

The overall mass balance is given under the cyclic constraint in Equation (6.15), which states that the total liquid volume in the storage system has to be returned to its initial state at the end of the period. The heat amount received from the process is equal to the heat amount given back to the process (no heat accumulation at the end of the period).

$$\sum_{t=1}^{nt}(cf_{hs} \cdot d_{p,t} \sum_{l=1}^{nl}(\sum_{h_l=1}^{ns_{h,l}} \boldsymbol{f_{u,p,t}} \dot{M}_{h,l,u,p,t} - \sum_{c_l=1}^{ns_{c,l}} \boldsymbol{f_{u,p,t}} \dot{M}_{c,l,u,p,t})) = 0 \tag{6.15}$$

cf_{hs} is a unit conversion factor, $d_{p,t}$ the duration of time slice t in period p and $\boldsymbol{f_{u,p,t}}$ the multiplication factor of storage unit u. $\dot{M}_{h,l,u,p,t}$ and $\dot{M}_{c,l,u,p,t}$ correspond to the nominal flow-rates of the hot and cold stream respectively in time slice t. The cyclic constraint assumes that the total initial water content is equal to the total final water content in all virtual tanks. The water content ($\boldsymbol{M_{l,t}}$) after each time slice t is calculated for each tank l in Equation (6.16). $\boldsymbol{M_{0,l}}$ is the initial water content of tank l.

$$\boldsymbol{M_{l,p,t}} = \boldsymbol{M_{0,l}} + \sum_{t=1}^{t}(cf_{hs} \cdot d_{p,t} \sum_{l=1}^{nl}(\sum_{h_l=1}^{ns_{h,l}} \boldsymbol{f_{u,p,t}} \dot{M}_{h,l,u,p,t} - \sum_{c_l=1}^{ns_{c,l}} \boldsymbol{f_{u,p,t}} \dot{M}_{c,l,u,p,t})) \quad \forall l = 1...,nl \quad \forall t = 1...,nt \tag{6.16}$$

Furthermore, Equation (6.17) imposes a positive level for each tank, and Equation (6.18) limits the total volume in the storage system. This limit is fixed for solving the MILP problem and could be optimized in an outer problem with a sensitivity analysis or a multi-objective optimization.

$$\boldsymbol{M_{l,p,t}} > 0 \qquad \boldsymbol{M_{0,l}} > 0 \qquad \forall l = 1...,nl \quad \forall t = 1...,nt \tag{6.17}$$

$$\sum_{l=1}^{nl} \boldsymbol{M_{l,t}} < M_{tot} \qquad \sum_{l=1}^{nl} \boldsymbol{M_{0,l}} < M_{tot} \qquad \forall l = 1...,nl \quad \forall t = 1...,nt \tag{6.18}$$

In order to guarantee that the storage network acts not as transfer network, Equation (6.19) becomes necessary. This is particularly important when heat exchange restrictions are defined in the problem. When the storage network can be a transfer network for indirect heat recovery, Equation (6.19) can be omitted.

$$\begin{aligned} &\boldsymbol{M_{0,l}} + \sum_{t=1}^{t}(cf_{hs} \cdot d_{p,t} \sum_{l=1}^{nl}(\sum_{h_l=1}^{ns_{h,l}} \boldsymbol{f_{u,t}} \dot{M}_{h,l,u,p,t})) \\ &- \sum_{t=1}^{t-1}(cf_{hs} \cdot d_{p,t} \sum_{l=1}^{nl}(\sum_{h_l=1}^{ns_{c,l}} \boldsymbol{f_{u,p,t}} \dot{M}_{c,l,u,p,t})) - \boldsymbol{M_{l,p,t}} \geq 0 \quad \forall l = 1...,nl \quad \forall t = 1...,nt \end{aligned} \tag{6.19}$$

6.3.3.2 Definition of thermal streams for the heat integration

The heat exchange with other process or utility units is shown on Figure 6.2. The cold storage stream $c_l(t)$ is heated up by process excess heat and is going from a lower temperature tank l at temperature $T_{st,l}$ to tank $l+1$ at temperature $T_{st,l+1}$. Whereas the hot storage stream $h_l(t)$ corresponds to water coming from a higher temperature level $l+1$ and delivering heat to the process units to reach a temperature of $T_{st,l}$. Regarding the previously presented heat cascade the hot storage stream is either accounted in the term $\sum_{h_{s,k}=1}^{ns_{h,s,k}} \boldsymbol{f_{u,p,t}} \dot{Q}_{h,s,k,u,p,t}$ of Equation (6.8) when the storage units is defined in a sub-system. Or for storage unit of the heat transfer system it is accounted in the term $\sum_{h_{hts,k}=1}^{ns_{h,hts,k}} \boldsymbol{f_{u,p,t}} \dot{Q}_{h,hts,k,u,p,t}$ of Equation (6.11). By analogy the cold storage streams are included in Equation (6.8) or Equation (6.11).

As an example, a cold stream from tank l to tank $l+1$ is defined with the inlet temperature $T_{in} = T_{st,l}$ and the outlet temperature $T_{out} = T_{st,l+1}$. Using the specific heat capacity c_p of the storage fluid, the corresponding heat amount is calculated with Equation (6.20). By analogy, the heat load is given for the hot stream in Equation (6.21).

$$\dot{Q}_{c,l} = \boldsymbol{f_{u,p,t}} \cdot \dot{M}_{c,l,u,p,t} \cdot c_p \cdot (T_{st,l+1} - T_{st,l}) \tag{6.20}$$

$$\dot{Q}_{h,l} = \boldsymbol{f_{u,p,t}} \cdot \dot{M}_{h,l,u,p,t} \cdot c_p \cdot (T_{st,l} - T_{st,l+1}) \tag{6.21}$$

6.3.3.3 Heat losses in storage tanks

Heat losses can be included in the approach. The heat losses of a storage tank at temperature $T_{st,l}$ are proportional to the amount of liquid in the storage tank and depend on the temperature level. The heat losses will be modeled by adding a cold stream to the heat cascade which corresponds to a given heat load to compensate the losses and to maintain the temperature of the tank.

The heat losses are a function of the surface area and the temperature of the tank. The heat loss model is a priori a non-linear dynamic model that has to be approximated to define a linear model in the set of equations. Therefore, Equation (6.24) has been developed to estimate the heat losses in the tank.

The heat losses depend on the isolation of the tank and would require a detailed calculation. The data from (EDF, 2011) is used, where a heat loss coefficient k_{hl} is estimated to 1/1000 kW/m^2K. The area and the volume of the tank can be calculated with Equations (6.22) and (6.23) as a function of the diameter and the height of the tank.

$$A = \pi \cdot d \cdot h \tag{6.22}$$

$$V = \pi \cdot \frac{d^2}{4} \cdot h \tag{6.23}$$

Knowing the current stored mass from Equation (6.16), the corresponding volume and the surface of the tank can be calculated. Then, by considering the heat loss coefficient k_{hl}, the heat losses can be associated in Equation (6.24). ρ is the density of the considered storage fluid in the tanks. In the following examples water is used. f_{hl} is a factor to account for the heat losses on the top and bottom of the storage tank (here 1.1).

$$\dot{Q}_{hl,l,p,t} = k_{hl} \cdot \frac{f_{hl} \cdot 4 \cdot M_{l,p,t}}{\rho \cdot d} \cdot (T_{st,l} - T_a) \qquad \forall l = 1..., nl \quad \forall t = 1..., nt \tag{6.24}$$

To take into account the heat losses, new cold streams corresponding to the heat losses and the associated temperature are added to the heat cascade. The multiplication factors of these cold streams have been fixed to consider the calculated heat losses.

It is assumed that each tank has a fixed diameter of 3m. This value is correct for storage tanks higher than 50m^3 but under estimates the heat losses of smaller tanks. However, this value can be accepted for the purpose of the targeting method presented here.

6.3.3.4 Summary of the model

The above presented multi-period multi-time slice approach is generic and can be adapted through the definition of the minimum and maximum temperature levels of the storage tanks, temperature discretization (virtual number of tanks) and the maximum volume accepted for in the storage system.

As a first approximation, the temperature ranges can be determined, by analyzing the process grand composite curve. A fine temperature discretization allows to better identify the optimal temperature levels, but increases on the other side the problem size. With a compromise of well chosen temperature discretization levels, the optimization model allows to calculate real needed temperature levels in the storage system.

The advantage of adding the heat losses are on the one hand to make the problem more realistic and on the other hand, to ensure that the storage fluid will be used as soon as possible.

In industry, often stratified tanks are used, since only one tank with the total water volume is necessary. The main advantages are that less space is needed and also the investment costs are decreased. But on the other side, the heat losses are higher due to the thermocline where hot and cold water are in direct contact (Walmsley et al., 2010). By fixing the number of tanks to 2 (2 temperature levels), the proposed approach can be applied. The bottom of the tank corresponds to the lower temperature, while the top of the tank corresponds to the higher temperature. The estimation of the heat losses and the investment costs have to be modified.

6.4 Case study

6.4.1 Problem statement

The approach is illustrated on the previous example of a cheese factory (Appendix B.2). In order to optimize a multi-period multi-time slice problem, information on the time dimension have to be added.

In the previous chapters, average heat loads were used, but to solve the multi-time slice problem instantaneous heat loads for each time slice are necessary. The process is therefore decomposed in time slices and periods. Three typical days (periods) are distinguished: week-days, Saturdays and Sundays. The detailed time slice decomposition is given in Figure 6.3 for the week-day and is described in Appendix B.2.3 in Figures B.14 and B.15 for Saturdays and Sundays.

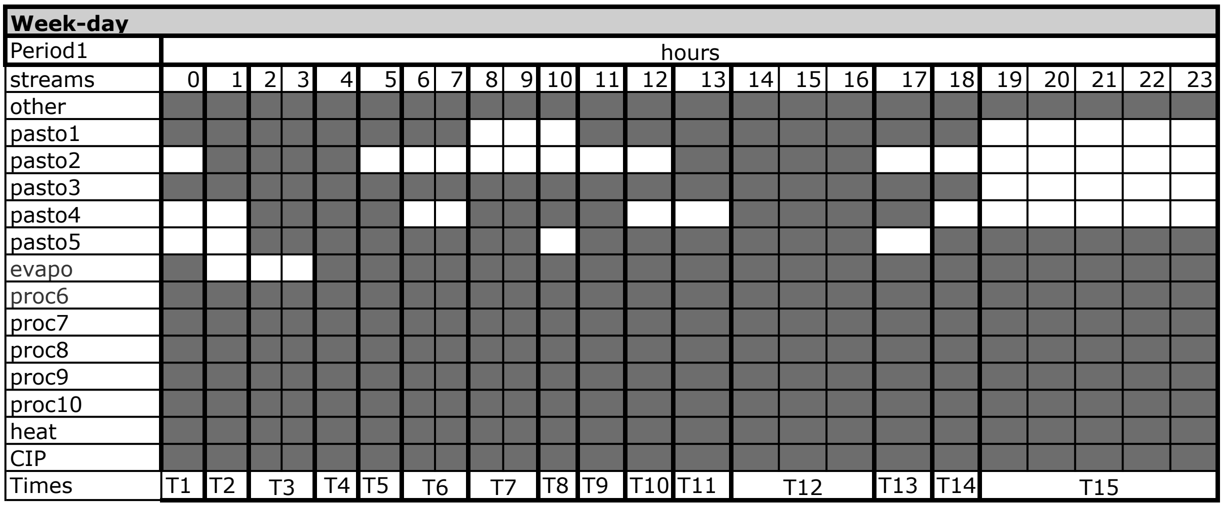


Figure 6.3: Working hours for period 1 (week-day)

As seen in Section 3.4.3.3 (pseudo multi-period approach, using TAM), the instantaneous heat loads are linked in Equation (6.25) to the average heat load q in kWh per ton of produced cheese. $\dot{Q}^*_{p,t}$ is the instantaneous heat load of period p and time slice t and M_p is the total product quantity.

$$q = \frac{\sum_{p=1}^{np} \sum_{t=1}^{nt} \dot{Q}^*_{p,t} \cdot d_{p,t}}{M_p} \tag{6.25}$$

The previous shown MILP formulation is applied independently for the three periods $p = 1, 2, 3$. The total operating costs is obtained by summing the operating costs of each period (Equation 6.26)

$$OpC = \sum_{p=1}^{np} OpC_p \tag{6.26}$$

It is assumed that 12 000 tons of cheese are produced each year. The main optimization results of the pseudo multi-period period with the integration of a conventional steam boiler, a refrigeration cycle and cooling water (Case 1a in France Table B.11 are summarized: with a total electricity consumption of 1039 MWhe and a fuel consumption of 28302 MWh, the operating cost corresponds to 1174 k€. This can be compared to the multi-period multi-time slice approach.

Case 0 The current situation which is the same than Case 0 of the pseudo multi-period approach in Table B.11 is used for comparison.

Case 1 Multi-period multi-time slice heat integration is realized with the integration of a conventional steam boiler, a refrigeration cycle and cooling water. The results can be compared with the French scenario of Case 1a (Table B.11), using the pseudo multi-period approach (detailed in Section 5.5.1).

Case 2 As Case 1 but with the integration of an additional heat pump. Its operating conditions are fixed based on the analysis of the grand composite curve (Comparable with the pseudo multi-period approach result of Case 1b in Table B.11).

Case 3 As Case 1 but additional storage system is considered (without considering the heat pump).

Case 4 As Case 2 but an additional storage system is considered simultaneously with the heat pump and the other utility units.

The storage model is discretized into 8 temperature levels between 20 °C (tank1) and 90 °C (tank8) with a temperature step of 10 °C. The initial range of temperature for the storage tank can be obtained from the analysis of the restricted matches approach.

As in the previous chapters, it is assumed that the heat pump and the process units are defined in separate sub-systems. The intermediate heat transfer units that transfer the heat between the heat pump and the process are integrated in the same way as in the pseudo multi-period approach.

A supplementary equation (Equation (6.19)) is added to the problem in order to avoid the storage units to act as heat transfer system.

6.4.2 Results overview

The global result of Case 0 to 4 are given in Table 6.1. Figure 6.4 shows the corresponding saving potential compared to the reference case (case 0). Case 5 and Case 6 will be detailed later. The results are obtained with fuel and electricity costs, CO_2 emissions and primary energy of Table 2.3 for France.

6.4.3 Results analysis and discussion

6.4.3.1 Comparison with pseudo - multi-period

For the pseudo multi-period approach, the same case study has been used in Section 5.5.1. In order to prevent the heat exchange between non-simultaneous process operation, the concerned units are defined as sub-systems. Heat transfer units for the indirect heat recovery are not considered.

Table 6.1: Multi-period multi-time slice results: Case 0 to Case 4

	Unit	Case 0	Case 1	Case 2	Case 3	Case 4
				hp	st	hp st
OpC	[k€/year]	1506	1177	1098	1169	1081
E_f	[MWh/year]	34740	28566	25164	28349	24417
E_{el}	[MWh/year]	2328	917	1792	919	1984
Q_{cw}	[GWh/year]	n.a.	12	9	11	9
M_{CO_2}	[t/year]	7232	5855	5248	5811	5115
E_p	[GJ/year]	183772	139359	134363	138400	133267

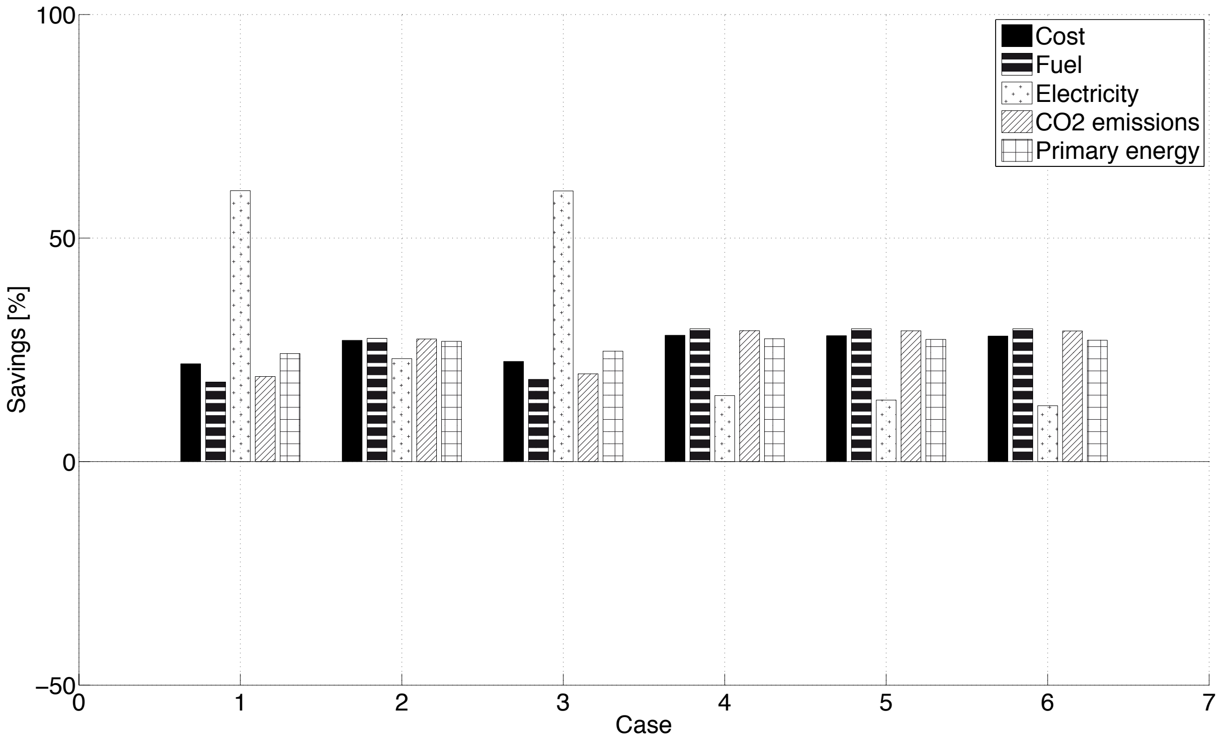


Figure 6.4: Saving potential of the cheese factory with multi-period multi-time slice resolution

Therefore, the results of Case 1 (without storage and heat pump) are comparable to Case 1a (pseudo multi-period approach) in Table B.11. The total operating cost with the pseudo multi-period approach and a cheese production of 12000 tons corresponds to 1174 k€ per year and is almost the same than the result of the multi-period multi-time slice approach.

The electricity consumption is higher in the approach of pseudo multi-period. This is due to the restricted matches. As the multi-time slice approach allows the direct heat exchange between simultaneous process operations, it is not necessary to introduce restricted matches due to non-simultaneous operations. On the contrary, the time average approach imposes restricted matches between two process operations with different operating schedules, even if some of their process operations work simultaneously for one given time slice. Thus, for this given time-slice and these two process operations, the multi-time approach allows direct heat exchange while the time average approach imposes heat exchange restrictions. Since the restricted matches in the time average approach concern the streams needing refrigeration below the pinch point, the utilization rate and consequently the electricity consumption of the refrigeration cycle are slightly higher for the pseudo multi-period calculation.

Case 2 shows that the integration of a closed cycle R245fa heat pump can save almost 7% of the operating costs, whereas the pseudo multi-period approach predicts a saving potential of 10% of the operating costs.

To conclude this section, it is important to note that both approaches are valid and give similar results. The heat pump potential is however overestimated in the pseudo multi-period approach for this case study.

The advantage of the pseudo multi-period approach is that the size of the problem is smaller and less information (e.g. scheduling) is necessary. However the multi-period multi-time slice approach can evaluate the real size of equipments and the storage potential including the operation schedule of the time slices.

6.4.3.2 Storage between time slices in a period

As shown in Table B.11, the storage potential is relatively low in the example of the cheese factory. As already mentioned in Section 5.5.1, the heat recovery potential between non-simultaneous process streams is rather low. The same conclusions are drawn when solving the multi-time slice problem.

However a small storage potential could be exploited and the optimal storage capacity can be calculated. Choosing a very high virtual amount of possible storage, the optimal heat storage amount can be optimized. This corresponds to 647 m^3. Figure 6.5 shows the optimal storage distribution of Case 3. This seems to be very high and in the following (Section 6.4.4), it will be seen that the storage volume can be reduced, by choosing a smaller upper bound for the water content.

The first important fact to notice is that only temperature levels 2 & 3 are interesting for storage. This means that heat is stored at 40 °C and after giving the stored heat to the process, the water is stored in the cold tank of 30 °C. The evaporation unit is not working in time slices 1 & 2, where the heat previously stored in tank 2 is used to satisfy a part of the hot utility at low temperature ($<$ 40 °C). It is also interesting to see that the hot tank 2 is filled only in time slices 14 and 1, just before the heat is needed, in order to avoid heat losses in the tank.

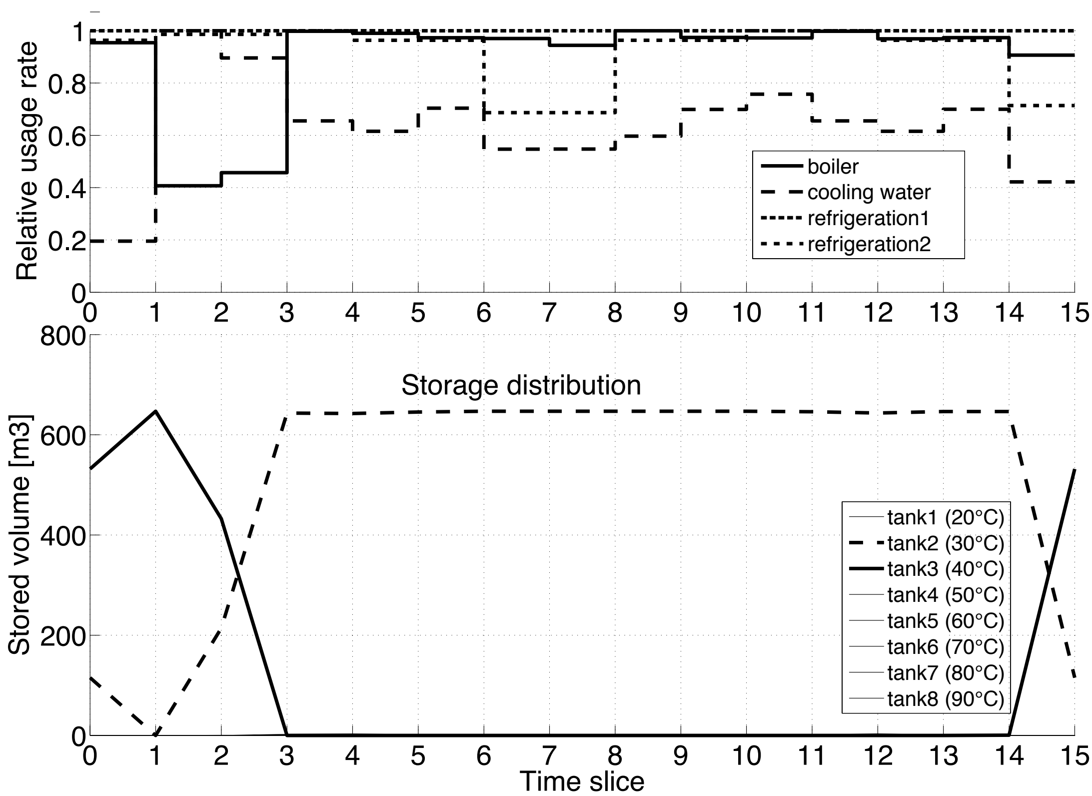


Figure 6.5: Storage distribution of Case 3 without heat pump integration

The utilization rate of utilities is also shown in Figure 6.5. The energy consumption of the boiler can be directly related to the operating hours of the evaporation unit. It is also shown that more cooling water is necessary when the evaporation unit is turned off.

6.4.3.3 Storage and heat pumps

From the energy efficiency point of view, the combination of storage and heat pump units can be particularly interesting. A heat pump can recover low grade heat to satisfy the needs of cold streams at higher temperature. The combination with storage units enables the heat recovery between hot and cold process demands that do not occur at the same time. Moreover, it increases the heat pump working hours and reduces its size, which leads to a higher profitability, especially if one considers that the use of an intermediate fluid and storage tank will be necessary to practically integrate the heat pump.

The storage distribution is shown on Figure 6.6 for Case 4 where the highest heat pump utilization rate is reached in time slice 12. At the same time storage tank 7 at 80 °C is filled. This heat is then used in time slices 2 & 3. The result of Figure 6.6 seems not be satisfying, since the heat pump works only at partial load (around 40%), except for one time slice the heat pump is fully operated.

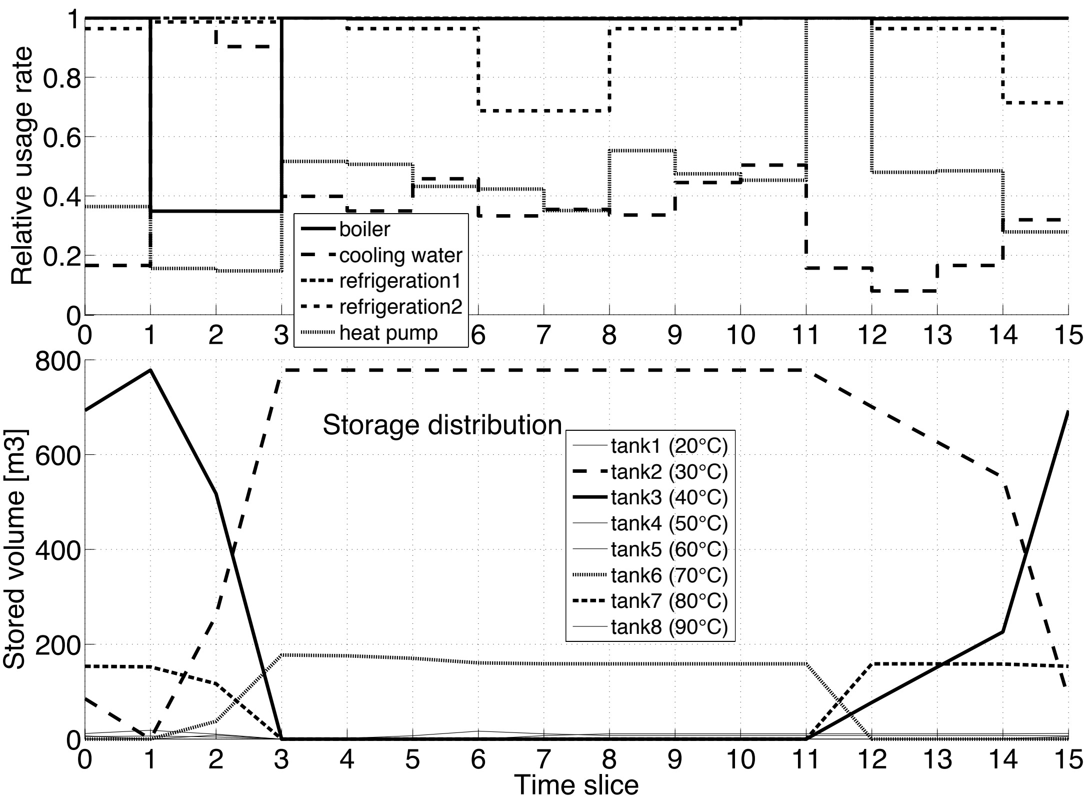


Figure 6.6: Storage distribution of Case 4, including integration of a heat pump

To improve the results the investment costs related to storage units and new heat pumps will be considered in the next section.

6.4.4 Results improvements through thermo-economic analysis

The investment for the storage tank (Bourig, 2010) can be estimated with Equation (6.28), where the volume [m3] is calculated in Equation (6.27). Equation (6.28) is added to Equations (3.26) to (3.32) of the thermo-economic evaluation presented in Section 3.4.5. Investment costs for additional required space of new storage units are not taken into account.

$$V_{l,t} = \frac{M_{l,t}}{\rho} \tag{6.27}$$

$$InvC_{st} = 2297 \cdot max(V_{l,t})^{(0.6918)} \qquad [Euro] \tag{6.28}$$

6.4.4.1 Storage tank volume

Since the investment costs should be as low as possible, a sensitivity analysis on the storage volume has been performed. The above presented Case 4 will be compared to the following cases with a limit on the storage volume.

Case 5 Storage with a total limit of 500m^3 and heat pump are considered simultaneously.

Case 6 Storage with a total limit of 100m^3 and heat pump are considered simultaneously.

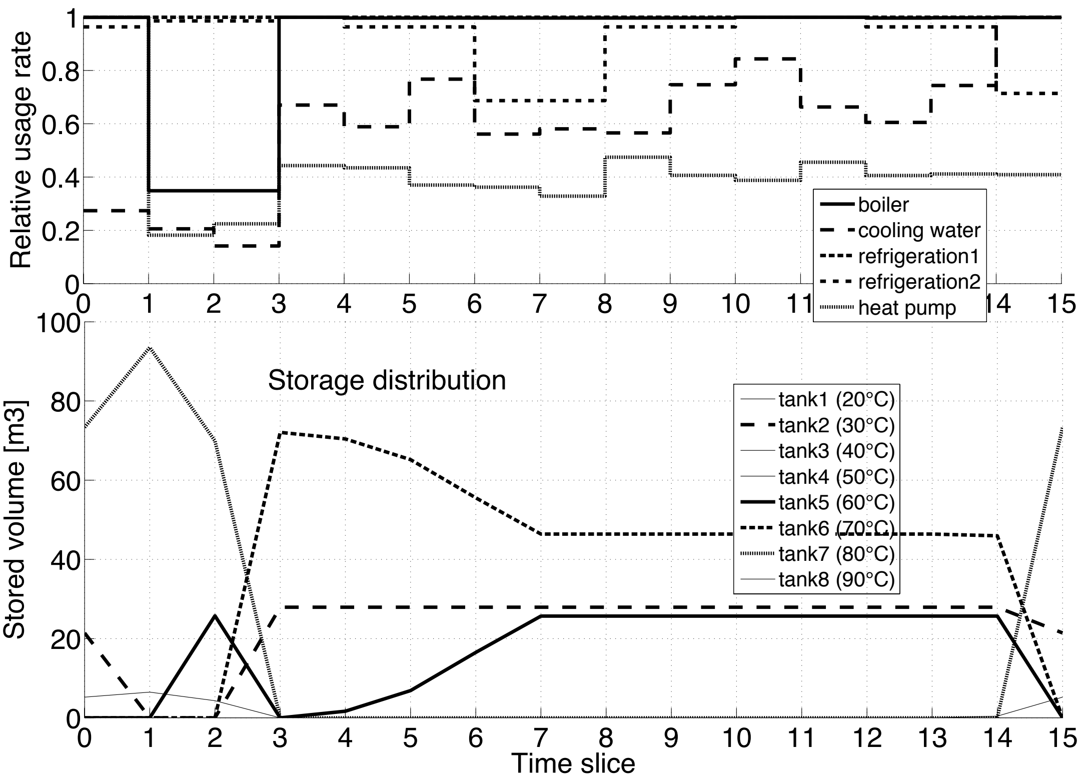


Figure 6.7: Storage distribution of Case 6, including integration of a heat pump

Since less storage volume is available, the heat has to be stored at higher temperature. Therefore when the total storage limit is 100 m^3, tanks 2, 5, 6 & 7 are used whereas the other temperature levels will not be used.

The composite curves can also be drawn in each time slice of every period. (e.g. Figures 6.8 and 6.9 for period 1 and time slice 12)

6.4.4.2 Total annual costs

In order to limit the storage capacity, it is important to take into account the related investment costs. Instead of optimizing only the operating costs, the total annual cost including the annual

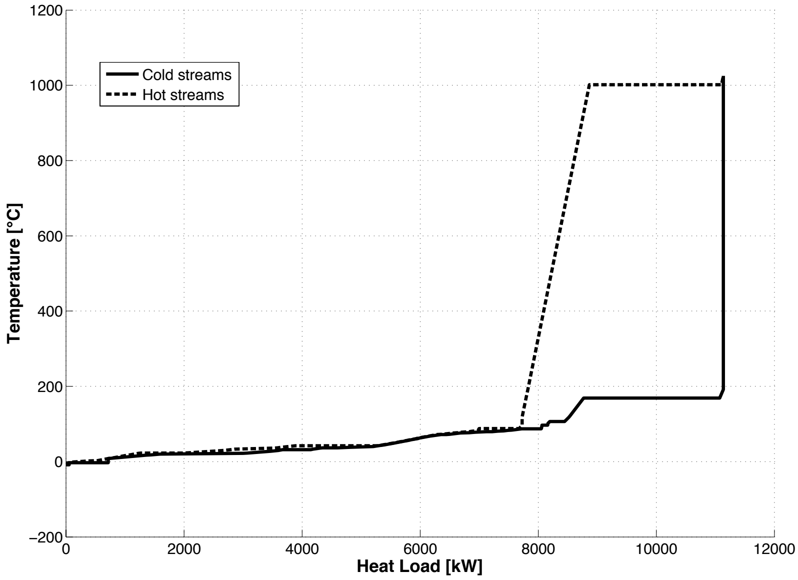


Figure 6.8: Hot and cold composite curves of period 1 and time slice 12

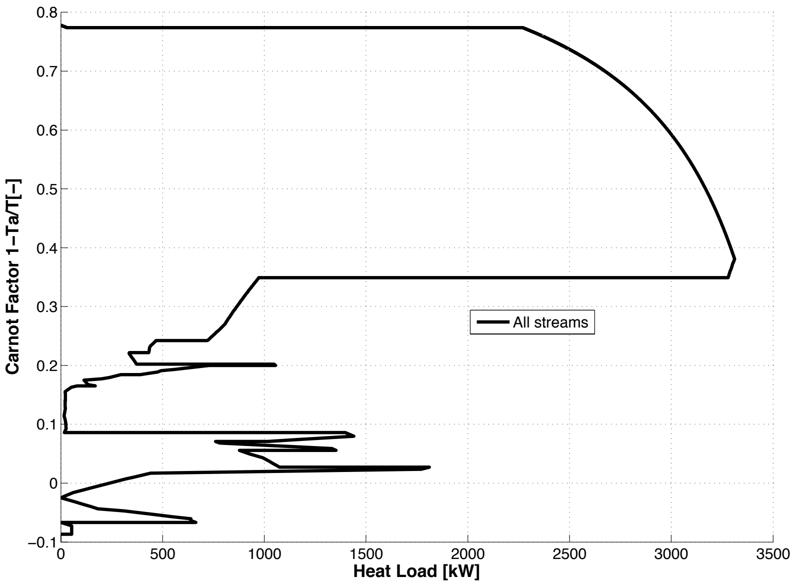


Figure 6.9: Carnot grand composite curve of period 1 and time slice 12

investment costs are optimized. The objective function for each period p is given in Equation (6.29).

$$\begin{aligned} OpC_p = min(\frac{i(1+i)^n}{(1+i)^n-1} \cdot (InvC_{hp} + InvC_{st}) + \\ cy_p \cdot \sum_{t=1}^{nt} d_{p,t} \cdot (\sum_{f=1}^{nf} (c^+_{f,p,t} \sum_{u=1}^{nu} \boldsymbol{f_{u,p,t}} \dot{E}^+_{f,u,p,t}) + c^+_{el,p,t} \boldsymbol{\dot{E}^+_{el,p,t}} - c^-_{el,p,t} \boldsymbol{\dot{E}^-_{el,p,t}} + \sum_{u=1}^{nu} \boldsymbol{f_{u,p,t}} c_u)) \end{aligned} \tag{6.29}$$

Figure 6.10 shows the storage distribution and the corresponding relative utility usage rate for Case

6, when the total annual costs are minimized.

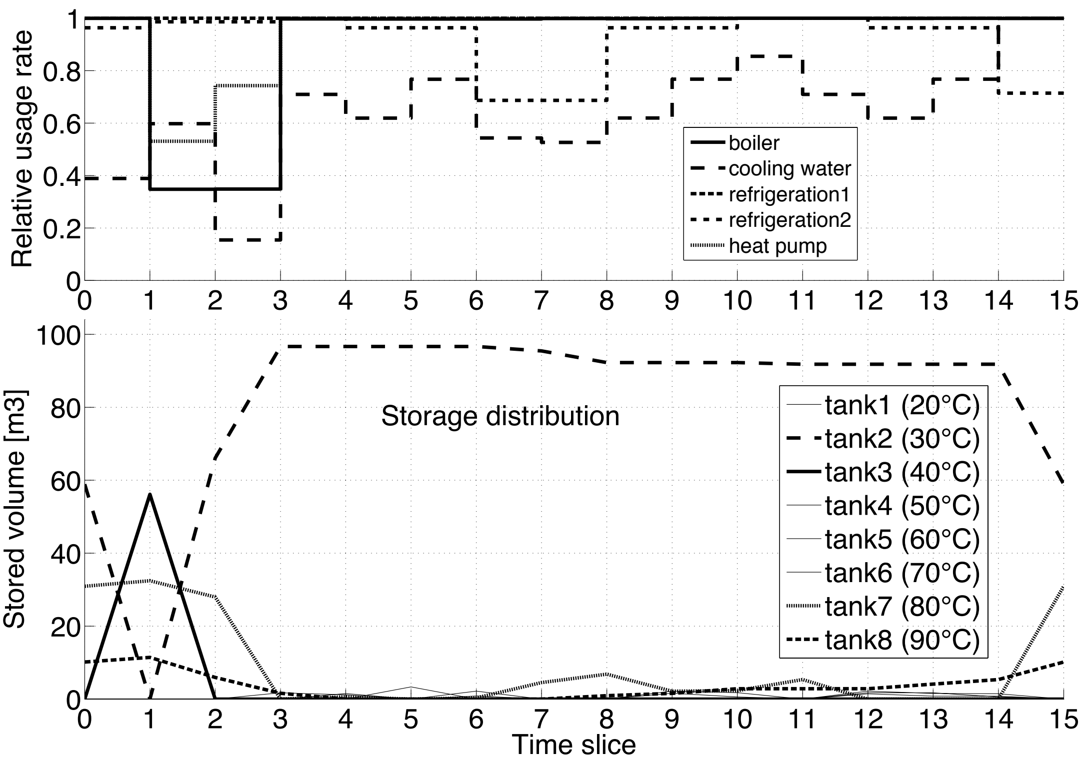


Figure 6.10: Storage tank distribution for Case 6 while minimizing total annual costs

The optimization of the total costs, results in nearly the same operating costs. The main advantage is that the investment is considered which leads to smaller installations and smaller investment costs.

6.4.4.3 Results and discussion

Table 6.2 shows the investment costs related to the heat pump and the storage system. The payback time is evaluated without considering the cost of new heat exchangers and compared to the current situation. It is important to note that the heat recovery potential is here mainly realized by basic heat exchangers.

The possibility to include the annual investment costs in the objective function, enables to reduce the equipment size and therefore the payback time. The heat pump is smaller but works at full capacity except for two time slices where the evaporation unit does not work.

It is also interesting to analyze the problem size. Table 6.3 compares the number of constraints and variables and the computation time for the MILP problems.

Introducing a heat pump and the corresponding restricted matches (between heat pump and process) increases the number of constraints and variables and also the computation time. The intro-

Table 6.2: Global multi-period multi-time slice results: Case 1 to Case 6

	Unit	Case1	Case 2 hp	Case 3 st	Case 4 hp st	Case 5 hp st 500	Case 6 hp st 100	Case 6 tot hp st 100
OpC	[k€/year]	1177	1098	1169	1081	1082	1084	1084
$InvC_{hp}$	[k€]	0	453	0	786	496	892	415
$InvC_{st}$	[k€]	0	0	455	703	495	214	184
PB	[year]	0	5.7	56.5	15.5	10.5	12.0	6.5
AP	[k€/year]	0	43	-28	-24	15	4	45

Table 6.3: Example problem size of period 1

MILP Problem	Constraints	Variables	Computation time
Case 1	4671	4826	0.016 sec
Case 2	12683	11586	0.187 sec
Case 3	10496	10124	0.094 sec
Case 4	19918	18054	1.560 sec

duction of a storage system (Case 3) increases also the problem size. Consequently the problem size of Case 4 which integrates both the heat pump and the storage system becomes bigger. Especially the computation time is more than 8 times higher than in Case 2. Cases 5 and 6 have the same number of constraints and variables than Case 4. However, the computation time for data handling is not considered here and can be significantly higher (e.g. 1-2 hours for Case 4).

Case 1 could be compared to Case 1a, calculated with the time average approach in Table B.11. The time average problem is solved in 0.062 seconds with 2187 constraints and 2005 variables. The multi-period multi-time slice approach needs one optimization for each period, the example of one period with 15 time slices is given in Table 6.3 (Case 1).

6.5 Conclusions

This chapter introduced the multi-period multi-time slice approach that is illustrated by a case study from the food industry. This approach can integrate non-continuous process operation and seasonal variations.

Results are presented on a multi-period real case study. The method seems to be promising to solve multi-period problems and the small saving potential can be explained by several facts. First the process of the case study in the current situation is already quite efficient. Nevertheless, the method showed that the saving potential is high especially through optimal heat recovery using simple heat exchangers.

Concerning heat pump integration, the temperature difference is big, since the technology representation has been chosen. It would be worth to study more in detail the main energy consumer, the evaporation unit. This has been done in Section 3.5.2.2, where the evaporation unit has been modified and a mechanical vapour re-compression unit replaces the thermal vapour re-compression. Rather than taking the evaporation unit as a black box, saving potential can be exploited by modifying the evaporation unit to make it more efficient.

This complete approach is new and before going further, several problems have to be fixed: the number of constraints and variables and thus the problem size is increasing. The resolution time of the MILP is acceptable, but the required data handling to define the input problem takes significantly more time than for the pseudo multi-period approach. This depends of course on the number of time slices.

Finally, there is a need of more detailed results analysis and also the heat load distribution and the heat exchanger network has to be adapted, to be able to solve the complete problem. Especially for the process flexibility which is strongly linked to the heat exchange connections this is in an important topic. The intermediate heat networks of the previous chapter are therefore very useful to make the process operation independent from each other, since the heat exchange could be realized through the networks. Keeping in mind that the heat exchanger network is still too complex for mono-period problems, it will not be easy to compute the heat exchanger network for large scale multi-period multi-time slice systems.

The multi-period multi-time slice approach is a good way to evaluate the efficiency of industrial processes with batch or discontinuous operations where storage can play a role. It has been demonstrated that the pseudo multi-period approach presented in Chapter 3 gives similar values for the target, but does not provide information on the size of the storage tanks and on their operating conditions. However, since the obtained results are similar, both approaches are valid. The main advantage of the multi-time slice approach presented here is that the utility equipments sizing and storage units can be estimated from the beginning. The investment calculation is therefore more realistic.

It has been shown that the investment costs play an important role and have to be considered. There are two possibilities. Either investment costs can be considered as a part of the objective function in the MILP formulation (e.g. Section 6.4.4.2) or the second possibility is to use the multi-objective optimization. The multi-objective optimization could thus characterize the trade-off between process operating and investment costs of heat recovery, storage and heat pumps and identify optimal solutions. It could be for example interesting to have continuous heat pump usage to increase their profitability.

Another open point is the optimization of the temperature levels of the storage tanks. One option could be to define a fine discretization of the temperature levels in the virtual storage system. The second option could be a non-linear optimization approach, based on a multi-objective optimization strategy as it has been developed for heat pump integration in Chapter 4. In the same way the temperature level of the storage tanks could be defined as the optimization variables.

Chapter 7

Implementation

This chapter summarizes the implementation approach and gives a small overview on the tool that is developed and validated in this thesis.
Related publication: Approach: Bolliger et al. (2009).

7.1 Introduction

The design of energy conversion systems is based on models which describe the mass and energy balances for the different process units and their integration into the global system. These models generate the data needed to analyze the overall system efficiency and to establish performance indicators, using for example exergy analysis, process integration with pinch analysis or thermo-economic evaluation. The increasing complexity of the system, the highest degree of integration and the increasing number of energy conversion options together with the demand of applying different performance indicators require more systematic approaches. This chapter proposes a tool which systematically tackles the integrated system design by dissociating technology modeling from the methods for the analysis and the synthesis of integrated systems.

In recent years, research activity in energy conversion system analysis and design considers more and more complex systems, often composed by combining smaller sub-systems. The domain covers multiple system scales from equipment design (Palazzi et al., 2007) to process design (e.g. biomass conversion processes (Gassner and Maréchal, 2009)) to industrial processes (e.g. industrial processes (Becker et al., 2011d)) and even urban systems (Maréchal et al., 2008).

In order to address the problem of handling complex models, research recently focused on developing tools for exchanging information and allow the interoperability of modeling softwares. For example, The DOME platform (distributed object-based modeling environment) Kraines and Wallace (2003) implemented a model based co-current system design and engineering platform. It has been

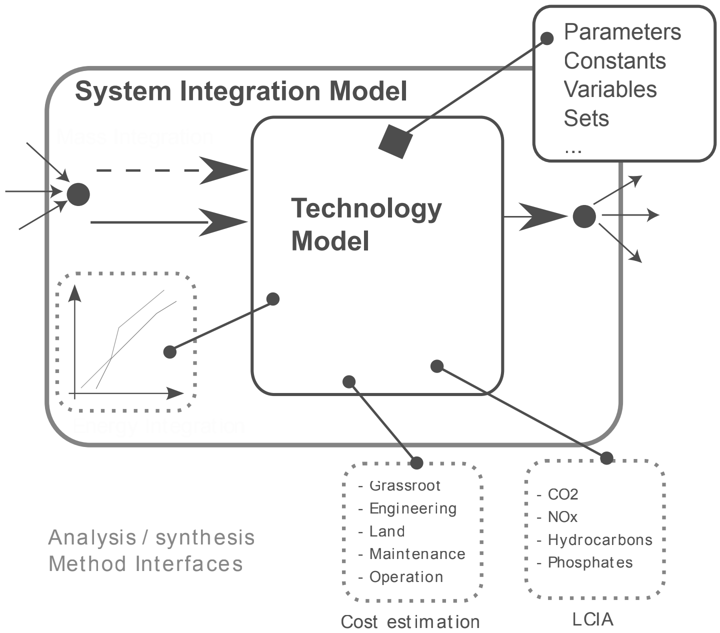


Figure 7.1: Schematic view of the separation of physical model from method analysis related data (Bolliger et al., 2009)

successfully applied to urban systems and allows to interconnect versatile sub-systems models using web services. The CAPE-OPEN Morales-Rodríguez et al. (2008) initiative on the other hand has been developed by the process engineering community to allow the interoperability of flow-sheeting tools, unit models and thermodynamic packages. Although these methods give the opportunity to construct very complex models, most of the time they are only focused on the process flow-sheet calculation problem.

However, the design of the energy conversion systems requires the application of one or more analysis and synthesis methods in order to deduce the performance indicators and the information about the interactions between the various technologies of the global system. The major drawback of the existing approaches is that they do not separate energy and mass balances modeling from the system analysis and the synthesis methods. Models are built in one single block containing all the information. The reuse of the same models in different study contexts becomes therefore very difficult and often requires a partial recoding of the model.

7.2 Conceptual approach

A new approach is proposed to handle and connect models by separating the chemical and physical models of the process units from the methods used for system-scale analysis and synthesis. The method is based on a generic syntax which describes the different sub-system models. A schematic representation of the concept is shown in Figure 7.1.

The syntax provides a generic abstraction layer that describes the models independently from their modeling environment and considers their reuse in different projects. This also gives the possibility to develop a data base of technology models that may be used to develop process superstructures.

A model representing a technology is the combination of a set of modeling parameters used to represent the characteristics of the technology and its environment and of a set of equations that describes the thermo-physical and chemical conversion operations of the technology. Models can be connected via their input and output streams and or by shared model parameters.

Regarding the applied system analysis methods, dedicated interfaces store the required information in set of structured variables. The interfaces include the extraction and the generation of the required data for the analysis methods from the model results. As example, for the process integration method, the interface includes the definition of the hot and cold streams of the technology as well as the related energy and material flows (electricity, water, resources,...).

The interest of the proposed approach is the separation of tasks: the development of a new method does not require the modification of the physical model and does not affect the information handled by the other methods. However, it may require specific development when particular data are needed.

The syntax also defines other data, in relation with the insertion of the model into a shared data base. These information concern the model classification, its documentation, the history of modifications and more generally data allowing to establish the model accuracy and quality. The defined syntax is generic: it can be implemented in any programming language accepting structured variables; the content of the structures is suitable to describe any energy conversion technology and more generally any model which can be built with a black-box technique. More details can be found in the thesis of Bolliger (2010b).

7.3 Tool

Figure 7.2 gives an overview on the implementation. As mentioned before the models and the analysis methods are separated. The platform "Osmose" currently programmed with Matlab links the following parts:

- Frontend: This is the very first function to be called. From there, all important parameters can be modified and the problem is started.
- Methods: Depending on the problem statement and the inputs in the frontend, the necessary methods are considered and the problem is solved.

- Technology models: Each model is described using the Energy Technology syntax (Bolliger, 2010a). The model can apply external functions to generate necessary information.
- Data: The data file gives the possibility to separate confidential data from the model files. Also for structuring, it becomes easier to collect all the necessary data in a separate file.

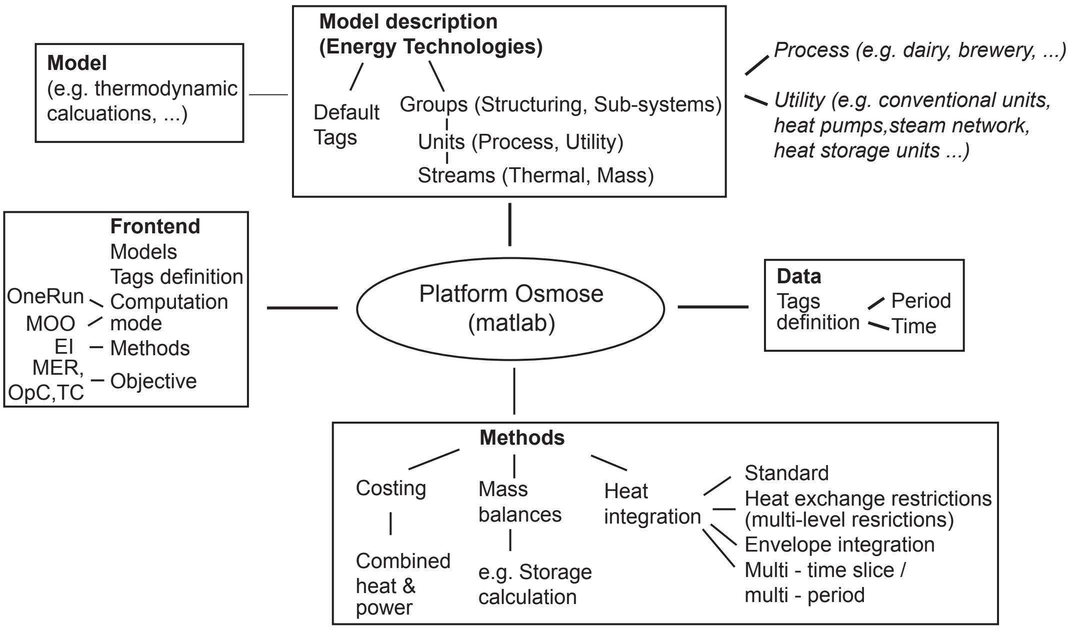


Figure 7.2: Overview of the platform Osmose

When the problem is solved post computations (e.g. thermo-economic analysis) can be done and a part which treats the reporting and storing the output files is also available (Matlab).

7.3.1 Frontend

The frontend defines the problem: models, tags, computation mode and the objective functions.

In this thesis the main goal is to solve the energy integration problem by minimizing the operating costs (OpC) or the total costs (TC) which include the annualized investment costs.

Figure 7.2 shows that two types of computation modes can be defined in the frontend. The "OneRun" corresponds to a single run. This computation mode is used to solve the heat integration (for example the dairy process or the cheese factory together with the utility models in Chapter 3). Concerning the heat pump integration, the operating conditions (temperature levels) are fixed before solving the heat cascade by using a mixed integer linear programming formulation. The second type ("MOO") is used when the problem becomes non-linear. As shown in Chapter 4, the problem can be decomposed in an outer non-linear and an inner linear problem. The outer problem is solved by multi-objective optimization based on an evolutionary algorithm and the inner problem is solved as before by mixed integer linear programming. Therefore, instead of fixing the heat pump operating conditions, they can be optimized in the outer problem. Objective functions for the outer problem have to be defined. The evolutionary algorithm in the used tool is programmed in Matlab.

7.3.2 Methods

To solve the energy integration problem, several methods are available (e.g. standard, with heat exchange restrictions, ...). For the energy integration part one method is chosen, however it is also possible and frequent to combine it with other methods like mass balance or costing calculations.

The methods can be solved by mixed integer programming, using ampl (solver: cplex) or the open source software glpk (solver glpsol).

7.3.3 Technology models

The models are described with a generic syntax and necessary information is generated by solving the model. A technology model can be a unit (e.g. steam network), a complete process (e.g. dairy) or a collection of heat pump models (e.g. heat pump data base). Regarding the thermodynamical model calculation, different external software tools may be necessary. The platform is able to handle the necessary input data, to solve the specific problem and to recover the results. As example, the heat pump cycles are calculated using the flow-sheeting software Belsim ValiGui.

The used models are summarized in Table 7.1.

Table 7.1: Energy technologies models

Model Name	Type	Software	Method
Dairy	Process	Matlab	Heat integration
Brewery	Process	Matlab	Heat integration
Cheese factory	Process	Matlab	Heat integration
Pulp & paper (simplified)	Process	Matlab	Heat integration
Steam boiler	Utility	Matlab	Heat integration & Costing
Co-generation engine	Utility	Matlab	Heat integration & Costing
Cooling water	Utility	Matlab	Heat integration & Costing
Steam network	Utility	Matlab	Heat integration & Costing
Heat recovery loop	Utility	Matlab	Heat integration & Costing
Heat storage	Utility	Matlab	Heat integration & Costing & Mass balance
Heat pump data base	Utility	Matlab/ Belsim ValiGui	Heat integration & Costing

Chapter 8

Conclusions and perspectives

8.1 Conclusions

Reducing the energy consumption and increasing the energy efficiency in an industrial process become more and more important with raising energy prices. In this context, the potential of industrial heat pumps and their integration into industrial processes have been studied.

First, a short methodology to rapidly evaluate a heat pump potential has been presented. For this, it is assumed that the heat pump is correctly integrated and that its optimal operating conditions are known.

One of the key challenges in this thesis was to define heat pump integration possibilities, based on pinch analysis and process integration. By analyzing the grand composite curve, heat pump potentials and their operating conditions can be identified. The flow-rates are then optimized, by calculating the heat cascade with a MILP formulation. In particular, it has been shown that the simultaneous integration of heat pumps with other energy conversion units identifies more heat pump opportunities than conventional pinch analysis. The dairy process can be mentioned, as example: the heat pump potential is increased due to its optimal combination with the refrigeration cycle. Relative operating cost savings up to 56% can be reached.

At this stage, the operating conditions of heat pumps and other utility units have been chosen manually with the help of the grand composite curve. Different options can be tested and evaluated, by applying thermo-economic analysis. In addition to the operating cost savings, the CO_2 emissions and the primary energy savings are analyzed. It shows the important role of the electricity mix. To define the profitability of new equipments, the investment costs and the related pay-back times are estimated.

The second objective was to create an automatic optimization approach to identify not only heat pump opportunities, but also their optimal operating conditions. In order to systematically evaluate heat pump integration, the methodology has been implemented in a multi-objective optimization approach. The two objectives are the operating and the investment costs. The approach is decomposed in a non-linear outer (master) and a linear inner (slave) problem. The temperature levels of the heat pumps are optimized in the master problem, whereas the selection and the optimal flow-rates are define in the slave problem. This one solves the heat cascade using the MILP formulation. Finally, optimal heat pump solutions lay on the resulting Pareto front. The main advantage is that the final solution can be chosen among several potential solutions, by applying economic, environmental or other criteria. Furthermore, this approach enables to integrate technological limits, like ranges of volumetric flow-rates or pressure ratios of a given compressor, in the targeting stage. Thus, the solutions are not only optimal but also technologically feasible.

Taking into account practical implementation issues of an industrial process, several limits of the methodology have been discovered. Although process integration methods are promising, the basic methodology had to be adapted in order to get more realistic solutions.

The first adaption concerns heat exchange restrictions. Many different industrial constraints (e.g. safety reasons, product quality, large distances, ...) can prevent a direct heat exchange. The basic methodology assumes that for the heat recovery any heat exchange between hot and cold streams is possible. To have more realistic solutions, heat exchange constraints should be included in the problem statement. For this, the heat cascade equations have been modified and a new sub-system concept has been developed. The industrial processes can be organized in several sub-systems and heat can only be exchanged indirectly between sub-systems through heat transfer units. For heat pumps, heat exchange restrictions can become important, for example if heat cannot directly be transferred to the process due to safety reasons.

The penalty due to heat exchange restrictions is reduced through the integration of intermediate heat transfer units. In order to identify optimal ones, the new envelope composite curves, also optimized with a MILP formulation, have been developed.

The extension to multi-level heat restrictions has also been realized. Thus, it is possible to define sub-systems in sub-systems. The multi-level sub-system approach is particularly interesting when different levels of heat exchange restrictions are necessary. A typical example is a large scale industrial process with different locations interconnected by heat transfer networks. Direct heat exchange restrictions between the locations are not possible due to a long distance and moreover, heat exchange restrictions can be imposed in a second level between process units of the same location because of safety reasons.

Heat exchange restrictions can also be useful for multi-period problems. Process operations which are not working simultaneously are defined in separate sub-systems, since direct heat exchange is not possible. Indirect heat transfer units (except storage units) can be integrated. The multi-level sub-system approach gives even the possibility to integrate at the same time other heat exchange restrictions, as for example product quality reasons in a second level of sub-systems.

As industrial process are often operated in batch mode, and the storage units become important to realize heat recovery between non-simultaneous streams, a second adaption has been proposed. It includes heat exchange restrictions between sub-systems. Two different dimensions of non-simultaneous operations can be distinguished. First, for a multi-period problem, it is considered that there is no link, and therefore no storage between periods (e.g. independent production scenarios). The second dimension is related to the multi-time slice problem, where storage units are the link between different time slices. This approach enables to better evaluate the size of new utilities, and especially the size of heat pumps and storage units. The heat transfer between different time slices is possible through the integration of storage units. Their size and temperature levels are selected. This enables the complete integration of a process with several periods (e.g. typical days without storage from one day to another) and several time slices (e.g. hour of a typical day and storage possibility during the day).

The developed methodologies have been tested and validated with several case studies. Furthermore, heat pump integration has been demonstrated with the help of application cases. As example, the analysis of a brewery process showed relative operating cost savings up to 40%, and up to 50% for a cheese factory.

Finally, the implementation approach has been shortly discussed. A generic syntax, included in a common platform, enables to have reusable models and facilitates the interaction between different applied methods, which are often linked to specific software tools.

8.2 Perspectives

The heat pump potential has been shown on several case studies. In order to guarantee the final practical implementation in an industrial process, following points should be considered.

- The results of the thermo-economic analysis strongly depends on the quality of input data. Especially the information on investment costs of heat pumps is one of the most important input parameters. It has been shown that any type of heat pump can be integrated. For example, standard heat pumps with lower investment costs could be considered. It is crucial to have good investment estimations corresponding to the current heat pump market and therefore a detailed market analysis. The implementation of these results into the data base improves the quality of the results.

- The biggest missing part concerns the heat load distribution and the heat exchanger network. At the moment it is not always possible to solve it for complex problems with a high number of streams. A methodology to split the problem into smaller problems and to solve them separately, could reduce the problem size. It would be a first step to solve the heat exchanger design problem for large scale industrial problems. The optimization of the heat load distribution could include geographical coordinates in order to estimate the the distance and the corresponding piping costs.
- The methodology could be extended to include other limits, as for example, to restrain the size of heat exchanger in order to avoid small heat exchangers and to make the heat exchanger network more practical.
- Another limiting factor concerns the programming part. Beside the creation of a user friendly interface, also the resolution speed and the robustness of the programming should be considered.

In the following, other suggestions to improve the process integration analysis are summarized.

- The heat pump data base could be extended to new heat pump models. For example, new refrigerants, or even mixtures could, be defined and modeled. Also new heat pump models (e.g. absorption heat pumps or ORC-ORC systems) could be modeled and included in the data base. The optimization and the decision variables have to be adapted.
- The environmental indicators could be extended to other environmental performance indicators than the CO_2 emissions. For example a complete life cycle analysis could be performed to evaluate the results.
- The multi-period problem could be solved by combining the possibility of storage and scheduling optimization.
- Furthermore control and regulation problems could be included in the approach.

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Appendix A

Heat pump saving potential

A.1 Operating costs, CO_2 emissions and primary energy savings

The saving potential of operating costs, CO_2 emissions and primary energy is resumed in this appendix, when the heat is supplied by a heat pump, instead of a co-generation device.

From Equation (2.30) the operating cost savings can be deduced.

$$\Delta OpC = \left(\frac{c_f}{\epsilon_{th}} - \frac{c_{el} \cdot \epsilon_{el}}{\epsilon_{th}} - \frac{c_{el}}{COP} \right) \dot{Q}_{th} \cdot d \tag{A.1}$$

The electricity to fuel price ratio (k_{cost}) is defined in Equation (A.2).

$$k_{cost} = \frac{c_{el}}{c_f} \tag{A.2}$$

Equation (A.3) gives the relative operating cost saving potential as a function of k_{cost}.

$$\Delta OpC_{rel} = \Big(1 - k_{cost}(\frac{\epsilon_{th}}{COP} + \epsilon_{el})\Big) \tag{A.3}$$

By using heat supplied from a heat pump, the CO_2 saving is expressed by Equation (A.4).

$$\Delta M_{CO_2} = \left(\frac{m_f}{\epsilon_{th}} - \frac{m_{el} \cdot \epsilon_{el}}{\epsilon_{th}} - \frac{m_{el}}{COP} \right) \dot{Q}_{th} \cdot d \tag{A.4}$$

k_{CO_2} is introduced as the ratio of CO_2 content of electricity and fuel (Equation (A.5)). The relative CO_2 emissions reduction is expressed by Equation (A.6).

$$k_{CO_2} = \frac{m_{el}}{m_f} \tag{A.5}$$

$$\Delta M_{CO_2 rel} = \Big(1 - k_{CO_2}(\frac{\epsilon_{th}}{COP} + \epsilon_{el})\Big) \tag{A.6}$$

By analogy, primary energy savings can be introduced by Equations (A.7) to (A.9).

$$\Delta E_p = \left(\frac{e_{p_f}}{\epsilon_{th}} - \frac{e_{p_{el}} \cdot \epsilon_{el}}{\epsilon_{th}} - \frac{e_{p_{el}}}{COP}\right) \dot{Q}_{th} \cdot d \tag{A.7}$$

$$k_{E_p} = \frac{e_{p_{el}}}{e_{p_f}} \tag{A.8}$$

$$\Delta E_{p_{rel}} = \left(1 - k_{E_p}(\frac{\epsilon_{th}}{COP} + \epsilon_{el})\right) \tag{A.9}$$

A.2 Graphical saving potential maps

The relative saving potential can be estimated as a function of the sink and source temperature and a theoretical Carnot efficiency (η_{COP}). Typical value for η_{COP} is 55%. The COP of a heat pump can be defined by Equation (A.10).

$$COP = \frac{\dot{Q}_{th}}{\dot{E}_{hp}} = \eta_{COP} \frac{T_{sink}}{T_{sink} - T_{source}} \tag{A.10}$$

With Equation (A.10) Equation (A.3) becomes Equation (A.11).

$$\Delta OpC_{rel} = \left(1 - k_{cost} \cdot (\frac{\epsilon_{th}}{\eta_{COP}} \cdot \frac{T_{sink} - T_{source}}{T_{sink}} + \epsilon_{el}\right) \tag{A.11}$$

Equations (A.6) and (A.9) can be transformed by analogy.

$$\Delta M_{CO_2 rel} = \left(1 - k_{CO_2} \cdot (\frac{\epsilon_{th}}{\eta_{COP}} \cdot \frac{T_{sink} - T_{source}}{T_{sink}} + \epsilon_{el}\right) \tag{A.12}$$

$$\Delta E_{p_{rel}} = \left(1 - k_{E_p} \cdot (\frac{\epsilon_{th}}{\eta_{COP}} \cdot \frac{T_{sink} - T_{source}}{T_{sink}} + \epsilon_{el}\right) \tag{A.13}$$

Table A.1 shows the energy prices (according to IEA (IEA, 2009) and Eurostat (Eurostat, 2009)), CO_2 emissions for the electricity mix and the corresponding primary energy in different countries (according to Ecoinvent (Frischknecht et al., 2005)).

Table A.1: Cost, CO_2 emission and primary energy for given electricity mix

	c_f €/kWh	c_{el} €/kWh$_{el}$	m_{el} kg/kWh$_{el}$	$e_{p_{el}}$ MJ/kWh$_{el}$
FR	0.039	0.062	0.092	11.788
DE	0.050	0.108	0.631	10.945
CH	0.036	0.069	0.113	8.094
US	0.018	0.052	0.745	12.399

The CO_2 content of natural gas is considered to be 0.202kg/kWh and the primary energy for natural gas has a value of 4.5MJ/kWh. Table A.2 defines the corresponding k factors.

Table A.2: k factors

	k_{cost}	k_{CO_2}	k_{E_p}
FR	1.58	0.46	2.62
DE	2.16	3.12	2.43
CH	1.92	0.56	1.80
US	2.89	3.69	2.76

In the following, the saving potential maps of operating costs, CO_2 emissions and primary energy are presented in Figures A.1 - A.3 for France and Germany and in Figures A.4 - A.6 for Switzerland and US.

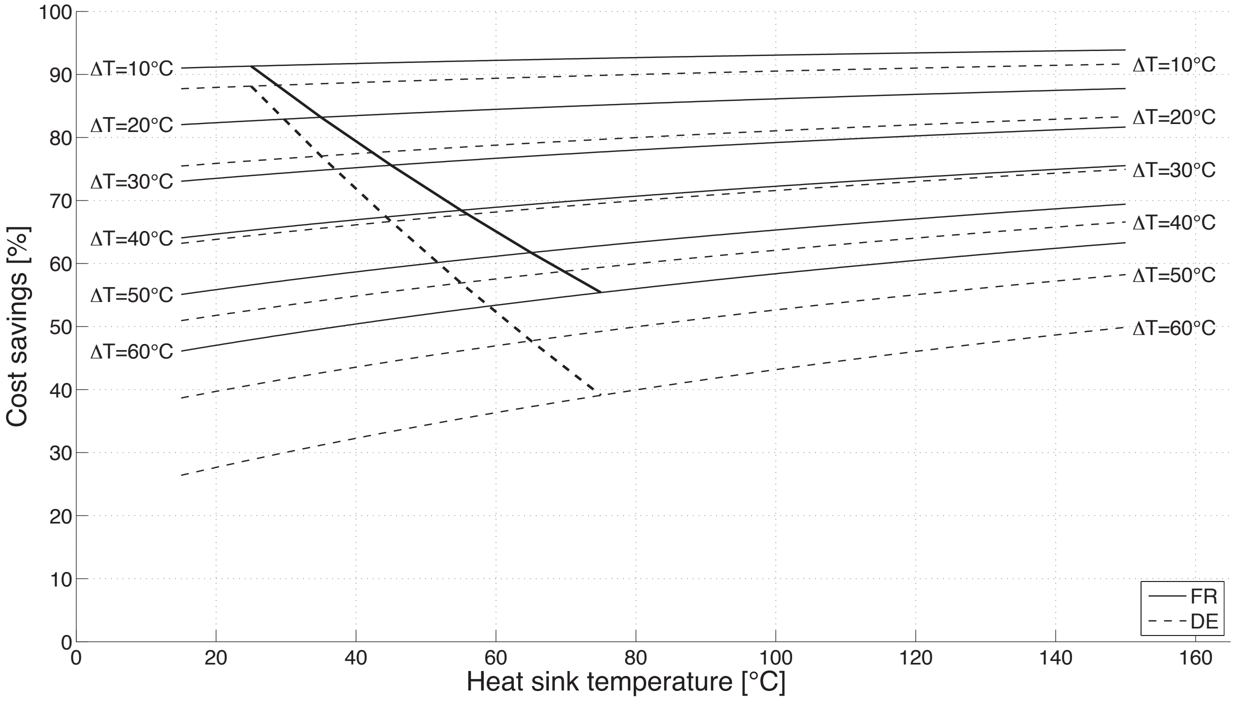


Figure A.1: Relative operating cost savings (Equation (A.11) / (A.3)) as a function of the heat sink temperature and the temperature lift (ΔT) for France and Germany

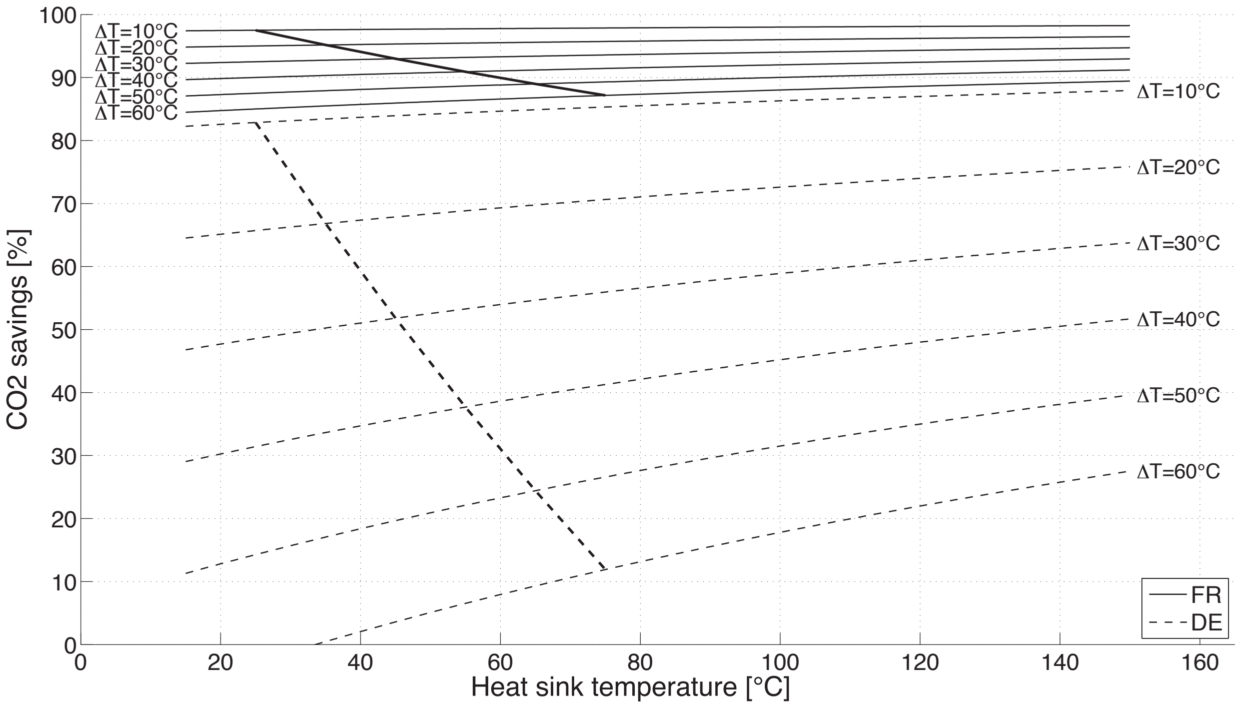


Figure A.2: Relative CO_2 emission savings (Equation (A.12)) as a function of the heat sink temperature and the temperature lift (ΔT) for France and Germany

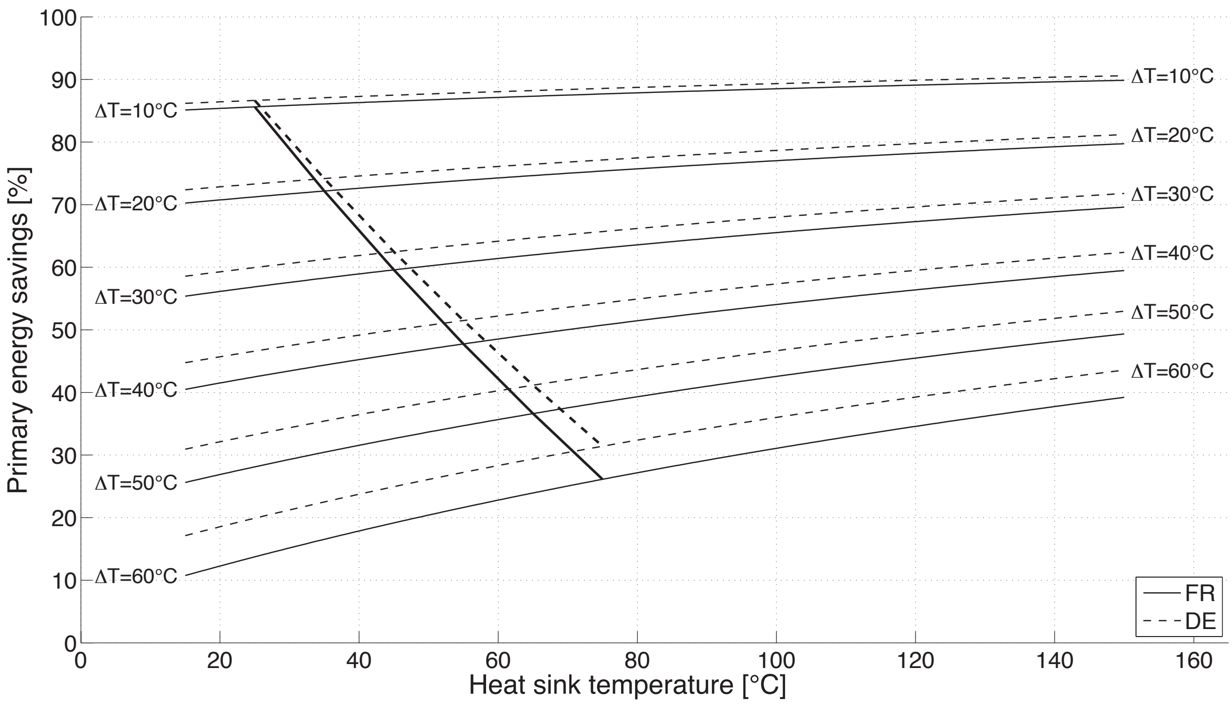


Figure A.3: Relative primary energy savings (Equation (A.13)) as a function of the heat sink temperature and the temperature lift (ΔT) for France and Germany

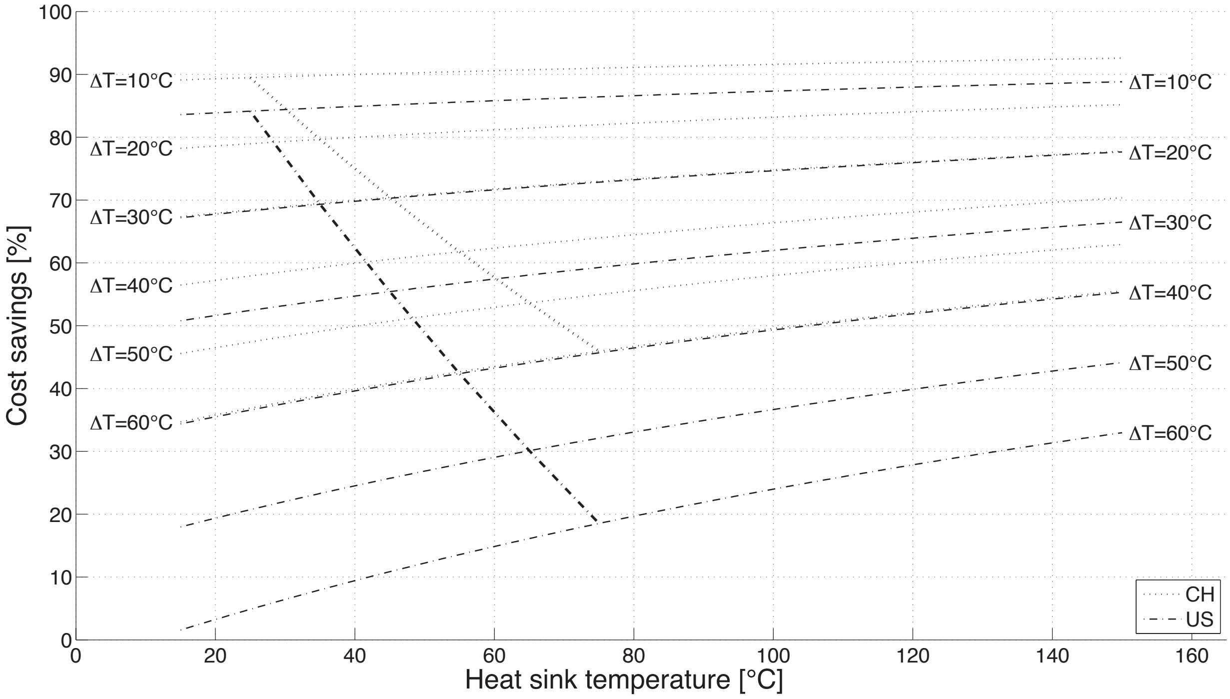


Figure A.4: Relative operating cost savings (Equation (A.11) / (A.3)) as a function of the heat sink temperature and the temperature lift (ΔT) for Switzerland and US

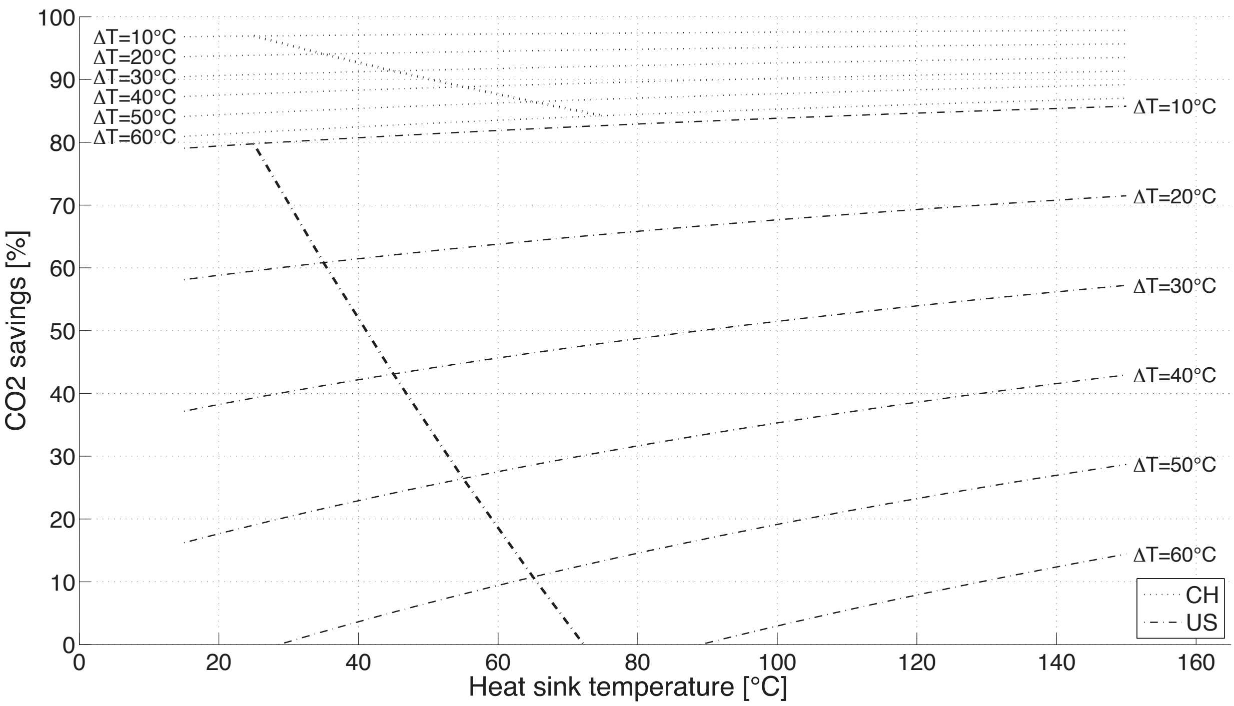


Figure A.5: Relative CO_2 emission savings (Equation (A.12)) as a function of the heat sink temperature and the temperature lift (ΔT) for Switzerland and US

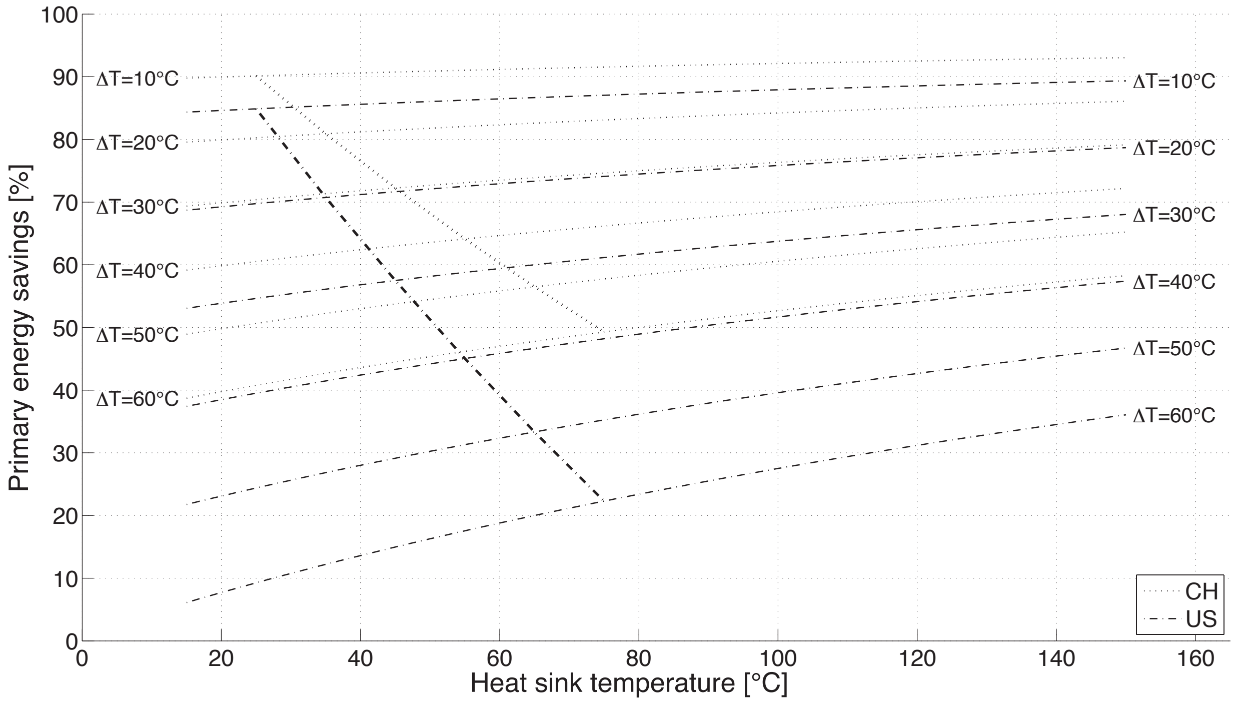


Figure A.6: Relative primary energy savings (Equation (A.13)) as a function of the heat sink temperature and the temperature lift (ΔT) for Switzerland and US

Appendix B

Case studies

B.1 Dairy

B.1.1 Process description

In this dairy process, milk is transformed to produce concentrated milk, pasteurized milk and cream, yogurts and desserts. This is mainly achieved by heat exchanges and evaporation. Operating temperatures and flow rates are fixed by the process recipes. The process is described in Figure B.1. The received milk in the dairy is first cooled down and stored. Then the milk is treated by

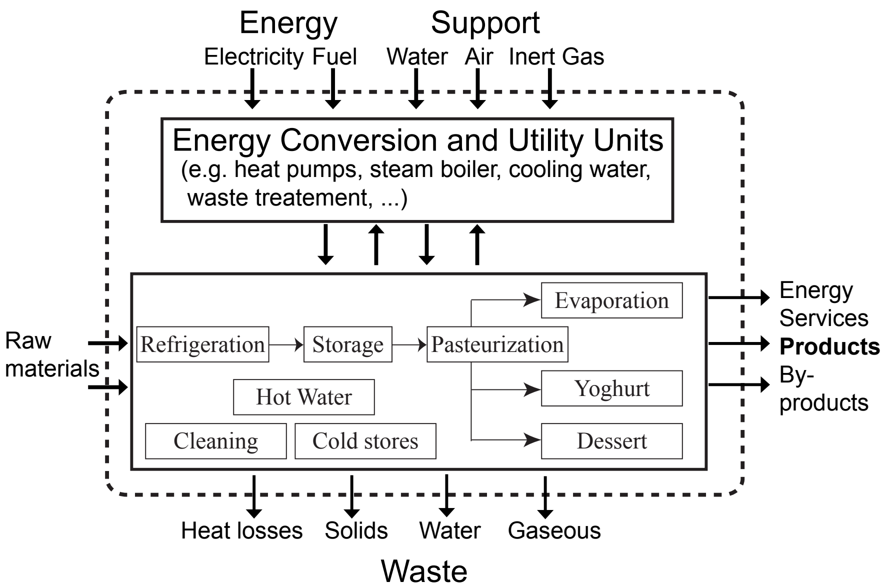


Figure B.1: Process description of the dairy

pasteurization: the milk is first preheated and the cream is separated by centrifugation. Both milk and skimmed milk are pasteurized and then cooled down to the storage temperature conditions. The pasteurized milk is distributed between concentrated milk, yoghurt and dessert production lines. One of the main energy consumer is the milk evaporation. The milk is heated up and then sent to a three effects evaporator. Steam is used in the first effect to evaporate a part of the milk.

The evaporated milk is then used for heating up the inlet milk stream and to evaporate the milk in the second effect. The same principle is used in the third effect. The evaporation temperature is higher than the saturation temperature and depends on the solid content of milk. To ensure the heat recovery in the evaporator, the pressure in the following effect decreases. The evaporation temperatures and pressures for each effect are given in Table B.1. A part of the evaporated milk

Table B.1: Operating conditions of multi-effect evaporator

Effect	1	2	3
T_{eva} [°C]	70.32	66.42	60.82
P [bar]	0.31	0.26	0.20

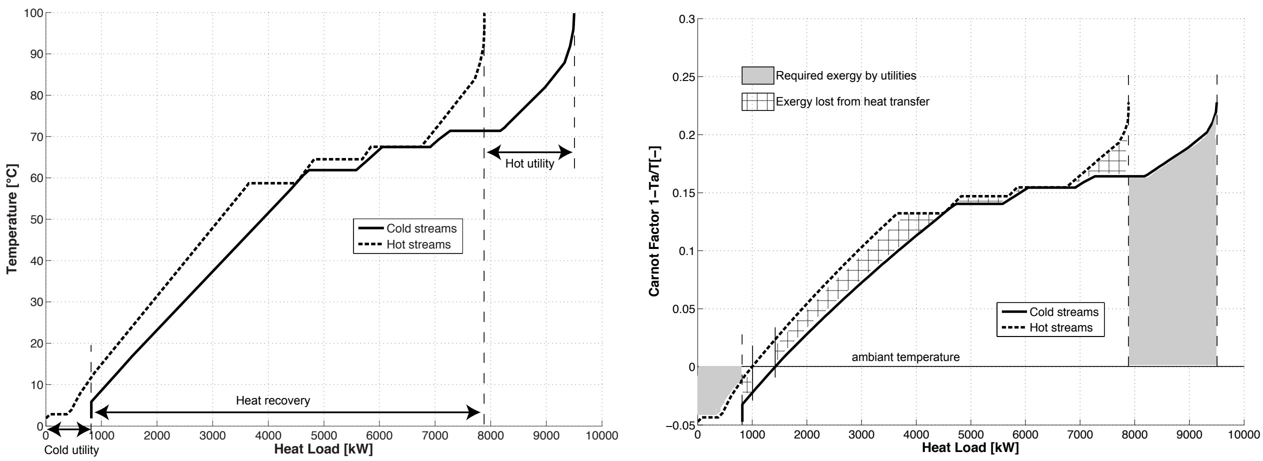
in the third effect is also used to preheat milk feed and the rest is cooled down with cold water. The energy requirements for the yogurt and dessert lines is first heating up the milk, and after homogenization, cooling down the product to the storage temperature. In process integration, it is also important to consider the auxiliary processes outside the direct process lines, like the cleaning in place (CIP) system, the hot water production and the cold stores.

B.1.2 Process integration

B.1.2.1 Definition of process requirements

Table B.2 presents the hot and cold process streams of the dairy under study.

B.1.2.2 Minimum energy requirements



(a) Hot and cold composite curves in the corrected temperature domain

(b) Carnot hot and cold composite curves

Figure B.2: Minimum energy requirement

Table B.2: Hot and cold streams of the dairy process

Unit	Name	Tin [°C]	Tout [°C]	Heat load [kW]	$\Delta T_{min}/2$ [°C]	Remarks
refrig	ref	6.0	4.0	76.0	2.0	refrigeration inlet milk
pasto	pasto1a	4.0	66.0	2356.0	2.0	preheating
	pasto2a	66.0	86.0	676.4	2.0	pasteurization milk
	pasto3a	86.0	4.0	2773.2	2.0	refrigeration milk
	pasto4a	66.0	98.0	119.7	2.0	pasteurization cream
	pasto5a	98.0	4.0	351.6	2.0	refrigeration cream
evap	eva1	4.0	70.3	504.0	2.0	preheating
	eva2	70.3	70.3	904.2	1.2	evaporation 1.effect
	eva3	66.4	66.4	864.1	1.2	evaporation 2.effect
	eva4	60.8	60.8	849.8	1.2	evaporation 3.effect
	eva5	60.8	4.0	151.5	2.0	refrigeration concentrated milk
	eva6	68.9	68.9	904.2	1.2	condensation 1.effect
	eva7	65.9	65.9	864.1	1.2	condensation 2.effect
	eva9	68.9	15.0	87.8	2.0	condensation 3.effect
	eva10	65.9	15.0	80.8	2.0	cooling condensates 1.effect
evap rec	eva8	60.1	60.1	849.8	1.2	cooling condensates 2.effect
	eva11	60.1	15.0	69.7	2.0	cooling condensates 3.effect
yog	yog1	4.0	94.0	1026.0	2.0	heating
	yog2	94.0	10.0	957.6	2.0	cooling
des	des1	4.0	90.0	817.0	2.0	heating
	des2	90.0	70.0	190.0	2.0	cooling
hwat	hw	15.0	55.0	167.2	2.0	hot water prodcution
CIP	CIP1a	58.7	70.0	188.6	2.0	maintain temperature CIP1
	CIP1b	65.0	15.0	104.5	2.0	recuperation waste heat CIP1
	CIP2a	67.5	80.0	209.5	2.0	maintain temperature CIP2
	CIP2b	75.0	15.0	125.4	2.0	recuperation waste heat CIP2
frigsto	frig	5.0	5.0	300.0	2.0	maintain storage temperature

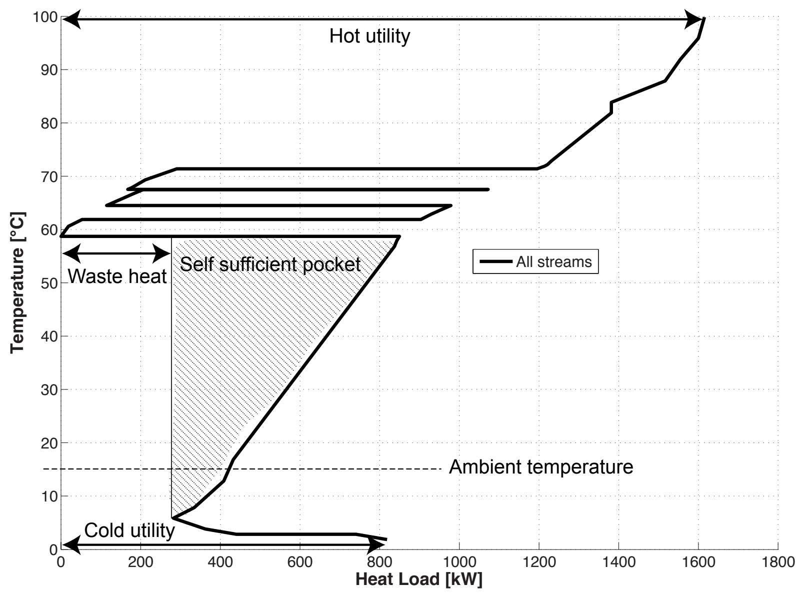


Figure B.3: Process grand composite curve

B.1.2.3 Utility integration

Following scenarios are evaluated.

Case 0 The current situation for the given period is used for comparison (reference case).

Case 1 Maximum heat recovery is calculated and the on site available refrigeration cycle and conventional boiler are integrated.

Case 2 On site available utilities like in Case1 are integrated. The evaporation temperatures of the refrigeration cycle are adapted to the process cooling demand.

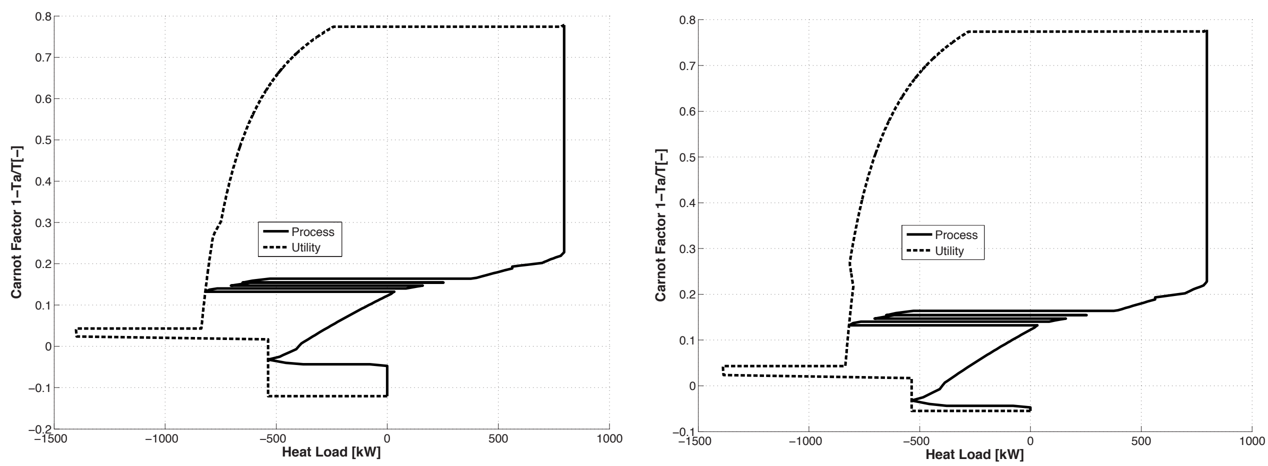
Case 3 As Case 2 but additionally, a co-generation engine is integrated.

Case 4 As Case 2 but additionally, a heat pump is integrated.

Case 5 As Case 4 but temperature levels of refrigeration cycle and heat pump are adapted.

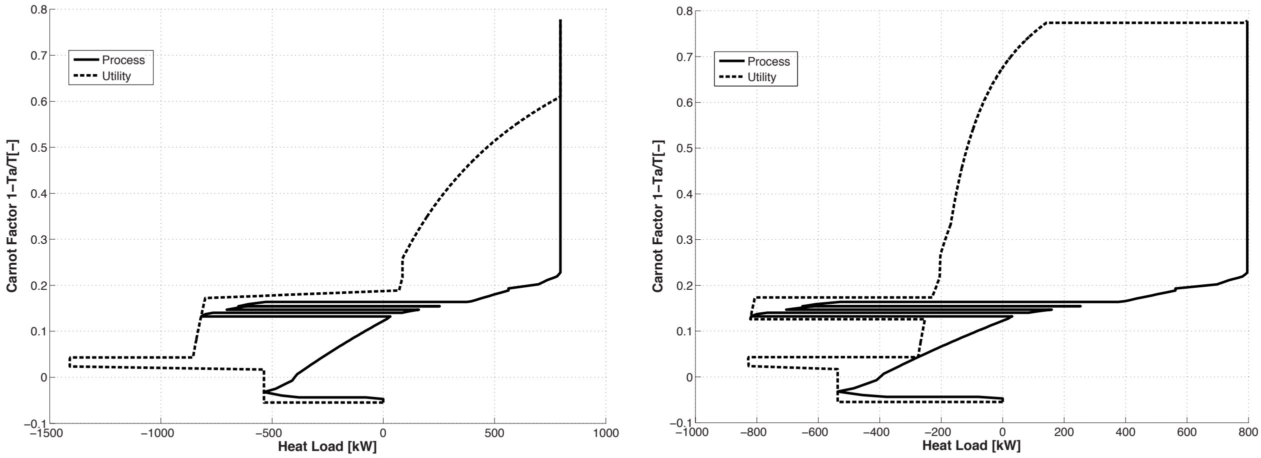
Case 6 Heat pump and refrigeration cycles with adapted temperature levels, co-generation and other on site available utilities (e.g. steam boiler) are integrated simultaneously.

The integrated composite curves are given in Figures B.4 to B.6.



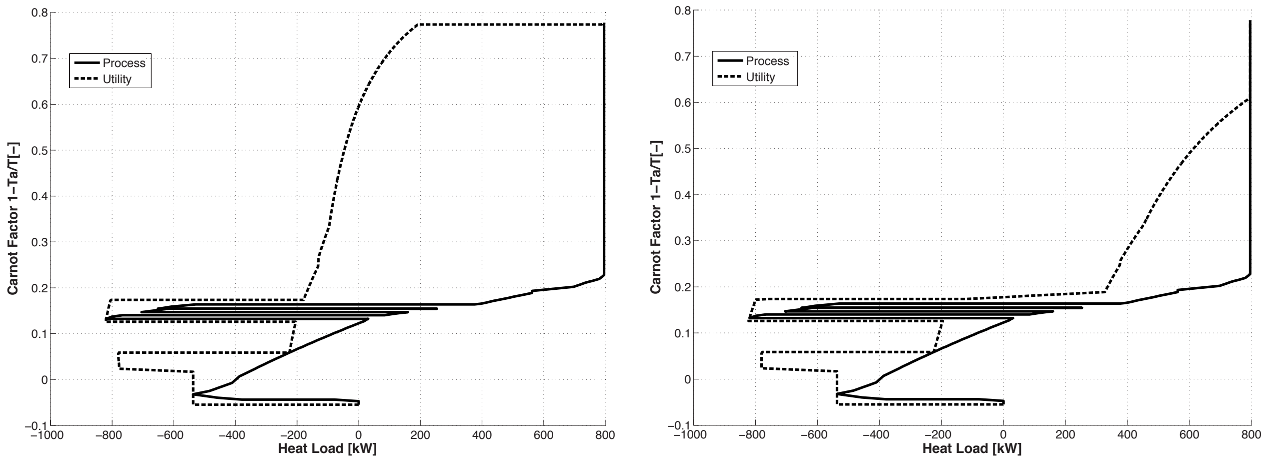
(a) Case 1 (b) Case 2

Figure B.4: Dairy integrated composite curves (Cases 1&2)



(a) Case 3 (b) Case 4

Figure B.5: Dairy integrated composite curves (Cases 3&4)



(a) Case 5 (b) Case 6

Figure B.6: Dairy integrated composite curves (Cases 5&6)

Table B.3: Results

	Unit	Case 0	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
OpC	[k€/year]	269	199	195	186	132	128	118
E_f	[MWh/year]	6164	4404	4563	8416	2772	2562	4671
E_{el}	[MWh/year]	443	429	252	-2900	382	439	-1308
Q_{cw}	[MWh/year]	n.a.	2283	2249	2311	767	635	645
M_{CO_2}	[t/year]	1286	929	945	1433	595	558	823
E_p	[GJ/year]	32954	24881	23504	3683	16979	16703	5595
$InvC$	[k€]		316	326	1517	486	489	1291
PB	[year]		4.5	4.4	18.3	3.6	3.5	8.6
AP	[k€/year]		44	48	-39	98	102	47

B.1.2.4 Summary of results

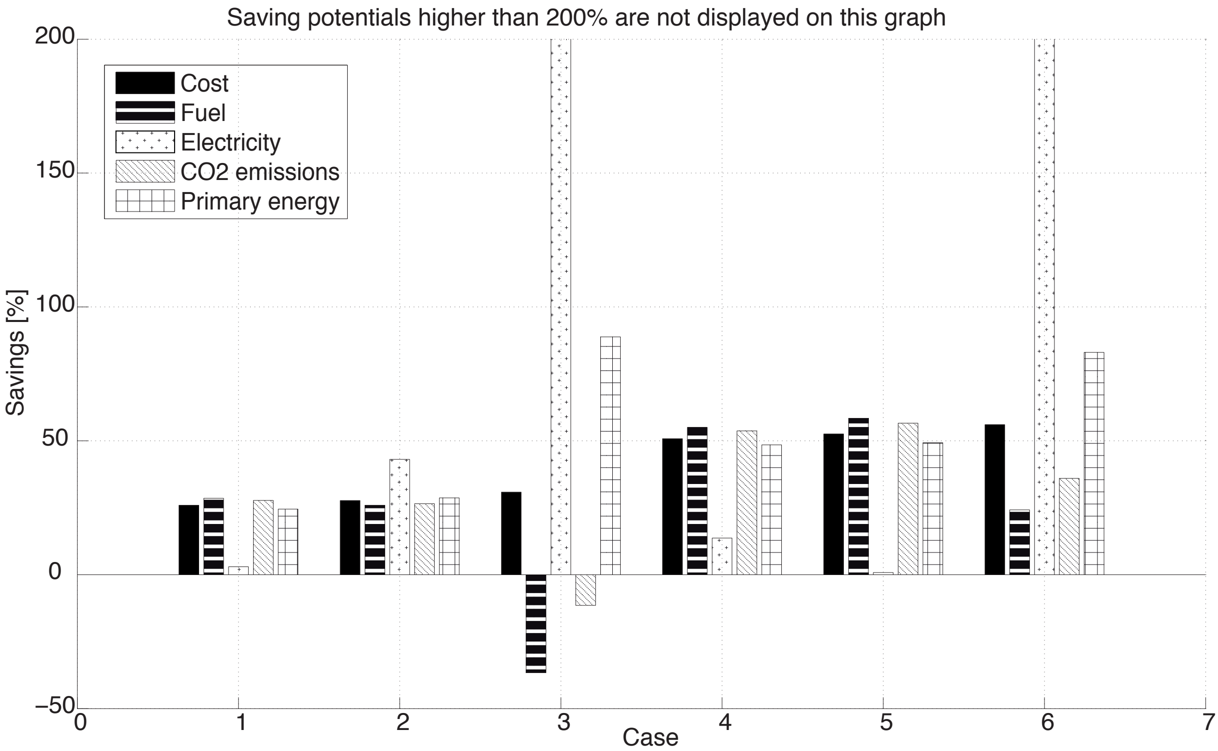


Figure B.7: Comparison of saving for Case 1 to 6

Table B.4: Detailed heat pump results for Case 1

Unit	Fluid	Compressor	Power	T_{eva}	T_{cond}	InvC	COP_H	COP_C
			[kW]	[°C]	[°C]	[kEuro]	[-]	[-]
HP 1	R717	CENTR	162	-18	30	0.0	4.3	3.3
COG	-	-	0	-	-	0	-	-

Table B.5: Detailed heat pump results for Case 2

Unit	Fluid	Compressor	Power	T_{eva}	T_{cond}	InvC	COP_H	COP_C
			[kW]	[°C]	[°C]	[kEuro]	[-]	[-]
HP 1	R717	CENTR	95	-2	30	0.0	6.6	5.6
COG	-	-	0	-	-	0	-	-

Table B.6: Detailed heat pump results for Case 3

Unit	Fluid	Compressor	Power	T_{eva}	T_{cond}	InvC	COP_H	COP_C
			[kW]	[°C]	[°C]	[kEuro]	[-]	[-]
HP 1	R717	CENTR	95	-2	30	0.0	6.6	5.6
COG	-	-	-3152	-	-	1221.74	-	-

Table B.7: Detailed heat pump results for Case 4

Unit	Fluid	Compressor	Power	T_{eva}	T_{cond}	InvC	COP_H	COP_C
			[kW]	[°C]	[°C]	[kEuro]	[-]	[-]
HP 1	R717	CENTR	95	-2	30	0.0	6.6	5.6
HP 2	water	CENTR	49	56	76	124.3	12.6	11.6
COG	-	-	0	-	-	0	-	-

Table B.8: Detailed heat pump results for Case 5

Unit	Fluid	Compressor	Power	T_{eva}	T_{cond}	InvC	COP_H	COP_C
			[kW]	[°C]	[°C]	[kEuro]	[-]	[-]
HP 1	R717	CENTR	112	-2	35	0.0	5.8	4.8
HP 2	water	CENTR	53	56	76	133.9	12.6	11.6
COG	-	-	0	-	-	0	-	-

Table B.9: Detailed heat pump results for Case 6

Unit	Fluid	Compressor	Power	T_{eva}	T_{cond}	InvC	COP_H	COP_C
			[kW]	[°C]	[°C]	[kEuro]	[-]	[-]
HP 1	R717	CENTR	112	-2	35	0.0	5.8	4.8
HP 2	water	CENTR	54	56	76	135.7	12.6	11.6
COG	-	-	-1750	-	-	821.4	-	-

B.2 Cheese

B.2.1 Process description

The main process operation steps of a cheese process are summarized on Figure B.8. The current energy bill corresponds to the energy consumption of 2895 kWh of natural gas and 194 kWh_{el} of electricity for one ton of produced cheese. The global process including auxiliary units (e.g. hot water production, space heating, cleaning in place, ..) is considered. Several pasteurization units are used to remove bacteria from milk, cream or water. The evaporation unit, one of the main consumers, consists of 5 effects and one thermal vapour re-compression. Before entering the evaporation in the first effect, the whey is first preheated to reach predefined operating conditions that leads to a first evaporation. Then, the remaining liquid is sent to the second effect at a lower pressure, and the vapour boiled off in the first effect is recovered to provide heat to the second effect. The same operation is repeated in the following effects. A part of the steam from the third effect is reused in the thermal vapour re-compression (TVR) driven by high pressure steam. The remaining heating and cooling requirements concern process units like forming, product refining, packaging or cold stores.

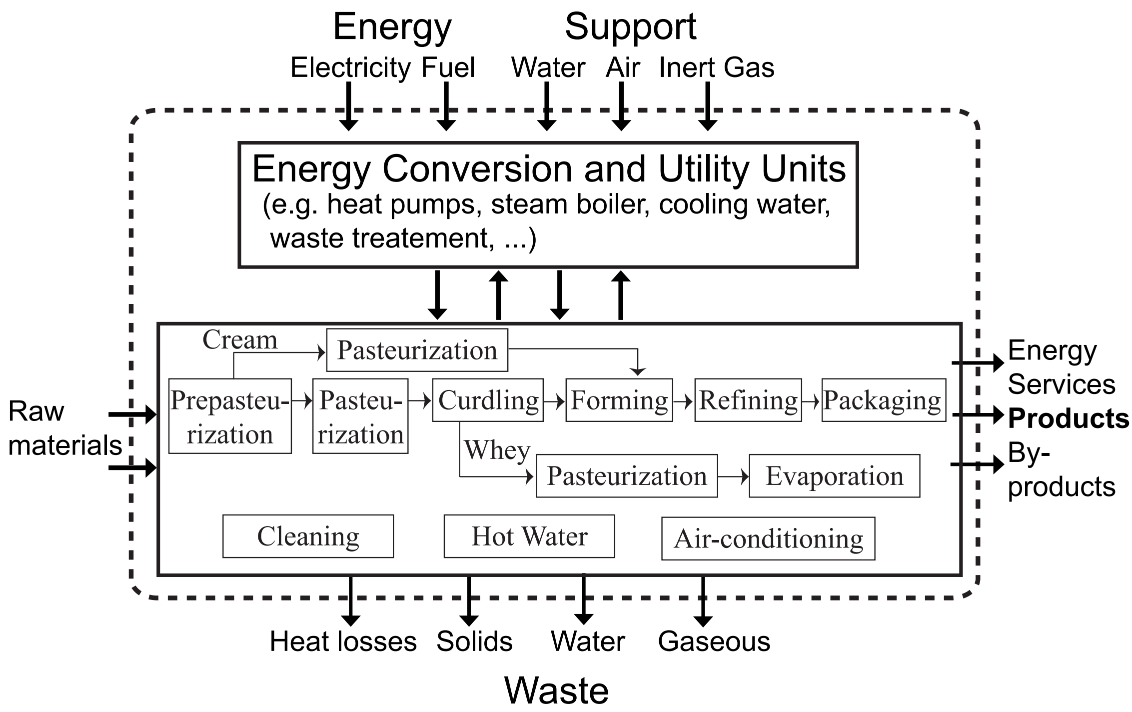


Figure B.8: Process description of the cheese factory

B.2.2 Process integration - average model

B.2.2.1 Definition of process requirements

The process operations are not simultaneous. In order to avoid multi-period problems, the energy consumption is expressed by a mean value related to the energy consumption of 1 ton of cheese (kWh/t). To make batch processes continuous and easier to solve, Linnhoff et al. (1988) introduced the time average approach. Heat exchange becomes more difficult for multi-period process and it has to be indirect through heat transfer units and storage tanks. For this, all units of different periods are defined as independent sub-systems. The heat transfer between those sub-systems can only be realized by intermediate heat transfer networks. For the cheese factory, following process units are not always working simultaneously and are therefore defined in sub-systems with no possibility of direct heat exchange (evapo, pasto 1-5 and proc6). Furthermore, the $\Delta T_{min}/2$ has to be adapted to include the fact that the real instant heat load will be higher than the mean heat load. For all other units, the $\Delta T_{min_u}/2$ value is calculated with Equation B.1.

$$\Delta T_{min_u}/2 \simeq \frac{d}{d_u} \Delta T_{min_{ref}}/2 \quad \text{(B.1)}$$

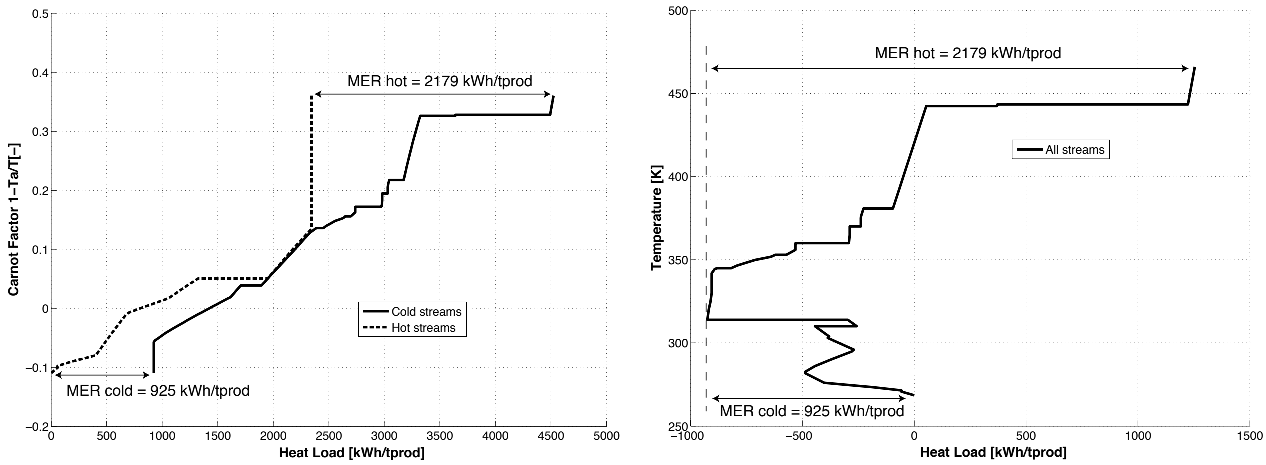
Finally, the complete list of streams with corresponding heat loads and $\Delta T_{min}/2$ values is reported in Table B.10.

Table B.10: Process streams of the cheese factory

Unit	Name	Tin [°C]	Tout [°C]	Heat load [kWh/t]	$\Delta T_{min_u}/2$ [°C]	t_{tot}/t_u [-]
other	other_c1	100	190	367.8	2.0	1
	other_h1	5	0.5	307	2.0	
	other_h2	-0.3	-2.5	56.9	2.0	
evapo tech	evapo_c1	100	190	993.7	3.1	1.7
	evapo_h1	44	5	32.9	3.1	
	evapo_h2	44	25	198.1	3.1	
	evapo_h3	44	44	627.8	3.1	
pasto1	pasto1_c1	6	48	568.2	3.6	1.8
	pasto1_c2	48	75	344	3.6	
	pasto1_h1	75	4	904.6	3.6	
pasto2	pasto2_c1	79	85	5	6.8	3.4
	pasto2_h1	54	4	41.9	6.8	
pasto3	pasto3_c1	74	80	84.1	2.9	1.5
	pasto3_c2	6	28	308.2	2.9	
pasto4	pasto4_c1	69	75	32.1	4.5	2.3
	pasto4_c2	8.5	26	83.3	4.5	
pasto5	pasto5_c1	66	76	54.2	2.9	1.4
proc6	proc6_c1	105	105	131	2.8	1.4
	proc6_c2	78	78	49.6	2.8	
	proc6_c3	95	95	49.6	2.8	
proc7	proc7_c1	15	55	40.4	2.0	1
proc8	proc8_c1	70	70	62.6	2.0	1
proc9	proc9_c1	35	35	33.8	2.0	1
proc10	proc10_c1	32	25	175.5	2.0	1
heat	heat_c1	35	35	153	2.0	1
CIP	clean_c1	85	85	238	2.0	1

B.2.2.2 Minimum energy requirements

First, the maximum heat recovery is computed. Exergy losses can be visualized on Figure B.9a, which shows the hot and cold Carnot composite curves. The process grand composite curve, presented in Figure B.9b, is used to define optimal temperature levels for the energy conversion system (utilities). Considering the self-sufficient pockets of Figure B.9b, it can be deduced that



(a) Hot and cold composite curves using the Carnot scale

(b) Grand composite curve

Figure B.9: Minimum energy requirements

the heat recovery between process streams concerns process streams below the pinch point. As process streams without heat exchange restrictions are mainly involved, the heat recovery penalty of restricted matches due to non simultaneously process operations is rather small. The penalty can be evaluated by comparing the case with heat exchange restrictions and no indirect heat recovery to the case without any heat exchange restrictions. The use of intermediate heat transfer networks could only recover a small amount of heat. The exact penalty due to non simultaneous operations can be evaluated using the methodology presented by Becker and Maréchal (2012a). In fact, the penalty of 27 kWh/ t corresponding to 1% of the hot utility and 3% of cold utility, is quite small and can be neglected in the following. This means that no supplementary heat transfer units will be integrated to reduce this penalty.

B.2.2.3 Utility integration

Regarding the necessary temperature levels for heat supply on Figure B.9b, following utility types are proposed to the process:

- Conventional steam boiler, refrigeration cycle and cooling water are available on site and no supplementary investment costs are necessary.
- A co-generation engine providing heat and electricity can be interesting because of rather low temperature levels for a significant part of the hot utility (<120°C). Corresponding investment costs are considered for the results analysis.
- Heat pump units can be integrated for two options; Both options are described below. In the first option, process modifications are not allowed while in the second option the process can

be modified. Investment costs for new installations are included in the results analysis.

All utilities will be integrated simultaneously and heat sources for the refrigeration cycle and heat pumps are considered in the optimization. For example, the cooling water will satisfy a part of the condensation heat of the refrigeration cycle and heat pumps ideally satisfy hot and cold heat demands at the same time. The final overall electricity balance contains the electricity consumption (e.g. from the process or heat pumps) and the electricity production (e.g. co-generation units).

The closed cycle heat pump and the co-generation engine are integrated through an intermediate heat transfer network which means that they cannot exchange heat directly with the process. The results are compared for the French and German context.

Option1 - No process modifications allowed
In the first approach, the process can not be modified. Thus, the direct heat exchange between a potential heat pump and the process is not allowed. The approach presented in appendix is applied. In the higher level two sub-systems (in this case the process and the heat pump) cannot exchange heat directly. Moreover, heat cannot be exchanged directly between different periods and hence the process operation units of different periods are defined in independent sub-systems (e.g. evaporation unit, pasteurization units) of the sub-system process. The evaporation unit can therefore not directly exchange heat with the pasteurization units or the heat pump, but it is possible to exchange heat with the other process units. Satisfying the required temperature levels, a closed cycle heat pump using the refrigerant R245fa has been chosen. By analyzing the shape of the grand composite curve, appropriate operating conditions for the heat pump and the intermediate heat transfer units can be estimated. Simultaneously, utility, heat pump and heat transfer units are considered in the MILP model to define the corresponding optimal flow rates that minimize the operating costs. The potential of a closed cycle heat pump without direct heat exchange is illustrated in Figure B.12a. The saving potential can be increased by minimizing the minimum temperature difference in the heat exchangers between the heat pump and the intermediate heat transfer fluids. The advantage of this approach is that only the investment costs related to the heat pump, heat exchangers and pipes have to be accounted. The process itself remains unchanged. Thus, the safety and product quality aspects are also maintained.

Option2 - Process modifications allowed
In the second option, process modifications are allowed. This leads to more heat pump integration opportunities. For example, operating pressures of the process can be modified to improve heat recovery, and mechanical vapour re-compression (MVR) units can replace the thermal vapour re-compression unit. **Case 2** replaces the thermal vapour re-compression with a mechanical vapour re-compression. The layout and the pressure levels of the evaporation effects are kept. The steam leaving the third effect at about 60 °C is compressed mechanically to 74 °C. As the ΔT is less than 18 °C, a dynamic compressor is suitable and therefore selected (Figure B.12b). In **Case 3**, all effects are realized in parallel and mechanical vapour re-compression is integrated. The temperature difference for a mechanical vapour re-compression in Figure B.12c is small (<10 °C). The pressure levels of the five effects are adapted, so that all effects evaporate at about 70 °C. In **Case 4**, the pressure of effects are modified and mechanical vapour re-compressions are included: first the new temperature levels have to be defined. Assuming that the heat exchange area in the effects will not be changed, the temperature levels of effects 4 and 5 are reduced (see Equation (B.2)). The new temperature levels are 48 °C (effect 4) and 32 °C (effect5). Theoretically, it could be possible to raise waste heat at 32 °C with a heat pump to satisfy a part of the heat demand in effect 1. However, high temperature lifts make such heat pump integration not optimal. The use of successive mechanical vapour re-compressions between the different effects have therefore been preferred (Figure B.12d).

The temperature of effect 5 is quite low. If these operating conditions can not be reached in the unit, it could also be possible to raise the temperature levels of all effects.

$$\dot{Q} = U \cdot A \cdot (T_{vap} - T_{prod}) \tag{B.2}$$

B.2.2.4 Results

Results of analyzed scenarios will be summarized in Table B.11. Several cases are compared to the current case (Case 0). First (for cases 1a-d) the current evaporation unit including its thermal vapour re-compression is modeled in the technology representation. Three scenarios with different hot utilities have been evaluated: boiler (Case 1a), boiler and heat pump (Case 1b), boiler and engine (Case 1c) and boiler, heat pump and engine (Case 1d). For heat pump integration option 2 (Cases 2,3 and 4) the evaporation units are modeled in the thermodynamic representation: MVR and boiler (Case 2a) and MVR, boiler and engine(Case 2b). For Cases 3a/b and Cases 4a/b the evaporation unit is first modified.

For all cases, the on-site available utilities (boiler, refrigeration and cooling water) are integrated. The first lines of Table B.11 indicates the selected option and additional proposed utilities. As mentioned above, the applied MILP formulation minimizes the operating costs (including fuel and electricity costs). Furthermore, the utility integration is optimized to satisfy the process demand and the product quality. Keeping the process recipe as it is, this paper focuses on the rational conversion of energy. To study the impact of different fuel and electricity prices and the corresponding electricity mix, the results of the case study have been compared for France and Germany.

B.2.2.5 Global saving results

All cases are compared to the current case before heat recovery (Case 0). The results are presented in Table B.11. Beside the operating costs, the global electricity and fuel consumption, CO_2 emissions and primary energy consumption are presented in this table. Bold values show the best option for each criterion. It is interesting to remark that the proposed utilities in most of the cases (Cases 2a/b,3a/b and 4a/b) are integrated in the same way for France and Germany. A difference can be recognized in Cases 1c/1d, where co-generation engines are favored in Germany. Compared to the French case, the optimal size of a co-generation engine in Germany would be more than twice as high in Case 1c. In Case 1d, the co-generation is not integrated at all in France. A sensitivity analysis of the fuel to electricity ratio could identify different integration zones for the heat pump and the co-generation engine. A major difference between the two countries concerns the CO_2 emissions. Taking the current case (Case 0), the CO_2 emissions in Germany are higher, however the possibility of integrating heat pumps and in particularly co-generation engines leads to a higher saving potential in CO_2 emissions. As the electricity in France is mainly produced by nuclear and hydro power plants, the interest of co-generation engines is smaller. Whereas in Germany, the CO_2 content in electricity (mainly produced by coal) is relatively high, and thus efficient co-generation engines reduce not only the energy consumption of the process but also the CO_2 emissions. Another advantage of co-generation engines is the possibility to sell the surplus of produced electricity to the grid (e.g. Cases 2b, 3b and 4b in France and Germany). However, this depends strongly on the given selling prices. Figures B.10 and B.11 show the saving potential in France and Germany respectively. The saving potential of a well integrated process without any supplementary heat pump (closed cycle heat pump or MVR) or co-generation unit is about 22% in France and 23% in Germany. This means that the heat recovery is fully exploited and for this,

Table B.11: Process integration results for France (FR) and Germany (DE)

Case		0	1a	1b	1c	1d	2a	2b	3a	3b	4a	4b
Option		-	1	1	1	1	2	2	2	2	2	2
Utilities		-	-	hp	cog	hp cog	mvr	mvr cog	mvr	mvr cog	3mvr	3mvr cog
OpC [e/t]	FR	126.0	97.8	87.6	95.9	87.6	88.9	85.5	86.2	81.3	69.4	**65.5**
	DE	166.0	127.3	117.4	117.7	114.6	117.9	93.3	117.7	94.0	95.3	**79.6**
e_f [kWh/t]	FR	2895.0	2358.5	1921.7	2527.1	1921.7	2023.7	3041.6	1770.5	2605.3	**1394.9**	1893.0
	DE	2895.0	2358.5	1921.7	2758.6	2293.0	2023.7	3041.6	1770.5	2605.3	**1394.9**	1893.0
e_{el} [kWh/t]	FR	194.0	86.6	197.1	-63.2	197.1	154.8	**-680.3**	270.2	-419.9	237.0	-174.6
	DE	194.0	86.6	197.1	-234.2	0.0	154.8	**-680.3**	270.2	-419.9	237.0	-174.6
Cooling [kWh/t]	FR	-	955.3	672.7	945.2	672.7	1003.6	1017.1	**464.1**	469.8	512.9	516.3
	DE	-	955.3	672.7	968.9	798.0	1003.6	1017.1	**464.1**	469.8	512.9	516.3
m_{CO_2} [kg/t]	FR	602.6	484.4	406.3	504.6	406.3	423.0	551.8	382.5	487.6	**303.6**	366.3
	DE	707.2	531.0	512.6	409.5	463.2	506.5	**185.1**	528.2	261.3	431.3	272.2
e_p [GJ/t]	FR	15.3	11.6	11.0	10.6	11.0	10.9	**5.7**	11.2	6.8	9.1	6.5
	DE	15.2	11.6	10.8	9.9	10.3	10.8	**6.2**	10.9	7.1	8.9	6.6
$\dot{E}_{hp}$ [kW]	FR	-	0.0	246.8	0.0	246.8	239.1	264.5	641.9	651.7	525.9	539.1
	DE	-	0.0	246.8	0.0	133.4	239.1	264.5	641.9	651.7	525.9	539.1
$InvC_{hp}$ [ke]	FR	-	0.0	794.6	0.0	794.6	516.7	565.9	1256.9	1274.2	1160.3	1185.4
	DE	-	0.0	794.6	0.0	456.9	516.7	565.9	1256.9	1274.2	1160.3	1185.4
$\dot{E}_{cog}$ [kW]	FR	-	0.0	0.0	-334.4	0.0	0.0	-1880.4	0.0	-1546.9	0.0	-927.2
	DE	-	0.0	0.0	-716.0	-326.7	0.0	-1880.4	0.0	-1546.9	0.0	-927.2
$InvC_{cog}$ [ke]	FR	-	0.0	0.0	413.4	0.0	0.0	961.7	0.0	846.9	0.0	628.6
	DE	-	0.0	0.0	552.7	410.5	0.0	961.7	0.0	846.9	0.0	628.6
PR [$years$]	FR	-	0.0	4.0	11.2	4.0	3.0	6.4	5.5	6.6	2.1	2.9
	DE	-	0.0	4.1	3.0	3.5	2.8	2.3	6.7	3.3	1.9	1.9

new heat exchangers are necessary, but no further utility units are considered. It is important to remark, that the optimal integration of heat pump units has to be compared to the case with optimal heat recovery of the process. The integration of heat pumps and co-generation units will reduce the operating costs. Either option1 (up to 30 %) or option2 where the maximum saving potential reaches up to 50 % of operating costs for the German case and up to 45% for the French case, can be chosen. The electricity consumption in some cases is higher than in the current case, this is due to the integration of heat pumps. An example of integrated composite curves is given

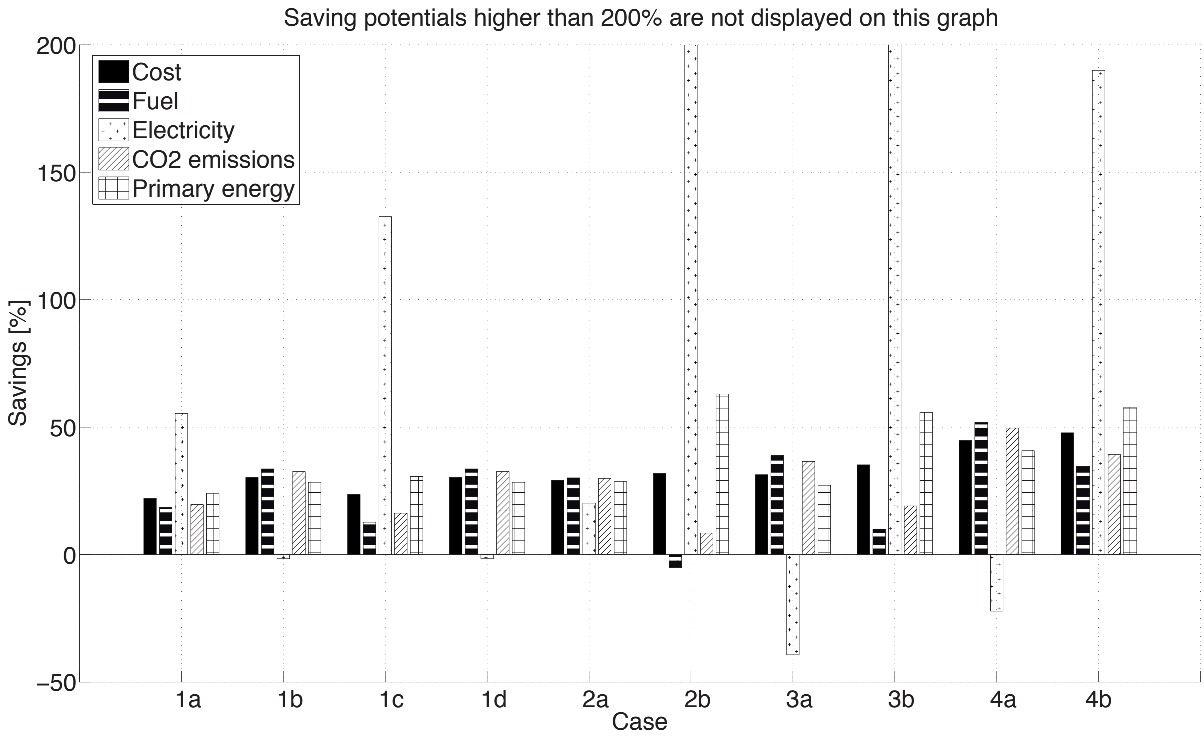


Figure B.10: Saving potentials in France

for France for the Cases 1d, 2b, 3b and 4b in Figure B.12. In Case 1d, the evaporation unit is modeled in the technology representation including the thermal vapour re-compression. A heat pump, a co-generation unit and intermediate heat transfer networks are integrated to ensure the indirect heat transfer between the process and the new utility units. The temperature difference for the heat sink and heat source of the heat pump is relatively high. Nevertheless the overall operating costs and especially the fuel consumption can be reduced. In the following cases, the evaporator is modeled in the thermodynamic representation and more saving opportunities can be pointed out. The necessary temperature difference is smaller in case 2b, where the thermal vapour re-compression is replaced by the mechanical vapour re-compression. By modifying the layout of the evaporation effects, temperatures differences of a mechanical vapour re-compression can be reduced and the saving potential becomes higher. Figure B.12 shows the Carnot composite curves. The surface between the utility and and process composite curves represents the exergy losses.

B.2.3 Process integration - multi-period model

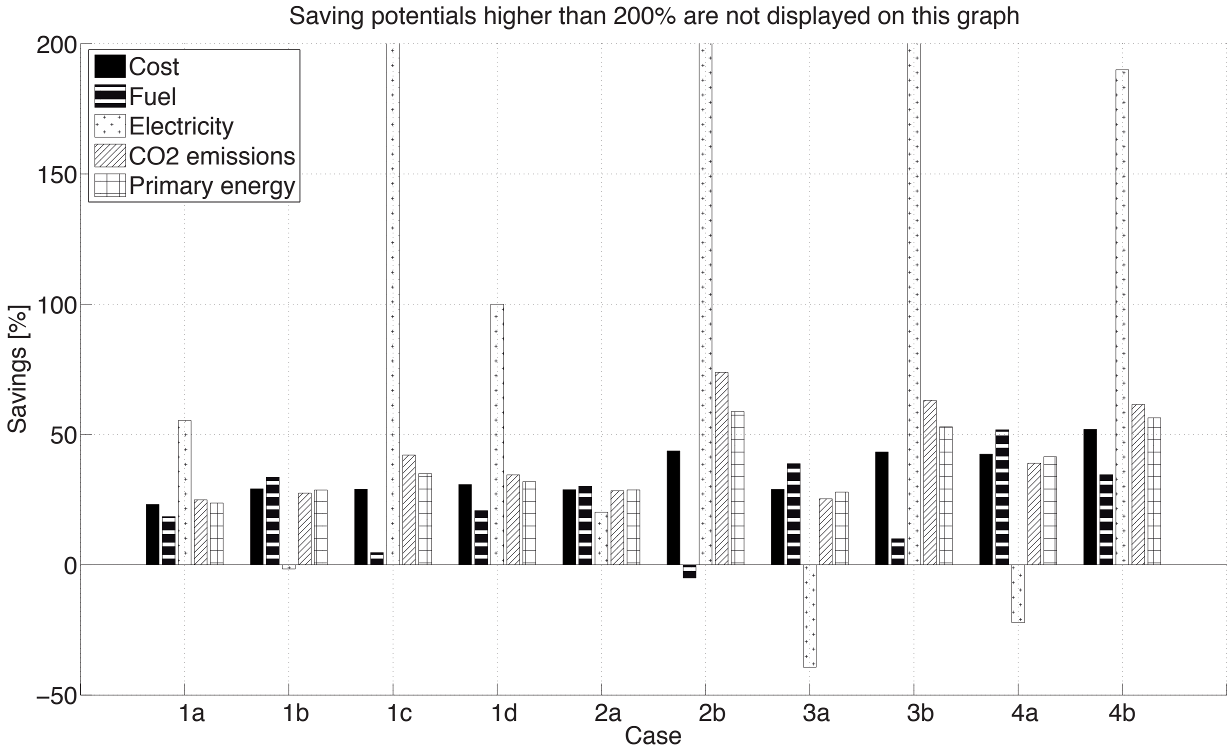
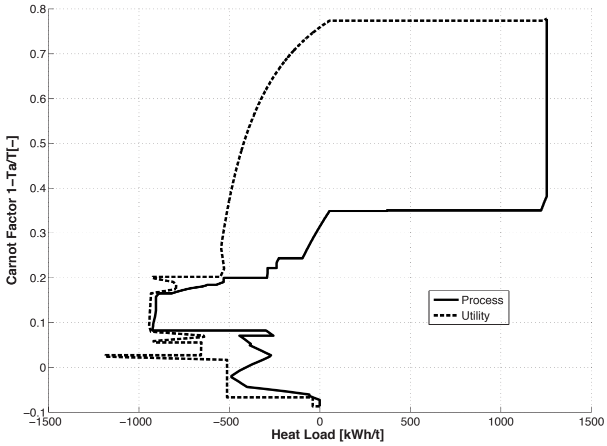


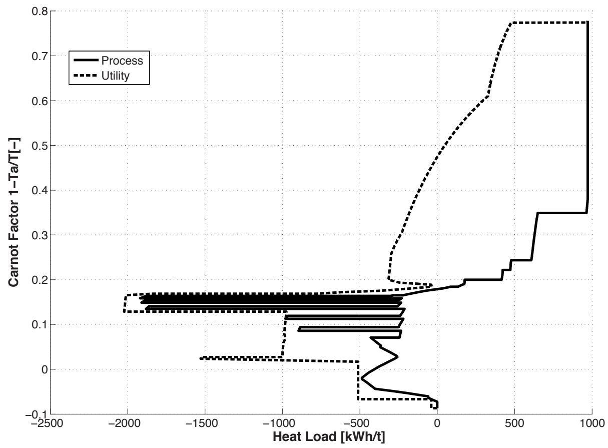
Figure B.11: Saving potentials in Germany

Table B.12: Specific heat pump results

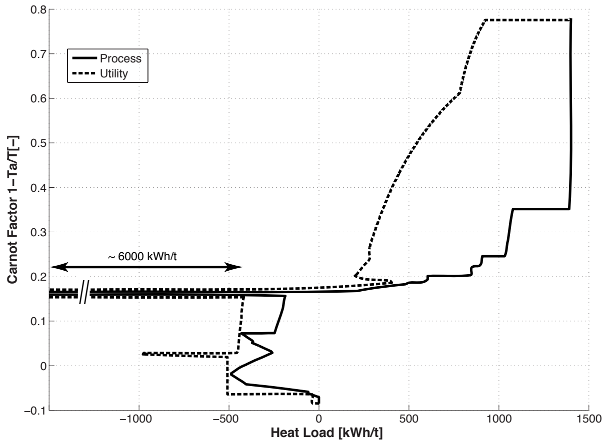
Case	Type	Fluid	T_{eva}	T_{cond}	$\dot{E}_{hp/mvr}$	COP
		[kW]	[°C]	[°C]	[k€]	[-]
1b FR/DE	HP	R245fa	30	90	386	3.6
1d FR	HP	R245fa	30	90	386	3.6
1d DE	HP	R245fa	30	90	209	3.6
2a FR/DE	MVR	water	57	74	239	14.8
2b FR/DE	MVR	water	57	74	264	14.8
3a FR/DE	MVR	water	66	74	642	32.3
3b FR/DE	MVR	water	66	74	652	32.3
4a FR/DE	MVR1	water	28	36	141	28.5
4a FR/DE	MVR2	water	46	52	92	40.4
4a FR/DE	MVR3	water	57	74	294	14.8
4b FR/DE	MVR1	water	28	36	141	28.5
4a FR/DE	MVR2	water	46	52	92	40.4
4a FR/DE	MVR3	water	57	74	307	14.8



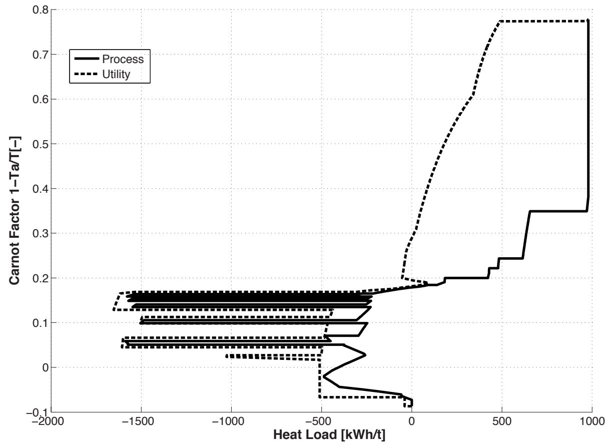
(a) Integrated composite curves Case 1d



(b) Integrated composite curves Case 2b



(c) Integrated composite curves Case 3b



(d) Integrated composite curves Case 4b

Figure B.12: Comparison of Carnot composite curves of the cheese factory

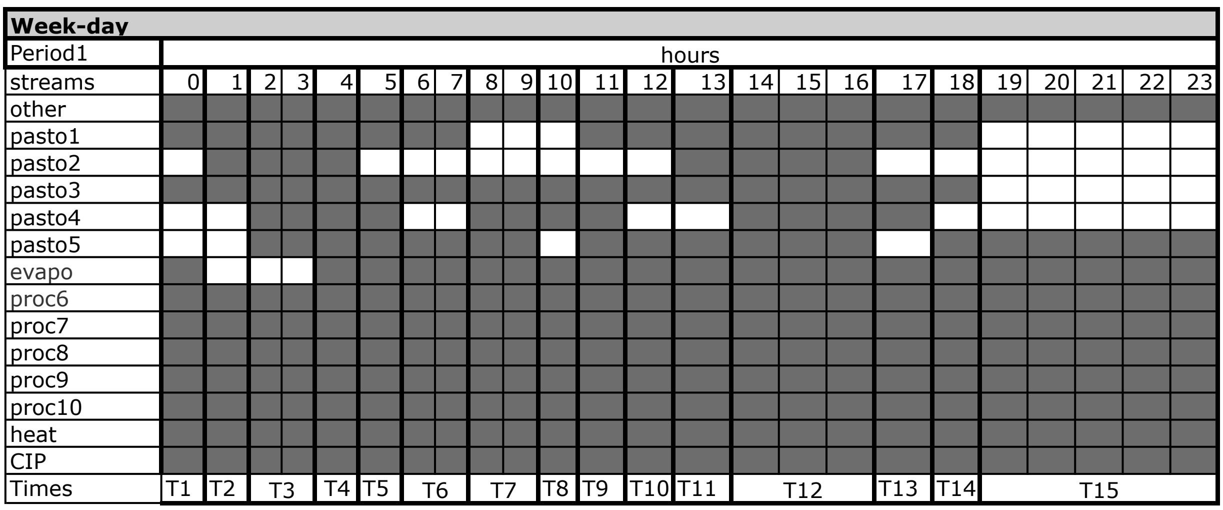


Figure B.13: Working hours for period 1 (week)

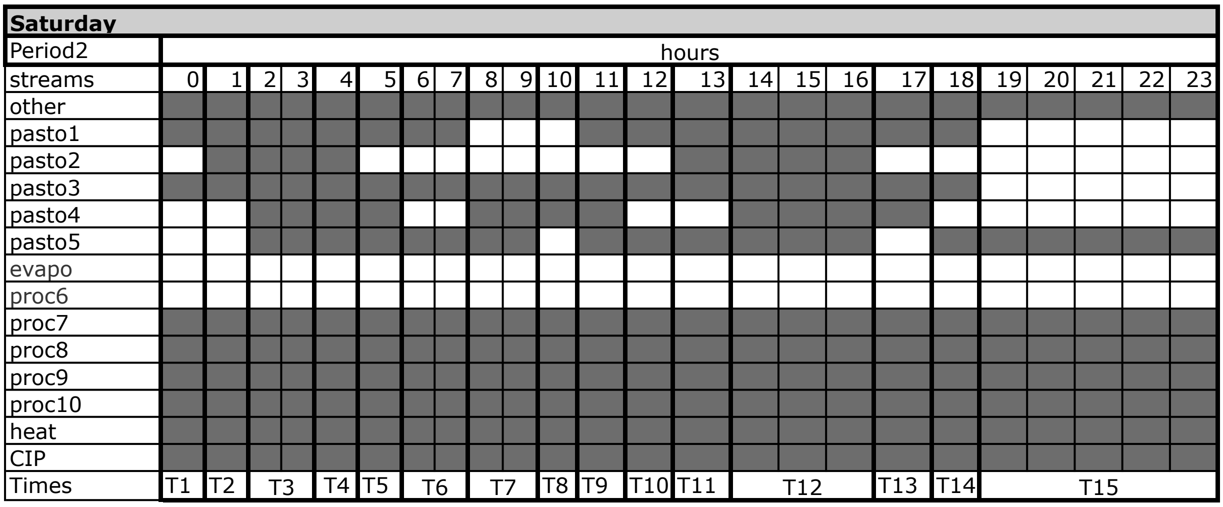


Figure B.14: Working hours for period 2 (week-end)

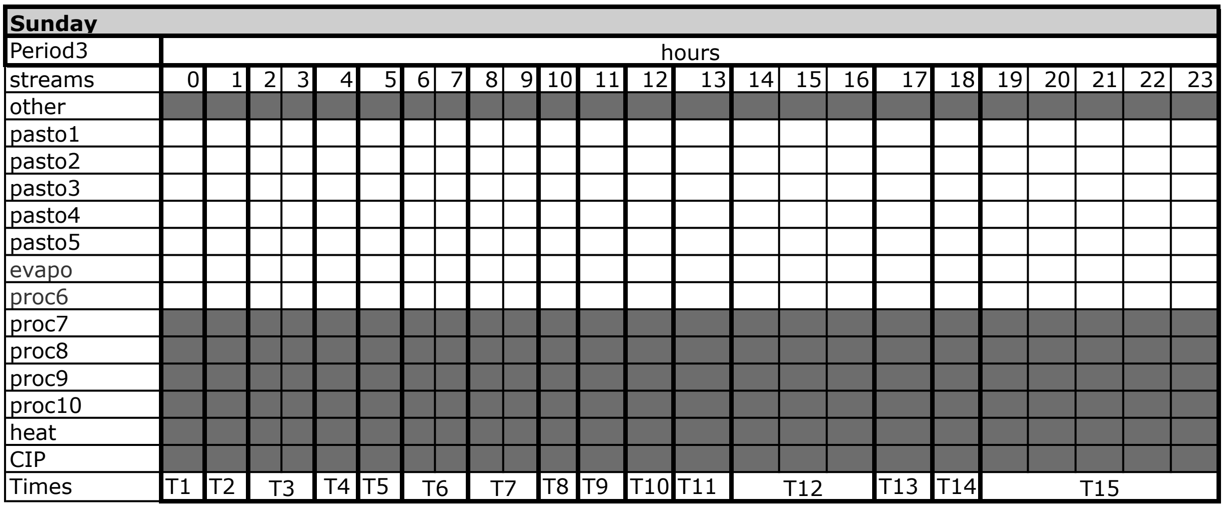


Figure B.15: Working hours for period 3 (week-end)

B.3 Brewery

B.3.1 Process description

The brewery studied corresponds to a typical brewing process. The target temperatures of the streams and the proportions of ingredients are determined by the product recipe. The process is described in Figure B.16.

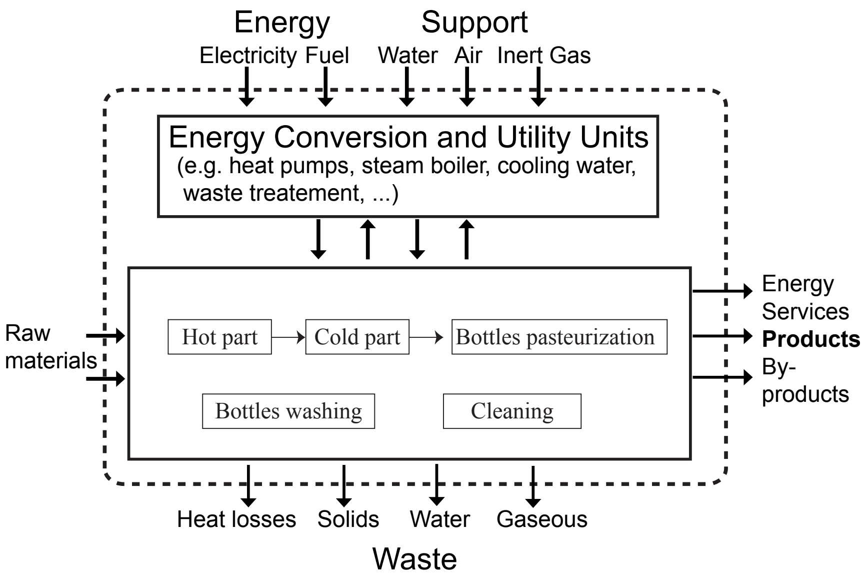


Figure B.16: Process description of the brewery

The brewing house is associated with beer production and is split into two parts:

- **a hot part**(*mashing*), described by the block flow diagram of Figure B.17, where the blend of water and malt (*Mash*) is firstly brewed at high temperature (76°C) so that the activated enzymes transform malt starch into sugar. The *Mash* is then filtered to obtain the wort, which is boiled with hops to develop beer flavors. Wort boiling is an energy intensive operation. The wort is clarified in a whirlpool to remove the hops and eventually cooled to the pitching temperature.

- **a cold part**, illustrated in Figure B.18, mainly consisting of the wort fermentation by yeast, at constant temperature (11°C), during 2 weeks. The beer is then chilled (-2°C) and clarified before being stored in insulated tanks where it ends its maturation.

The rest of the process consists of the beer packaging. In the process under study, four conditioning lines package the beer in new bottles, in kegs and in returnable bottles that are washed beforehand. The bottles filled with beer are then pasteurized.
A Cleaning in Place (CIP) system with effluent recovery, designed to wash the tanks, is also modeled in the study.

Figure B.17: Block flow diagram of the hot part

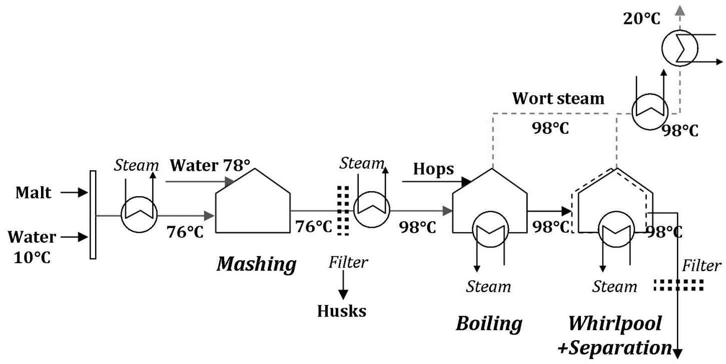
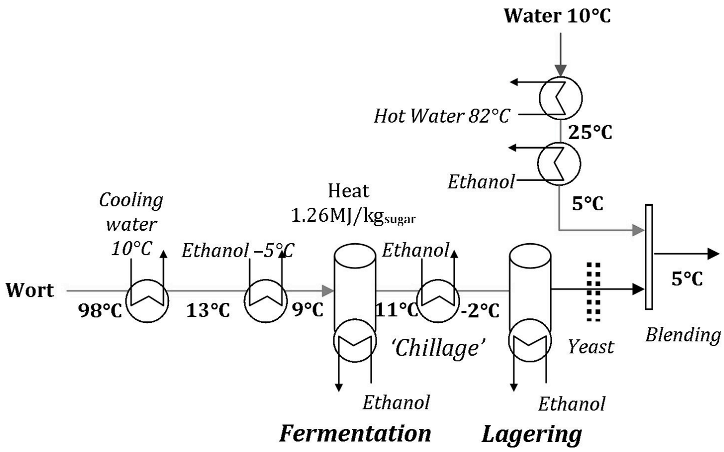


Figure B.18: Block flow diagram of the cold part



B.3.2 Process integration

B.3.2.1 Process integration assumptions

Table B.13: Chosen Values of $\Delta T_{min}/2$

Stream State	$\Delta T_{min}/2$ [Δ°C]
Liquid	2.5
Evaporating	0.8
Condensing	1.7

The Pinch Analysis of the brewery is performed using the following key hypotheses:

- Thermal losses during heat transfers are not taken into account.
- Despite the fact that the units are operated in batch mode, a time averaging approach is used, where all the process operations are considered as being simultaneous. This is done by calculating the overall energy consumed per unit of product and dividing it by the mean hourly production. The yearly operating time of the brewery is 4992h.

- For each stream, the corresponding $\Delta T_{min}/2$ is chosen according to the existing equipments. The associated values of $\Delta T_{min}/2$ may not be optimal; however they are used in the study, as they correspond with the existing heat exchangers available in the factory.

B.3.2.2 Modeling of the Conditioning Lines

The opportunity of recovering heat from the conditioning lines is worth studying, since bottle washing and pasteurization devices represent important energy consumers in breweries. In the process under study, the conditioning lines account for more than 32% of the current heating demand, which reveals the importance of modeling and integrating these units when undertaking the pinch analysis of breweries.

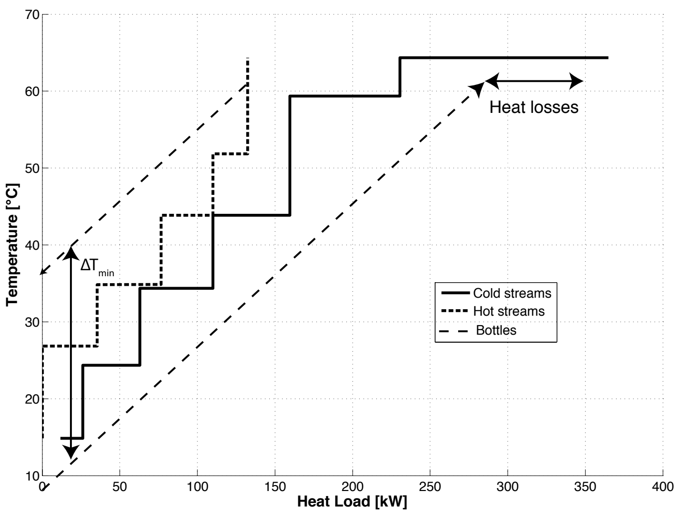
As an example, the modeling of the bottle pasteurization device is presented in this paragraph. The device is considered as a sequence of soaking baths transferring their heat to the beer bottles passing through them. The bottles are successively heated and cooled; the baths thus require respectively heating and cooling supplies in order to keep a constant temperature level.

The model considers the different baths at their corresponding temperatures. This representation enables the determination of internal heat recovery potential, as well as between the baths and other process streams.

In the study, permanent regime is considered. The current bottle pasteurization device consists of ten baths maintained at constant temperature levels. The input and output temperatures of the bottles are respectively 8°C/281K and 30°C/303K. The main soaking bath is kept at 62°C/335K.

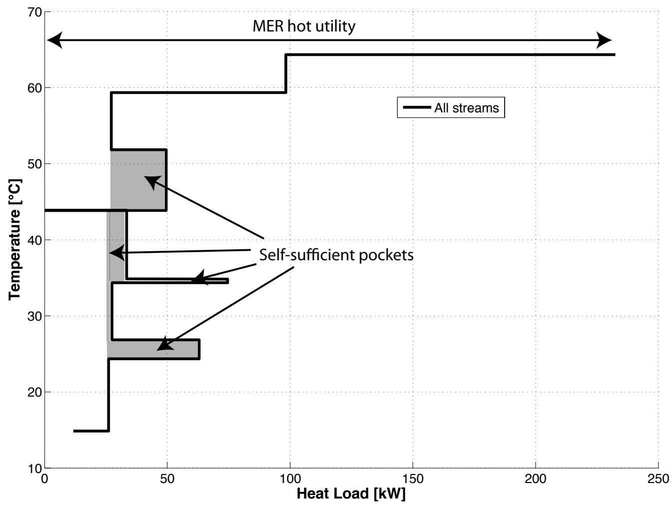
The computed heat loads of the different baths correspond to the sum of the bottle heat loads and the heat losses to the surroundings (from conduction-convection and from radiation). The composite curves associated with the device can thus be obtained and represented in a $(T - \dot{Q})$ diagram. Figure B.19 shows the composite curves corresponding to the device operating on the production line n°4. 40000 bottles of 0.33L/unit are currently pasteurized per hour by the machine.

Figure B.19: Bottle pasteurization device (line 4) composite curves



In Figure B.19, the hot streams are associated with the baths in heat excess and the cold ones with those requiring heat. The grand composite curve corresponds to the enthalpy (heat) difference between the hot and cold streams for each temperature interval. In Figure B.20, the integration shows the possible heat recovery that can be obtained by transferring hot water from one cooling bath (hot stream at constant temperature) to a heating bath (cold stream at constant temperature).

Figure B.20: Bottle pasteurization device (line 4) grand composite curve



It can be noted that the device heat recovery potential is determined not only by the ΔT_{min}, but also by the number of baths and by their temperature levels. Thus, the bottling system design can be optimized using pinch analysis, in particular through the definition of the minimum number of baths and their corresponding volumes that can be expressed as a function of the speed of the bottle processing.

B.3.2.3 Definition of process requirements

Table B.14 presents the hot and cold process streams of the brewery process under study.

B.3.2.4 Minimum energy requirement

The process requirements identified for the process operation units are used to calculate the maximum energy recovery in the system. Figure B.21 presents the brewery composite curves resulting from the definition of the hot and cold streams identified in the process.

The computation of the MER for the identified process operation units enables the identification of opportunities for energy saving.

B.3.2.5 Utility integration

The analysis of the Grand composite curve (Figure B.22) leads to the following observations:

- Heat is required at relatively low temperature levels which offers the opportunity to integrate combined heat and power (CHP) and heat pumping systems.

Table B.14: Hot and cold streams of the brewery process

Unit	Name	Tin [°C]	Tout [°C]	Heat load [kW]	$\Delta T_{min}/2$ [°C]	Remarks
hotcold	bc_c1	10.0	25.0	61.5	2.5	
	bc_h1	11.0	11.0	483.8	2.5	
	bc_h2	11.0	-2.0	182.2	2.5	
	bc_h3	-2.0	-2.0	35.4	2.5	
	bc_h4	25.0	5.0	82.0	2.5	
	bh_c1	10.0	76.0	597.6	2.5	
	bh_c2	10.0	76.0	478.6	2.5	
	bh_c3	76.0	98.0	325.5	2.5	
	bh_c4	98.0	98.0	455.1	0.8	
	bh_h1	98.0	98.0	455.1	1.7	
	bh_h2	98.0	20.0	60.6	2.5	
	bh_h3	98.0	9.0	1247.3	2.5	
wash	bw_c1	37.3	37.3	9.7	2.5	
	bw_c2	50.0	50.0	11.8	2.5	
	bw_c3	73.0	73.0	232.6	2.5	
	bw_c4	55.1	55.1	3.0	2.5	
	bw_h1	37.7	37.7	8.8	2.5	
	bw_h2	25.6	25.6	7.3	2.5	
	bw_h3	19.6	19.6	4.1	2.5	
past4	bp4_c1	12.5	12.5	14.3	2.5	
	bp4_c2	22.0	22.0	36.9	2.5	
	bp4_c3	32.0	32.0	47.1	2.5	
	bp4_c4	41.5	41.5	49.5	2.5	
	bp4_c5	57.0	57.0	71.1	2.5	
	bp4_c6	62.0	62.0	134.3	2.5	
	bp4_h1	54.5	54.5	22.3	2.5	
	bp4_h2	46.5	46.5	33.4	2.5	
	bp4_h3	37.5	37.5	41.1	2.5	
	bp4_h4	29.5	29.5	35.5	2.5	
past12	bp12_c1	12.5	12.5	36.8	2.5	
	bp12_c2	22.0	22.0	70.2	2.5	
	bp12_c3	32.0	32.0	86.5	2.5	
	bp12_c4	41.5	41.5	90.7	2.5	
	bp12_c5	57.0	57.0	130.2	2.5	
	bp12_c6	62.0	62.0	321.8	2.5	
	bp12_h1	54.5	54.5	36.9	2.5	
	bp12_h2	46.5	46.5	57.4	2.5	
	bp12_h3	37.5	37.5	72.0	2.5	
	bp12_h4	35.0	35.0	62.0	2.5	
past3	bp3_c1	5.0	73.0	334.3	2.5	
	bp3_h1	73.0	8.0	319.5	2.5	
CIP	make_up	10.0	95.0	250.5	2.5	
	soap	20.0	95.0	6.4	2.5	
	distr	83.0	95.0	145.8	2.5	
	discharge	83.0	15.0	206.6	2.5	

- The pinch temperature (T^*_{pinch} =12.5°C) corresponds to the ambient conditions. It allows for integrating the hot stream of the refrigeration system as a heat source for the process.

- Provided that a heat pumping system is used to satisfy the needs at medium temperature,

Figure B.21: Brewery process composite curves

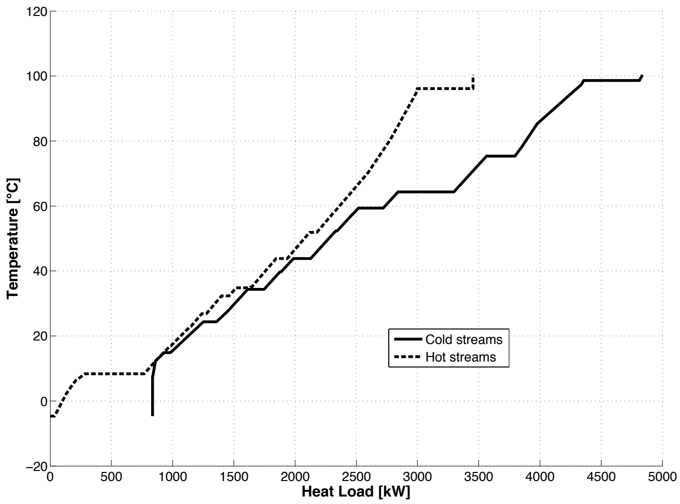
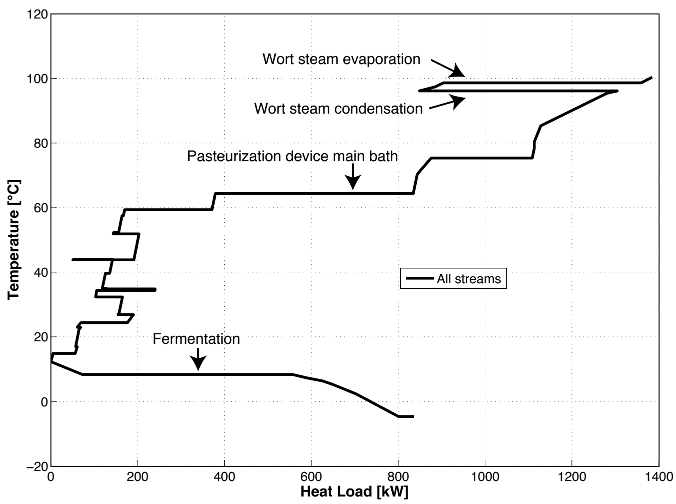


Figure B.22: Brewery process grand composite curve



an MVR system can be used to recover the condensation of wort steam at high temperature. This would enable lower temperature heating requirements to be satisfied by the cooling water of a co-generation engine. Thus, the size of the MVR system will be related with the heat delivered by the co-generation system.

- A refrigeration utility with multiple levels of evaporation represents an appropriate solution in order to minimize the exergy losses below the pinch temperature.

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Curriculum vitae

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