

Thermodynamics and sustainable development

The use of exergy analysis and the reduction of irreversibility

R. L. Cornelissen

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THERMODYNAMICS AND SUSTAINABLE DEVELOPMENT

The use of exergy analysis and the reduction of irreversibility

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Summary

Introduction

The term exergy was introduced by Rant in the fifties. The concepts on which it is based, the first and second law of thermodynamics, were already established in the 19-th century. The first thermodynamic analyses, based on the first and second law, were performed at the end of the 19-th and the beginning of the 20-th century. The explicit use of exergy analysis for the assessment of different types of systems has been developed quite slowly. Nowadays, there are still a few researchers and engineers, who use exergy analysis. Many of them limit themselves to the first law only.

A brief overview of the different components of exergy and how to calculate these are given. The difference between energy analysis and exergy analysis is presented and it is shown that for carrying out an exergy analysis an energy analysis has to be performed first. The value of an exergy analysis in addition to an energy analysis has been discussed. The status of the use of exergy analysis is shown in a literature review on exergy analysis from 1985 to 1994.

It is the objective of this thesis to demonstrate exergy analysis as a powerful instrument to obtain sustainable development. An important aspect of sustainable development is the minimisation of irreversibilities caused by the use of non-renewables. In order to limit the scope of this thesis sustainable development associated with production is considered only. The demonstration cases from practice are selected on four levels: components from process units, process units, complete production processes and, finally, products from production processes. This approach requires life cycle analysis aspects to be included in the exergy analysis.

Results

When performing an exergy analysis a measure for the degree of thermodynamic perfection of the different processes is desired. This measure is often called efficiency. However, the use of efficiencies causes confusion, when not properly defined. In this thesis three different definitions of exergetic efficiencies are distinguished, the simple efficiency, the rational efficiency and the efficiency with transiting exergy. The simple efficiency and the efficiency with transiting exergy give both answers without taking into account the function of the system. The rational efficiency is more practical, because first the function of the system has to be defined. An example of an air separation unit is used, where the three definitions are compared. The rational efficiency is used further on in this thesis.

A comparison is made in the field of heat exchangers between the rational efficiency, based on the first and second law, and the effectiveness, based on the first law only, of which the latter one is widely used as a performance criterion. It is shown here that there is no linear relation between the effectiveness and rational efficiency and that the rational efficiency should be used from the viewpoint of fuel saving.

Exergy analyses have been performed on process units dealing with distillation. The overall rational efficiency of the crude oil distillation plant, including the heater and the heat exchanger network, is 0.052. The biggest causes of irreversibility are the use of high-quality fuel for the production of middle-quality heat and the dissipation of low quality energy in the environment. The irreversibility could be reduced by 16% using this low quality energy. A significant cause of irreversibility is the distillation process itself, as can be seen from the exergy destruction of about one-fourth of the total in the columns, excluding the reboilers. It is proposed to replace the heater by a combined heat and power plant and to use the low temperature waste heat in a district heating system. The overall rational efficiency of the cryogenic air separation plant is 0.28. The possibilities of reducing exergy losses in this type of plant are more limited. It is shown that in the specific plant analysed improving the compressors yield a considerable exergy saving. Including other suggested improvements the exergy loss can be reduced by 25%. However, the cryogenic air separation plant itself is well designed from an exergetic point of view and for a further improvement of the rational efficiency of air separation alternative processes have to be used or developed.

To determine the thermodynamic perfection of a system, not only processes that occur within the system will be taken into account, but also all kinds of interaction between energy and material flows outside the system's boundaries. Only then the actual performance of the system and its impact on the environment can be evaluated. The irreversibility during the complete life cycle allows to evaluate the degree of thermodynamic perfection of the production processes and to conduct the assessment of the whole process chain.

On basis of the life cycle approach in combination with exergy analysis a method has been developed, which is called the Exergetic Life Cycle Analysis (ELCA). This method is an extension to the already established method of Life Cycle Assessment (LCA). A comparison is made with the LCA. It is explained that the ELCA should be a part of every LCA, because life cycle irreversibility is the most appropriate parameter for the depletion of natural resources. The ELCA has been extended to the Zero-ELCA to include the abatement exergy of emissions. This is the exergy use due to the abatement and the re-use or environmental friendly disposal of the emissions. So all environmental problems associated with emissions can also be taken into account and not only depletion of natural resources.

A textile washing machine has been optimised by making use of the ELCA for the detergent quantity and washing temperature for three different water heating systems, namely the conventional electrical heating, gas fired heating and a district heating system.

A relation has been established to calculate and minimise the life cycle irreversibility with the amount of detergent used and the washing temperature for the three different heating systems. It is shown that the conventional electrical heating spiral is a very inefficient heating system in terms of life cycle irreversibility. It is, therefore, recommended to replace this system by a system based on gas fired heating or, if possible, on a district heating system. This will reduce the life cycle irreversibility by 1/3 and by more than 1/2, respectively.

The ELCA can also be used to assess different types of production processes as is shown in the case of systems for heat distribution, where the heat is provided by a steam and gas turbine. These systems are compared to the conventional system of heating by high efficiency boilers on natural gas. It is shown that the irreversibility due to the building of the district heating system is negligible. The exergetic pay back period, defined as the additional exergy investment divided by the exergy saving, of the more exergy-efficient systems is between one and sixteen months. The main cause of the difference in irreversibility between the alternatives is due to the heat transfer to the environment. The exergy savings are ranging from 1/4 for widespread distributions systems to 1/2 for the dense distributions systems. Also a financial analysis is presented. The results of the ELCA and the financial analysis are combined to get the most cost effective investment for exergy saving.

In addition, a single component, a heat exchanger in a district heating system, has been analysed according to the principles of the ELCA. It leads to the optimal design for a heat exchanger from a thermodynamic point of view. The trade off between irreversibilities due to heat transfer, pressure drops and material use is shown. The different components of the life cycle irreversibility are more uniformly distributed when there are fewer restrictions on the design parameters. The minimisation of the life cycle irreversibility can not take place without taking into account the system parameters, because these influence the geometrical parameters. Especially, the tube length of the optimised heat exchanger is sensitive to system changes, while the inner diameters are insensitive to these. The optimised heat exchanger greatly reduces the life cycle irreversibility, when compared with the present system.

As an example of consumer products the ELCA of the porcelain mug and the disposable polystyrene cup is included. This is compared with the environmentally orientated LCA, which is extended with the Eco-indicator 95 to express the environmental problems in one indicator. It is shown that ELCA is a valuable addition to the LCA by including the environmental problem of depletion of natural resources. The ELCA shows that the disposable polystyrene cups make two times greater use of the exergy reservoir of natural resources than the porcelain mug. The Zero-ELCA demonstrates that the harmful emissions can be reduced by 80 to 95% by a small increase in the use of natural resources. The Zero-ELCA is shown to be a valuable approach because it gives a good indication of the effort in terms of natural resources and process development needed to prevent harmful emissions.

Conclusions

One of the keystones for obtaining sustainable development is the use of exergy analysis. Energy can never be lost as stated in the first law of thermodynamics. Exergy can be lost and this loss, called irreversibility, created during the use of non-renewables has to be minimised to obtain sustainable development. In this thesis it is shown on different levels how this can be done. At the same time it is shown that all environmental effects associated with emissions and the environmental effect of depletion can be expressed in terms of one indicator, which is based on physical principles.

Samenvatting

Inleiding

De term exergie wordt door Rant in the vijftiger jaren geïntroduceerd, maar de ideeën waarop deze term is gebaseerd, de eerste en tweede hoofdwet van de thermodynamica zijn al sinds de negentiende eeuw bekend. De eerste thermodynamische analyses, gebaseerd op de eerste en tweede hoofdwet, zijn al uitgevoerd aan de einde van de vorige eeuw en het begin van deze eeuw. Het gebruik van exergie voor het beoordelen van verschillende systemen is pas langzaam op gang gekomen. Een klein aantal onderzoekers en ingenieurs gebruikt exergie-analyse. De meesten beperken zich echter tot de eerste hoofdwet van de thermodynamica.

Een kort overzicht van de verschillende componenten van exergie en hoe deze te berekenen is gegeven. Het verschil tussen energie- en exergie-analyse wordt uitgelegd. Tevens wordt aangegeven dat voor het uitvoeren van een exergie-analyse eerst een energie-analyse moet worden gemaakt. De meerwaarde van een exergie-analyse boven een energie-analyse wordt bediscussieerd. Een literatuurstudie van 1985 tot 1994 is uitgevoerd om het gebruik en de toepassingen van exergie-analyse in kaart te brengen.

Het doel van dit proefschrift is om aan te tonen dat exergie-analyse een krachtig instrument is voor het bereiken van een duurzame samenleving. Een belangrijk aspect van duurzame ontwikkeling is de vermindering van irreversibiliteit, die veroorzaakt wordt door het gebruik van niet vernieuwbare brandstoffen. Om de vraagstelling te beperken is in dit proefschrift alleen gekeken naar duurzame ontwikkeling in relatie met productieprocessen. Een aantal praktijk voorbeelden is geselecteerd op vier niveaus: componenten van proceseenheden, proceseenheden, productieprocessen en tot slot de productieroute van producten.

Resultaten

Wanneer een exergie-analyse wordt uitgevoerd is een maat voor de thermodynamische perfectie van de verschillende processen gewenst. Voor deze maat wordt vaak het rendement genomen. Maar het gebruik van rendementen kan tot verwarring leiden als deze niet goed gedefinieerd zijn. In dit proefschrift zijn drie verschillende definities van exergetische rendement met elkaar vergeleken: het eenvoudige rendement, het functionele rendement¹ en het rendement met transit-exergie. Het eenvoudige rendement en het rendement met transit exergie geven beide een antwoord zonder naar de functie van het systeem te kijken. Het functionele rendement is praktischer in de zin dat eerst de functie van het systeem moet worden gedefinieerd. De drie rendementen zijn vergeleken voor een luchtscheidingseenheid. In het vervolg van het proefschrift is het functionele

¹ In het Engels rational efficiency.

rendement gebruikt. Voor warmtewisselaars is een vergelijking gemaakt tussen het functionele rendement, gebaseerd op de eerste en tweede hoofdwet, en de effectiviteit, gebaseerd op de eerste hoofdwet, die nog veel gebruikt wordt als maatstaf voor de prestaties. Er wordt aangetoond dat er geen eenduidige relatie is tussen de effectiviteit en het functionele rendement en dat vanuit het oogpunt van brandstofbesparing het functionele rendement gebruikt moet worden.

Er zijn exergie-analyses uitgevoerd voor proceseenheden van een destillatie proces. Het totale functionele rendement van de ruwe aardolie destillatie-eenheid, inclusief de brander en het warmtewisselaarsnetwerk, is 0.052. De belangrijkste oorzaken van het exergieverlies zijn het gebruik van hoogwaardige brandstoffen voor de productie van middelwaardige kwaliteit warmte en de dissipatie van lage kwaliteit warmte. Het exergieverlies kan met 16% gereduceerd worden wanneer de lage kwaliteit warmte nuttig gebruikt wordt. Een belangrijk exergieverlies vindt plaats in het destillatieproces zelf, wat blijkt uit het exergieverlies van 1/4 van het totaal in de destillatiekolommen, exclusief de reboiler. Er wordt voorgesteld om de brander te vervangen door een warmtekracht-eenheid en de lage kwaliteit warmte te gebruiken in een afstandverwarmingssysteem. Het totale functionele rendement is 0.28 voor de cryogene luchtscheidingsfabriek. De mogelijkheden voor het reduceren van exergieverliezen van deze fabriek zijn beperkter. Er wordt aangetoond dat in de geanalyseerde fabriek het verbeteren van de compressoren een significante exergiebesparing oplevert. Inclusief de andere voorgestelde verbeteringen geeft dit een exergiebesparing van 25%. Er kan geconcludeerd worden dat de cryogene luchtscheidingsfabriek goed ontworpen is vanuit een exergetisch oogpunt en dat voor verdergaande verbeteringen van het functioneel rendement alternatieve processen moeten worden gebruikt of ontwikkeld.

Om de thermodynamische perfectie van een systeem te bepalen moeten niet alleen de processen die binnen het systeem plaats vinden in beschouwing worden genomen, maar ook de interacties met energie- en materiaalstromen buiten het systeem. Alleen dan kan het werkelijke functioneren van het systeem en zijn invloed op het milieu worden geëvalueerd. De irreversibiliteit gedurende de gehele levenscyclus dient te worden bepaald om de mate van thermodynamische perfectie van het productieproces te evalueren en het gedrag van de gehele procesketen te beoordelen.

Op basis van de levenscyclus benadering in combinatie met exergie-analyse is een methode ontwikkeld, welke de Exergetische Levens Cyclus Analyse (ELCA) wordt genoemd. Deze methode is een uitbreiding van de al eerder ontwikkelde methode van de Levens Cyclus Analyse (LCA). Een wordt een vergelijking gemaakt met de LCA. Verder wordt uitgelegd dat de ELCA deel moet uitmaken van elke LCA, omdat de levenscyclus-irreversibiliteit de meest geschikte maat is voor uitputting van natuurlijke hulpbronnen. De ELCA is uitgebreid met de Zero-ELCA, waarin de verwijderingsexergieën van de emissies zijn meegenomen. Dit is het exergiegebruik wat ontstaat door verwijdering en hergebruik of milieuvriendelijke opslag van de emissies. Op deze wijze kunnen ook alle milieu-effecten gerelateerd aan emissies in rekening worden gebracht en is de analyse niet alleen beperkt tot uitputting van de natuurlijke hulpstoffen.

Er wordt aangetoond dat een wasmachine kan worden geoptimaliseerd door gebruik te maken van de ELCA voor het wasmiddelengebruik en de wastemperatuur voor drie verschillende waterverwarmings-systemen, namelijk conventionele elektrische verwarming, gas-gestookte verwarming en afstandsverwarming. Er is een relatie bepaald voor het berekenen en minimaliseren van de levenscyclus-irreversibiliteit aan de hand van het wasmiddelengebruik en de wastemperatuur voor de drie verschillende verwarmingssystemen. Verder is aangetoond dat de huidige elektrische spiraal een erg inefficiënt systeem is in termen van levenscyclus-irreversibiliteit. Daarom wordt aanbevolen om dit systeem te vervangen door een gas-gestookte verwarming of, indien mogelijk, afstandsverwarming. Dit zal de levenscyclus-irreversibiliteit reduceren met resp. een derde tot meer dan de helft.

De ELCA kan ook worden gebruikt voor het beoordelen van verschillende typen productieprocessen. Dit is gebeurd voor warmtedistributie waarvan de warmte wordt verschaft door een stoom- en gasturbine (STEG). Deze systemen zijn vergeleken met de huidige HR-ketel. Aangetoond is dat de irreversibiliteit t.g.v. het bouwen van een warmtedistributiesysteem verwaarloosd kan worden. De exergetische terugverdientijd van de exergie-efficiëntere systemen, gedefinieerd als de additionele exergie-investering gedeeld door de exergiebesparing, is tussen de één en zestien maanden. De belangrijkste oorzaak van het verschil in irreversibiliteit tussen de alternatieven is te wijden aan warmtelekken naar de omgeving. De exergiebesparing varieert van 1/4 voor de wijdvertakte systemen tot 1/2 voor de dichtvertakte systemen. Een financiële analyse is toegevoegd. De resultaten van de ELCA en de financiële analyse zijn gecombineerd om de meest kosteneffectieve investering voor exergiebesparing te bepalen.

In aanvulling hierop is een component in een afstand-verwarmingssysteem, te weten de warmtewisselaar, geanalyseerd in overeenstemming met de principes van een ELCA. Dit leidt tot een optimaal ontwerp voor de warmtewisselaar vanuit een thermodynamisch gezichtspunt. De uitwisseling tussen irreversibiliteiten door warmte-overdracht, drukverlies en materiaalgebruik wordt zichtbaar gemaakt. De verschillende componenten van de levenscyclus-irreversibiliteit zijn uniformer verdeeld wanneer er meer vrijheid is voor de ontwerp-parameters. De minimalisatie van de levenscyclus-irreversibiliteit kan niet plaatsvinden zonder rekening te houden met de systeem-parameters. Vooral de buislengte van de geoptimaliseerde warmtewisselaar is erg gevoelig voor systeemveranderingen, terwijl de binnendiameters daarvoor ongevoelig zijn. De geoptimaliseerde warmtewisselaar reduceert de levenscyclus-irreversibiliteit voor een groot deel in vergelijking met het huidige systeem.

Om de toepassing op consumenten producten te demonstreren is een ELCA van de porseleinen mok en het wegwerpbekertje uitgevoerd. Deze is vergeleken met de milieugeoriënteerde LCA, die is uitgebreid met de Eco-indicator-95-methode om alle milieu-effecten in één indicator tot uitdrukking te brengen. Aangetoond is dat de ELCA een waardevolle aanvulling is op de LCA voor het meenemen van het milieueffect van uitputting van natuurlijke hulpstoffen. De ELCA laat zien dat het wegwerpbekertje twee

keer zoveel gebruik maakt van de exergie-reservoirs van natuurlijke hulpstoffen dan de porseleinen mok. De Zero-ELCA toont aan dat de schadelijke emissies met 80 tot 95% kunnen worden gereduceerd bij een kleine toename in het gebruik van natuurlijke hulpbronnen. De Zero-ELCA is een waardevolle benadering omdat hij een goede indicatie geeft van de inspanningen die nodig zijn in termen van natuurlijke hulpbronnen en noodzakelijke procesontwikkeling voor het voorkomen van schadelijke emissies.

Conclusies

Een van de middelen voor het bereiken van duurzame ontwikkeling is het gebruik van exergie. Energie kan nooit verloren gaan volgens de eerste hoofdwet van de thermodynamica. Exergie kan wel verloren gaan en dit verlies, de irreversibiliteit, veroorzaakt tijdens het gebruik van niet vervangbare natuurlijke hulpbronnen moet worden geminimaliseerd om een duurzame ontwikkeling te realiseren. In dit proefschrift wordt voor verschillende niveaus aangetoond hoe dit gedaan kan worden. Tevens wordt gedemonstreerd hoe uitputting van natuurlijke hulpbronnen en alle milieueffecten gerelateerd aan emissies kunnen worden uitgedrukt in een waarde die is gebaseerd op fysische principes.

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Introduction

1.1 Thermodynamics and sustainable development

The basis of thermodynamics is stated in the first and second law. The first law describes the conservation of energy, while the second law is used to describe the quality of energy and material. This thesis focuses on the combination of the two laws, which is described in the concept of exergy analysis. The term exergy was introduced by Rant in 1953. The concepts of the first and second law were established by Von Mayer and Carnot, respectively, in the first half of the 19-th century. Important contributions to the thermodynamics were given by Gibbs in the second half of the 19-th century. The basis of the second law analysis was made by Gouy in 1889 and Stodola in 1898 by the law, which has been named after them (Szargut, 1980). A second law analysis has already been performed by Stodola in 1905 (Baehr, 1965 and Motz, 1989) and by Jouguet in 1907 (Jouguet, 1907). In 1932 Keenan (1932) defined the concept of physical exergy, at that time called availability, and used it for the analysis of a steam turbine. The concept of chemical exergy and its associated reference states has been introduced by Szargut (1986). The regular use of exergy analysis starts in the second half of this century.

In the seventies of this century the awareness awoke that the extensive use of natural resources would lead to the depletion of these resources. Later on, the pollution due to their extensive use was seen as a major problem for mankind. As a solution to this problem the concept of sustainable development was developed. This is defined by a United Nation committee as “a development that ensures the needs of the present society, without compromising the ability of future generations to meet their own needs” (Brundtland et al., 1987).

The use of exergy analysis in relation to the depletion of natural resources has already been proposed by the International Federation of Institutes for Advanced Study in the Energy Analysis Workshop on Methodology and Conventions in 1974 (Slessor, 1974). The study can be seen as one of the first in this field. However, its promotion of the use of exergy analysis was not very successful. It is followed by a limited number of people, of which Szargut is the most well known. He introduced several interesting concepts,

which will be discussed in this thesis. This thesis can be considered as a continuation of his research.

1.2 The method of exergy analysis

In this paragraph the method of exergy analysis used in this thesis is presented. In paragraph 1.5 the scope of the thesis is explained. The concept of exergy is extensively discussed in the books of Kotas (1995) and Szargut et al. (1988). A brief summary is given here. Exergy is defined as the maximum work potential of a material or of a form of energy in relation to its environment. This work potential can be obtained by reversible processes. However, in reality there are only irreversible processes. For practical reasons a reference environment has been defined for the environment. The reference environment is considered to be so large, that its parameters are not affected by interaction with the system under consideration. In this thesis the reference system as stated in Szargut et al. (1988) has been used with a reference temperature (T_0) of 298.15 K and a reference pressure (P_0) of 1 atm, unless otherwise stated. This system is similar to the one used by Kotas (1995).

To perform an exergy analysis first closed material and energy balances have to be made. Of course, a closed mass balance includes a closed atomic balance in case of non-nuclear reactions. The following three ways of exergy transfer are distinguished in this thesis: exergy transfer with work and with heat interaction and exergy associated with mass flow. Other components of exergy transfer are neglected, like potential and kinetic exergy or excluded, like nuclear effects.

1.2.1 Exergy transfer with work interaction

The exergy transfer with work interaction is associated with work transfer rate or shaft power. Because exergy is defined as the maximum work potential, it is equivalent to exergy in every respect.

1.2.2 Exergy transfer with heat interaction

The exergy transfer rate (\dot{E}) connected with the heat transfer rate (\dot{Q}_i) can be calculated by the following formula:

$$\dot{E} = \int_A \left(\frac{T - T_0}{T} \right) \dot{Q}_i dA \quad (1.1)$$

where A is the heat transfer area, T_0 is the temperature of the environment, T is the temperature at which the heat transfer takes places. When there is a uniform temperature distribution, $\int \dot{Q}_i dA = \dot{Q}_A$ and formula 1.1 becomes:

$$\dot{E} = \dot{Q}_A \left(1 - \frac{T_0}{T}\right) \quad (1.2)$$

1.2.3 Exergy transfer associated with material streams

Physical exergy

Physical exergy is the work obtainable by taking the substance through reversible processes from its initial state temperature T and pressure P , to the state determined by the temperature T_0 and the pressure P_0 of the environment. It can be calculated with

$$E_{\text{ph}} = H - H_0 - T_0(S - S_0) \quad (1.3)$$

where H is the enthalpy and S the entropy. The physical exergy can be split into a thermal and a pressure component, also called mechanical component.

By using the perfect gas laws in (1.3) and assuming a constant specific isobaric heat capacity (c_p) follows:

$$e_{\text{ph}} = c_p \left\{ (T - T_0) - T_0 \ln \frac{T}{T_0} \right\} + RT_0 \ln \frac{P}{P_0} \quad (1.4)$$

For solids and liquids the following formula can be obtained when assuming a constant specific heat (c):

$$e_{\text{ph}} = c \left\{ (T - T_0) - T_0 \ln \left(\frac{T}{T_0}\right) \right\} - v_m (P - P_0) \quad (1.5)$$

where v_m is the specific volume, determined at temperature T_0 .

Chemical exergy

Chemical exergy is equal to the maximum amount of work obtainable when the substance under consideration is brought from the environmental state, defined by the parameters T_0 and P_0 , to the reference state by processes involving heat transfer and exchange of substances only with the environment.

For the calculation of the chemical exergy of the reference gases, which form together the reference atmosphere, the work for getting the components at the standard pressure from the partial pressure of the reference state has to be determined. This is done by the following formula:

$$e_{ch} = RT_0 \ln P_0/P_{00} \quad (1.6)$$

where P_{00} is the partial pressure of the component in the reference state.

For non-reference pure components the chemical exergy can be determined by the following formula:

$$e_{ch} = -\Delta G_0 - \sum_i x_i e_{ch,i}^{in} + \sum_i x_i e_{ch,i}^{out} \quad (1.7)$$

with ΔG_0 is the Gibbs function of formation. ΔG_0 can be calculated from the general relation:

$$\Delta G_0 = \sum_{prod} v_k \Delta g_k - \sum_{react} v_j \Delta g_j \quad (1.8)$$

where v_k , v_j and Δg_k , Δg_j are the stoichiometric coefficients and the molar Gibbs functions of formation, respectively, j is the j -th co-reactant and k is the k -th product. The chemical exergy for mixtures can be calculated as follows:

$$e_{ch, mix.} = \sum_i x_i e_{ch,i} + RT_0 \sum_i x_i \ln \gamma_i x_i \quad (1.9)$$

where x_i is the mol fraction of the i -th component, R is the molar gas constant and γ_i is the activity coefficient. For ideal solutions the activity coefficient is equal to one.

For many fuels the chemical structure is unknown. To overcome this problem the chemical exergy for this fuels can be estimated on the basis of the net combustion value (NCV). The relation between the NCV and the chemical exergy is:

$$e_{ch} = \phi \cdot NCV \quad (1.10)$$

where ϕ can be calculated with formulas based on the atomic composition. For different fuel oils and petrol ϕ is between 1.04 and 1.08.

1.2.4 Irreversibility

The irreversibility, also called exergy destruction or exergy loss, is calculated by setting up the exergy balance and taken the difference between all incoming and outgoing exergy flows or in formula form

$$I = \sum_{in} E_i - \sum_{out} E_j \quad (1.11)$$

Another way of calculating the irreversibility can be done by the Gouy-Stodola formula, in which the entropy increase is multiplied by the environmental temperature, in formula form

$$I = T_0 \cdot \left(\sum_{out} S_j - \sum_{in} S_i \right) = T_0 \cdot \Delta S \quad (1.12)$$

1.3 Energy analysis and exergy analysis

In an energy analysis, based on the first law of thermodynamics, all forms of energy are considered to be equivalent. The loss of quality of energy is not taken into account. For example, the change of the quality of thermal energy as it is transferred from a higher to a lower temperature can not be demonstrated in an energy analysis. It shows the energy flow to be continuous. An exergy analysis, based on the first and second law of thermodynamics, shows the thermodynamic imperfection of a process, including all quality losses of materials and energy, including the one just described.

An energy balance is always closed as stated in the first law of thermodynamics. There can never be an energy loss, only energy transfer to the environment in which case it is useless. For pinpointing and quantifying the irreversibilities an exergy analysis has to be performed.

1.4 Literature review on exergy analysis from 1985 to 1994

A literature review on exergy analysis for the period 1985 to 1994 has been performed to get more insight in this research field. A database of 825 publications in the field of exergy analysis has been obtained. The source of this database is the DIALOG Database Energy Science & Technology, which contains 2.5 millions abstracts in the field of energy research and related topics. Most articles deal with energy conversion. In this subject the most attention has been devoted to power plants, heat sources and solar energy. A limited amount of articles has been published on processes or process plants. There are more articles published on the use of solar exergy than exergy analysis of processes or process plants. It can be concluded that the use of exergy analysis in the process industry is as yet limited (Cornelissen, 1994). A copy of the database on floppy discs can be obtained at the University of Twente. More information of how to use this database can be found in Appendix 1.

1.5 Scope of the thesis

The main aim of this thesis is to demonstrate exergy analysis as a powerful instrument to obtain sustainable development. In order to limit the scope of the thesis, this is demonstrated on different levels as listed in Table 1.1. Existing methods are adjusted and further developed to perform this type of analysis.

Table 1.1 The different analysed systems and the associated examples

System level	Example
component	heat exchanger
process unit	crude oil distillation plant, air separation plant
production system	textile washing process, district heating system
products	disposable cup, porcelain mug

For comparing different components, process units or even complete production systems a clear yard stick is required. In chapter two the different coefficient of performance, based on exergy, are compared. A conclusion is drawn as to which coefficient of performance is to be preferred.

In chapter three it is shown on a component level, in this case the heat exchanger, that the exergetic efficiency is preferable to the effectiveness, which is based on the first law only. Chapter four shows two conventional exergy analyses of process units: the crude oil distillation unit and the air separation unit.

To determine the thermodynamic perfection of a system, not only processes that occur within the system will be taken into account, but also all kinds of interaction between energy and material flows outside the system's boundaries. The irreversibility during the complete life cycle allows to evaluate the degree of thermodynamic imperfection of the production system. In chapter five the life cycle irreversibility is shown to be the measure for depletion of natural resources. So, the concept of Exergetic Life Cycle Analysis (ELCA) is introduced. It is discussed that it is a valuable addition to the environmentally orientated LCA. An extension to the ELCA, the Zero-ELCA, is developed to include environmental effects associated with emissions and it can in particular cases replace the LCA.

The ELCA is performed on two product systems. In chapter 7 the life cycle irreversibility of the product system is minimised for various alternatives. In chapter 8 an ELCA has been performed to compare the life cycle irreversibility of the various alternatives. However, sometimes only a component of a product system has to be optimised. It is demonstrated how this can be done by using the concept of ELCA.

Finally in chapter 9, two consumer products are compared with an LCA, including the Eco-indicator 95, and an ELCA. With the Zero-ELCA is demonstrated that with a

relative small increase of the life cycle irreversibility the emissions, and off course the problems associated with it, can be greatly reduced.

1.6 Publications underlying this thesis

The thesis is a compilation of the following publications. However, the chapters referring to the listed publications are not a copy. Changes have taken place to fit them into the underlying work and to take into account comments from participants in the conferences and from reviewers from ASME and other associations.

Table 1.2 Publications underlying this thesis

Publications	Related to chapter
Cornelissen, R.L., 1994, <i>Bibliography on Exergy Analysis and Related Techniques 1985-1994</i> , University of Twente.	1
Cornelissen, R.L., Hirs, G.G. and Kotas, T.J., 1995, <i>Different Definitions of exergetic efficiencies</i> , Proceedings of JETC IV, Nancy, France, pp. 335-344	2
Cornelissen, R.L., Hirs, G.G., Smeding, S.F. and Raas, J.L., 1997, <i>Performance criteria for a heat exchanger</i> , Achema'97, Frankfurt am Main.	3
Cornelissen, R.L., Hirs, G.G., Smeding, S.F. and Raas, J.L., 1997, <i>Performance criteria for a heat exchanger</i> , TAIES'97, Beijing, pp. 163-167	3
Cornelissen, R.L., Hirs, G.G. and Kotas, T.J., 1995, <i>An exergy analysis of an oil distillation process</i> , Proceedings of the Second Law Analysis of Energy Systems: towards the 21-st century, Ed. Sciubba and Moran, Rome, Italy, pp. 417 - 429	4
Cornelissen, R.L. and Hirs, G.G., 1997, <i>Exergy Analysis of Cryogenic Air Separation</i> , FLOWERS'97, Florence, pp. 931-937	4
Moze, E., Cornelissen, R.L., Hirs, G.G. and Boom, R.M., <i>Exergy Analysis of the Conventional Textile Washing Process</i> , FLOWERS'97, Florence, pp. 939 -948	6
Cornelissen, R.L., Hirs, G.G., Lie, A.B.K. and Steenderen, P., 1996, <i>Exergy-Economics of a District Heating System</i> , ASME AES-Vol. 36, Atlanta, USA, pp. 221-226	7
Cornelissen, R.L. and Hirs, G.G., 1996, <i>Exergetic Optimisation of a Heat Exchanger</i> , Proceedings of ECOS'96, Stockholm, Sweden, pp. 41-48	8
Cornelissen, R.L. and Hirs, G.G., 1997, <i>Exergetic Optimisation of a Heat Exchanger</i> , Journal of energy conversion and management, Vol. 38, No 15-17, pp. 1567-1576	8

1.7 Nomenclature

- A = heat transfer area, [m²]
c = specific heat capacity, [J/kgK]
c_p = specific isobaric heat capacity, [J/(kg·K)]

E, e = exergy, specific exergy, [J, J/kg]
 \dot{E} = exergy rate, [J/s]
 G, g = Gibbs function of formation, specific Gibbs function of formation, [J, J/kg]
 H = enthalpy, [J]
 I = irreversibility, [J]
 P = pressure, [Pa]
 Q = heat, [J]
 R = molar gas constant, [J/(mol·K)]
 S = entropy, [J/K]
 T = temperature, [K]
 v_m = specific volume, [m³/kg]
 x_i = mol fraction of the i-th component

Greek

ϕ = ratio of chemical exergy to net combustion value

Superscripts

in = incoming

out = outgoing

Subscripts

ch = chemical

j = j-th co-reactant

k = k-th product

ph = physical

0 = environmental

00 = reference state

1.8 References

Baehr, H.D., 1965, *Definition und Berechnung von Exergie und Anergie*, Brennst.-Wärme-Kraft 17, nr. 1, pp. 1-6.

Cornelissen, R.L., 1994, *Bibliography on Exergy Analysis and Related Techniques 1985-1994*, University of Twente.

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Different Definitions of Exergetic Efficiencies

Abstract

Three definitions of exergetic efficiencies for steady state processes are presented in this chapter. The first definition is the ratio of the total outgoing exergy flow to the total incoming exergy flow. The second form of exergetic efficiency, called the rational efficiency, is defined by Kotas (1995) as a ratio of the desired exergy output to the exergy used. The last form of exergetic efficiency was introduced by Kostenko and has been further developed by Brodyansky, Sorin and Le Goff (1994). This form is the ratio of the total outgoing exergy flow minus the untransformed components of exergy to the total incoming exergy flow minus the untransformed components of exergy. The three forms of efficiencies have been applied on the example of air separation. The advantages and disadvantages of the different forms of efficiency are discussed. The rational efficiency is used further on in this thesis.

2.1 Introduction

A yardstick is required to compare the performance of different process plants or units. For this purpose it has been decided to use criteria of performance based on exergy analysis, known generally as exergetic efficiencies. A yardstick of this type can be used to compare the performance of different types of production processes with the same form of output and can help in selecting the one with the least exergy destruction. An exergetic efficiency can also be used to compare the degree of thermodynamic perfection of different processes in a given plant.

In addition to the calculation of exergetic efficiencies of plants and plant components, it is generally helpful and instructive to calculate the irreversibility rates which take place in them.

2.2 Different ways of formulating exergetic efficiency as proposed in literature

This chapter is restricted to exergetic efficiencies of steady-state processes in open systems. For a proper use of the defined efficiencies, the boundaries of the system under investigation have to be clearly defined and must be chosen to include all irreversibilities associated with the process under consideration.

2.2.1 Simple¹ efficiency

The most simple form of exergetic efficiency is the simple efficiency. For the formulation of this efficiency the exergy balance for the incoming and outgoing flows is set up, where I is the irreversibility.

$$\dot{E}_{in} = \dot{E}_{out} + \dot{I} \quad (2.1)$$

The simple efficiency is the ratio of the total outgoing exergy flow to the total incoming exergy flow:

$$\eta = \frac{\dot{E}_{out}}{\dot{E}_{in}} \quad (2.2)$$

This is an unambiguous definition and can be used for all process plants and units. Unfortunately, it gives a good impression of the thermodynamic perfection of a system only when *all* the components of the incoming exergy flow are transformed to other components, e.g., in the case for power stations. Where this is not the case, the untransformed components gives a false impression of the performance of the process plant or unit. The sensitivity of the simple efficiency to changes in the process plants or unit reduces with increasing quantities of these untransformed components.

2.2.2 Rational efficiency

This form of exergetic efficiency is called the rational efficiency² and is defined by Kotas (1995) as a ratio of the desired exergy output to the exergy used.

$$\psi = \frac{\dot{E}_{desired\ output}}{\dot{E}_{used}} \quad (2.3)$$

¹ There is no name available in literature for this kind of efficiency, so the name “simple efficiency” has been adopted in this thesis

² It is called exergetic efficiency by Tsatsaronis (1993)

$\dot{E}_{desired\ output}$ is the sum of all exergy transfers from the system, which must be regarded as constituting the desired output, plus any by-product, which is produced by the system. The desired output is determined by examining the function of the system. \dot{E}_{used} is the required exergy input for the process to be performed. If \dot{E}_{used} and $\dot{E}_{desired\ output}$ are correctly identified in relation to a control surface, which completely encloses all irreversibilities related to the process under consideration, then, together, they should account for all exergy transfers. In other words there should not exist any exergy transfer terms in the exergy balances which cannot be included in either \dot{E}_{used} and $\dot{E}_{desired\ output}$, for if there were such transfers then they would result in external irreversibilities, which are excluded, a priori. Thus we can write

$$\dot{E}_{used} = \dot{E}_{desired\ output} + \dot{I} \quad (2.4)$$

Equation (2.4) can be of help in checking the choice of terms for exergy used and the desired output. Using (2.4) in connection with (2.3) the following alternative form of the rational efficiency can be obtained

$$\psi = 1 - \frac{\dot{I}}{\dot{E}_{used}} \quad (2.5)$$

The rational efficiency can be applied to any system, except to purely dissipative systems, because no desired product can be defined in this case.

2.2.3 Efficiency with transiting exergy

This form of exergetic efficiency, called efficiency with transiting exergy, was introduced by Kostenko and has been developed further by Brodyansky, Sorin and Le Goff (1994). This form of efficiency is an improvement on the simple efficiency, because it subtracts the untransformed components from the incoming and outgoing streams. The efficiency with transiting exergy is defined as follows:

$$\eta_e = \frac{\dot{E}_{out} - \dot{E}_{tr}}{\dot{E}_{in} - \dot{E}_{tr}} \quad (2.6)$$

where \dot{E}_{tr} is the transiting exergy. As explained in Sorin et al. (1994) "the transiting exergy can be considered as the part of the exergy which traverses a system without taking any part in the mechanical, thermal or chemical changes which take place in the system. To some extent, the transiting exergy is analogous to an inert species traversing a chemical reactor without being involved in the chemical rearrangements taking place in the reactor (for the analogy to be totally exact, the reactor should be isothermal and isobaric). The amount of transiting exergy does not affect the thermodynamic efficiency of the process and it should be subtracted from the incoming and outgoing exergy flows

before the efficiency is computed". If (2.6) is specified for the different components of exergy this gives

$$\eta_e = \frac{\sum_q (\dot{E}_{out,q}^Q - \dot{E}_{tr,q}^Q) + \sum_w (\dot{E}_{out,w}^W - \dot{E}_{tr,w}^W) + \left[\sum_b \sum_j (\dot{E}_{out,b}^{o,j} - \dot{E}_{tr,b}^{o,j}) \right] + \sum_b (\dot{E}_{out,b}^{\Delta p, \Delta T} - \dot{E}_{tr,b}^{\Delta p, \Delta T})}{\sum_q (\dot{E}_{in,q}^Q - \dot{E}_{tr,q}^Q) + \sum_w (\dot{E}_{in,w}^W - \dot{E}_{tr,w}^W) + \left[\sum_b \sum_j (\dot{E}_{in,b}^{o,j} - \dot{E}_{tr,b}^{o,j}) \right] + \sum_b (\dot{E}_{in,b}^{\Delta p, \Delta T} - \dot{E}_{tr,b}^{\Delta p, \Delta T})} \quad (2.7)$$

where q and w are indices of heat flows and the flows of the same kind of work, respectively, j is an index of all the system's materials components and b is index of the parts of the streams of matter at the systems input and output. The notation

\sum_q means $\sum_{i=1}^q$. In Figure 2.1 the indices b and j are shown in the example of a separation process. In the input stream the components j are mixed.

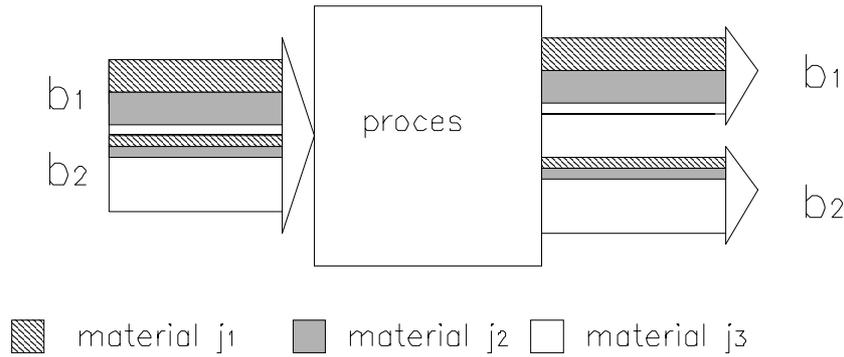


Figure 2.1 Schematic presentation of the indices b and j in the material balance of a separation process

To obtain the transiting exergy of a given energy flow (\dot{Q}, \dot{W}) the minimum of the overall exergy of a given kind at the system's inlet or at the outlet has to be taken.

$$\dot{E}_{tr,q}^Q = \min[\dot{E}_{in,q}^Q; \dot{E}_{out,q}^Q] \quad (2.8)$$

$$\dot{E}_{tr,w}^W = \min[\dot{E}_{in,w}^W; \dot{E}_{out,w}^W] \quad (2.9)$$

where the notation $\min[x; y]$ means that the minimum value of either exergy flow x or y is taken, whichever is the smaller of the two.

To define the transiting exergy in every material flow it is necessary to identify the parts of the streams of matter at the inlet of the system from which the outlet streams are

formed as a result of splitting and separating or at the outlet of the system in case of mixing. The transiting chemical exergy is defined by the sum of chemical exergy of the material components that exist simultaneously in flows at the system's input and output. For the transiting chemical exergy we have:

$$\sum_b \dot{E}_{tr,b}^o = \sum_b \sum_j \dot{E}_{tr,b}^{o,j} = \sum_b \dot{M}_b \left(\sum_j \min [e_{in,b}^{o,j}; e_{out,b}^{o,j}] \right) \quad (2.10)$$

The transiting thermal and mechanical components of exergy flows are defined as

$$\dot{E}_{tr,b}^{\Delta p, \Delta T} = \dot{M}_b \min [e_{in,b}(p_{min}, T_x); e_{out,b}(p_{min}, T_x)] \quad (2.11)$$

where $x = \min$, when $T_{in}, T_{out} \geq T_0$, $x = \max$, when $T_{in}, T_{out} \leq T_0$ and $x = T_0$, when $T_{in} \leq T_0; T_{out} \geq T_0$ or $T_{in} \geq T_0; T_{out} \leq T_0$. When no split up is made of the physical component into a thermal and a pressure component, the following formula can be used:

$$\dot{E}_{tr,b}^{ph} = \min [\dot{E}_{in,b}^{ph}; \dot{E}_{out,b}^{ph}] \quad (2.12)$$

2.3 An Example: Air Separation

In the following example the application of the three different forms of efficiencies will be demonstrated on an air distillation column, which is shown in Figure 2.2 (Sorin et al., 1994).

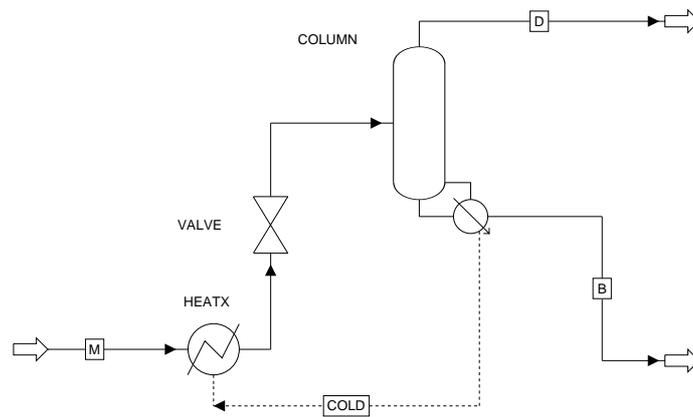


Figure 2.2 Air distillation column

The incoming mixture M is first cooled by the reboiler and cooled further by the throttling process. The incoming and outgoing flows are vapours. The properties and exergy of the streams are shown in Table 2.1. The column is assumed to operate adiabatically. For the sake of simplicity, the composition of air in the environment has been assumed to be 0.21 O₂ and 0.79 N₂ fractions by mole and the activity coefficients as described in formula 1.9 are taken all to be 1. The thermodynamic properties of stream B are equal to saturated oxygen vapour and the physical exergy of stream D is determined by assuming the vapour to behave as a mixture of perfect gasses. For stream D c_p is assumed to be 1.028 kJ/kg·K in the temperature range $T = 79$ K to $T = 298$ K at a pressure of 0.10 MPa. The environmental temperature T_o and pressure P_o are assumed to be 298 K and 0.10 MPa, respectively. The data used for the calculation of the exergy values comes from Perry (1984).

Table 2.1 Table of properties and exergy of the streams

Stream	Temp. (K)	Pressure (MPa)	Flow rate (kmol/h)	Composition		Chemical exergy (kW)	Physical exergy (kW)	Total exergy (kW)
				O ₂	N ₂			
M	139	4.00	7.00	21	79	0	23.83	23.83
B	90	0.10	1.00	99	1	1.03	1.23	2.26
D	79	0.10	6.00	10	90	0.18	8.60	8.78

2.3.1 Simple efficiency

Using (2.1) leads to the following exergy balance of the air separation column

$$\dot{E}_M = \dot{E}_B + \dot{E}_D + \dot{I} \quad (2.13)$$

The simple efficiency therefore is:

$$\eta = \frac{\dot{E}_{out}}{\dot{E}_{in}} = \frac{\dot{E}_B + \dot{E}_D}{\dot{E}_M} \quad (2.14)$$

Substituting the data from Table 2.1 in (2.14) gives $\eta = 0.463$

2.3.2 Rational efficiency

The exergy flow is divided into the thermal, mechanical and chemical component.

$$\dot{E} = \dot{E}^{\Delta T} + \dot{E}^{\Delta P} + \dot{E}^o \quad (2.15)$$

If no distinction is made between the thermal and pressure component they can be taken together as the physical component.

From the basic form of the exergy balance as given by (2.1) we get

$$\dot{E}_M = \dot{E}_B + \dot{E}_D + \dot{I} \quad (2.16)$$

By using (2.15) the exergy balance (2.16) becomes,

$$\dot{E}_M^{\Delta T} + \dot{E}_M^{\Delta P} + \dot{E}_M^{\circ} = (\dot{E}_B^{\Delta T} + \dot{E}_D^{\Delta T}) + (\dot{E}_B^{\Delta P} + \dot{E}_D^{\Delta P}) + (\dot{E}_B^{\circ} + \dot{E}_D^{\circ}) + \dot{I} \quad (2.17)$$

Case (i)

The desired output is considered to be the increase in chemical exergy.

$$\dot{E}_{desired\ output} = (\dot{E}_B^{\circ} + \dot{E}_D^{\circ}) - \dot{E}_M^{\circ} \quad (2.18)$$

Re-arranging the exergy balance (2.17) into the form given by (2.4) leads to:

$$(\dot{E}_B^{\circ} + \dot{E}_D^{\circ}) - \dot{E}_M^{\circ} = (\dot{E}_M^{\Delta T} + \dot{E}_M^{\Delta P}) - (\dot{E}_B^{\Delta T} + \dot{E}_D^{\Delta T}) - (\dot{E}_B^{\Delta P} + \dot{E}_D^{\Delta P}) - \dot{I} \quad (2.19)$$

Hence, the rational efficiency is

$$\psi = \frac{\dot{E}_{desired\ output}}{\dot{E}_{used}} = \frac{\dot{E}_B^{\circ} + \dot{E}_D^{\circ} - \dot{E}_M^{\circ}}{(\dot{E}_M^{\Delta T} + \dot{E}_M^{\Delta P}) - [(\dot{E}_B^{\Delta T} + \dot{E}_D^{\Delta T}) + (\dot{E}_B^{\Delta P} + \dot{E}_D^{\Delta P})]} \quad (2.20)$$

or, from (2.5)

$$\psi = 1 - \frac{\dot{I}}{(\dot{E}_M^{\Delta T} + \dot{E}_M^{\Delta P}) - [(\dot{E}_B^{\Delta T} + \dot{E}_D^{\Delta T}) + (\dot{E}_B^{\Delta P} + \dot{E}_D^{\Delta P})]} \quad (2.21)$$

Substituting the data from Table 2.1 in (2.20) the rational efficiency is calculated to be 0.0864.

Case (ii)

The desired output in this case is considered to be the increase in both the chemical exergy and the thermal component of exergy

$$\dot{E}_{desired\ output} = [(\dot{E}_B^{\circ} + \dot{E}_D^{\circ}) - \dot{E}_M^{\circ}] + [(\dot{E}_B^{\Delta T} + \dot{E}_D^{\Delta T}) - \dot{E}_M^{\Delta T}] \quad (2.22)$$

Re-arranging the exergy balance (2.17) according to (2.4) gives

$$\left[(\dot{E}_B^o + \dot{E}_D^o) - \dot{E}_M^o \right] + \left[(\dot{E}_B^{\Delta T} + \dot{E}_D^{\Delta T}) - \dot{E}_M^{\Delta T} \right] = \dot{E}_M^{\Delta P} - (\dot{E}_B^{\Delta P} + \dot{E}_D^{\Delta P}) - \dot{i} \quad (2.23)$$

This gives the following expression for the rational efficiency

$$\psi = \frac{\dot{E}_{desired\ output}}{\dot{E}_{used}} = \frac{\left[(\dot{E}_B^o + \dot{E}_D^o) - \dot{E}_M^o \right] + \left[(\dot{E}_B^{\Delta T} + \dot{E}_D^{\Delta T}) - \dot{E}_M^{\Delta T} \right]}{\dot{E}_M^{\Delta P} - (\dot{E}_B^{\Delta P} + \dot{E}_D^{\Delta P})} \quad (2.24)$$

or, from (2.5)

$$\psi = 1 - \frac{\dot{i}}{\dot{E}_M^{\Delta P} - (\dot{E}_B^{\Delta P} + \dot{E}_D^{\Delta P})} \quad (2.25)$$

With help of the data of Table 2.1 the rational efficiency as stated in (2.24) can be calculated. The thermal component of the exergy of the incoming mixture M has been determined by using a polynomial as function of $c_p(T)$. This gives a thermal component of the exergy of stream M of 6.55 kW. The rational efficiency as calculated from (2.24) is $4.47/17.28 = 0.271$

Of course, the question arises whether the rational efficiency defined in case (i) or the one in case (ii) should be used. This depends of the situation outside the distillation column. If the only desired product is the chemical exergy increase and the thermal exergy increase is destroyed somewhere after leaving the distillation column, the efficiency of case (i) is appropriate. However, as in most operating plants of air distillation, the thermal exergy increase of the leaving streams is used effectively, the efficiency defined in case (ii) is in agreement with these situations. An example of how to deal with this problem is shown in chapter four of this thesis and Cornelissen et al. (1995).

2.3.3 Efficiency with transiting exergy

As shown in (2.7) the transiting exergy in the different components must be calculated. Because there is no work done by the unit and no heat transfer takes place to or from the unit, the work and thermal exergy are zero.

As follows from (2.10) and Table 2.1, the transiting chemical exergy in each part of the stream is

$$\dot{E}_{tr\ M-B}^o = \dot{M}_B \left(\min [e_M^{O_2}; e_B^{O_2}] + \min [e_M^{N_2}; e_B^{N_2}] \right) = \dot{M}_B (e_M^{O_2} + e_B^{N_2}) \quad (2.26)$$

$$\dot{E}_{tr\ M-D}^o = \dot{M}_D \left(\min [e_M^{O_2}; e_D^{O_2}] + \min [e_M^{N_2}; e_D^{N_2}] \right) = \dot{M}_D (e_D^{O_2} + e_M^{N_2}) \quad (2.27)$$

The transiting thermal and mechanical components of exergy in each part of the stream, according to (2.11), is:

$$\dot{E}_{tr\ M-B}^{\Delta T, \Delta p} = \dot{M}_B \min [e_M(p_B, T_M); e_B(p_B, T_M)] = \dot{M}_B e_B(p_B, T_M) \quad (2.28)$$

$$\dot{E}_{tr\ M-D}^{\Delta T, \Delta p} = \dot{M}_D \min [e_M(p_D, T_M); e_D(p_D, T_M)] = \dot{M}_D e_D(p_D, T_M) \quad (2.29)$$

The chemical exergy change to be found in the numerator of (2.7) is determined with (2.26) and (2.27):

$$\begin{aligned} \Delta \dot{E}_{out}^0 &= \left[\sum_b \sum_j (\dot{E}_{out,b}^{o,j} - \dot{E}_{tr,b}^{o,j}) \right] = \dot{M}_B (e_B^{O_2} + e_B^{N_2}) + \dot{M}_D (e_D^{O_2} + e_D^{N_2}) \\ &\quad - \dot{M}_B (e_M^{O_2} + e_B^{N_2}) - \dot{M}_D (e_D^{O_2} + e_M^{N_2}) \\ &= \dot{M}_B e_B^{O_2} + \dot{M}_D e_D^{N_2} \end{aligned} \quad (2.30)$$

We have made use of $e_M^{O_2} = 0$, $e_M^{N_2} = 0$ in (2.30), since these are values of the specific chemical exergy in the reference state.

The change of the thermal and mechanical components of exergy in the numerator of (2.7) is obtained using (2.28) and (2.29):

$$\begin{aligned} \Delta \dot{E}_{out}^{\Delta p, \Delta T} &= \sum_b (\dot{E}_{out}^{\Delta p, \Delta T} - \dot{E}_{tr}^{\Delta p, \Delta T}) = \dot{M}_B [e_B(p_B, T_B) - e_B(p_B, T_M)] \\ &\quad + \dot{M}_D [e_D(p_D, T_D) - e_D(p_D, T_M)] \\ &= \dot{M}_B [\Delta e_B^{\Delta T}] + \dot{M}_D [\Delta e_D^{\Delta T}] \end{aligned} \quad (2.31)$$

where, $\Delta e_B^{\Delta T} = e_B(p_B, T_B) - e_B(p_B, T_M)$, which is the increase in the thermal component of exergy of flow B at pressure p_B and $\Delta e_D^{\Delta T} = e_D(p_D, T_D) - e_D(p_D, T_M)$, which is the increase in the thermal component of exergy of flow D at pressure p_D .

The chemical exergy change of the denominator of (2.7) is:

$$\begin{aligned} \Delta \dot{E}_{in}^o &= \left[\sum_b \sum_j (\dot{E}_{in,b}^{o,j} - \dot{E}_{tr,b}^{o,j}) \right] = \dot{M}_M (e_M^{O_2} + e_M^{N_2}) - \dot{M}_B (e_M^{O_2} + e_B^{N_2}) - \dot{M}_D (e_D^{O_2} + e_M^{N_2}) \\ &= -\dot{M}_B e_B^{N_2} - \dot{M}_D e_D^{O_2} \end{aligned} \quad (2.32)$$

We have used $(e_M^{O_2} + e_M^{N_2}) = 0$ in (2.32), since these are values of the specific chemical exergy in the reference state.

The exergy change of the thermal and mechanical components to be found in the denominator of (2.7) is:

$$\begin{aligned}\Delta \dot{E}_{in}^{\Delta p, \Delta T} &= \sum_b (\dot{E}_{in}^{\Delta p, \Delta T} - \dot{E}_{tr}^{\Delta p, \Delta T}) = \dot{M}_B [e_M(p_M, T_M) - e_B(p_B, T_M)] + \\ &\quad \dot{M}_D [e_M(p_M, T_M) - e_D(p_D, T_M)] \\ &= \dot{M}_B [\Delta e_B^{\Delta p}] + \dot{M}_D [\Delta e_D^{\Delta p}]\end{aligned}\quad (2.33)$$

which is the reduction of exergy flows in the system caused by the pressure drop at temperature T_M . Note, that in (2.33) $\Delta e_B^{\Delta p} = e_M(p_M, T_M) - e_B(p_B, T_M)$ and $\Delta e_D^{\Delta p} = e_M(p_M, T_M) - e_D(p_D, T_M)$

The exergetic efficiency with transiting exergy is now formulated using (2.7) and (2.30) - (2.33):

$$\eta_e = \frac{\Delta \dot{E}_{out}^o + \Delta \dot{E}_{out}^{\Delta p, \Delta T}}{\Delta \dot{E}_{in}^o + \Delta \dot{E}_{in}^{\Delta p, \Delta T}} = \frac{\dot{M}_B e_B^{O_2} + \dot{M}_D e_D^{N_2} + \Delta \dot{E}_B^{\Delta T} + \Delta \dot{E}_D^{\Delta T}}{-\dot{M}_B e_B^{N_2} - \dot{M}_D e_D^{O_2} + \Delta \dot{E}_B^{\Delta p} + \Delta \dot{E}_D^{\Delta p}}\quad (2.34)$$

Using data from Table 2.1 and expression (2.34) η_e can now be calculated. The components of the exergy of stream B and D are determined by assuming the streams to behave as a ideal gasses.

$$\eta_e = \frac{1.056 + 0.484 + 0.671 + 5.273}{0.030 + 0.3063 + 2.84 + 17.10} = 0.369\quad (2.35)$$

2.4 Discussion

The simple efficiency gives the highest efficiency compared to the other efficiencies. However, if the chemical exergy of the mixture was much higher, for example in crude oil separation, the simple efficiency would be close to unity. This gives the false impression that the column is working very efficiently. Therefore, the simple efficiency is of little use for individual process plants and units. A great difference between efficiency with transiting exergy and rational efficiency is the calculation of exergy changes of streams caused by pressure and temperature differences. In the case of rational efficiency the changes are calculated by subtracting the thermal and pressure components of exergy from the streams under consideration. In the case of the efficiency with transiting exergy the changes are calculated by subtracting the exergy of the stream(s) under consideration in the unchanged condition from the stream(s) in the changed condition (pressure or temperature change). When the streams can be considered to behave as ideal gasses, no difference will occur between the two calculation methods. However, when this is not the case, like air of 40 bar, the results will be different.

2.5 Conclusions

As shown the simple efficiency is of little use for individual process plants and units and may even lead to false conclusions. However, the simple efficiency can be useful for product design, as in the case of the calculation of the cumulative degree of perfection (see Szargut et al., 1988), when all the incoming exergy flow is transformed into the desired product(s). It maybe noted that both the rational efficiency and the efficiency with transiting exergy reduce to the simple efficiency in such a case.

The rational efficiency is a form of efficiency, which is easy to apply and gives a good insight into the process. To define the desired output and the used exergy for a unit it is necessary to be clear about the function of the unit in the context of the plant.

The efficiency with transiting exergy is a straightforward method for an appropriate definition of efficiency, which gives an unambiguous answer. However, the calculations can become very complex.

The rational efficiency is used in the following chapters of this thesis.

2.6 Nomenclature

E, e	= exergy, specific exergy
I	= irreversibility
max	= maximum
min	= minimum
P	= pressure
T	= temperature
η	= simple efficiency
η_e	= efficiency with transiting exergy
ψ	= rational efficiency

superscripts

o	= environmental state, chemical component ³
ph	= physical component
ΔP	= pressure component
Q	= heat
ΔT	= thermal component
W	= work
•	= rate with respect to time

³ The chemical component of a material is determined at the environmental state conditions, T_0 and P_0 , so the same notation can be used.

subscripts

B	=	bottom stream
D	=	distillate stream
in	=	incoming
M	=	mixture
out	=	outgoing
tr	=	transiting

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Performance criteria for a heat exchanger

Abstract

In this chapter the value of the exergetic efficiency is demonstrated. The performance of heat exchangers is usually measured by the effectiveness as defined by Kays and London (1984). The effectiveness is the amount of heat actually transferred divided by the maximum amount of theoretically transferable heat. So the effectiveness is based on the first law and it does not take into account quality losses of the transferred heat and pressure drops of the fluids. A high effectiveness is no guarantee for a good performance of the heat exchanger. Therefore a new criterion is used, which is not only based on the first, but also on the second law of thermodynamics: the rational efficiency. An equation is derived which includes the effectiveness to calculate the rational efficiency. An example has been included in which both performance criteria are compared.

3.1 Introduction

The effectiveness is widely used for quantifying the performance of heat exchangers. However, there is no direct relation between effectiveness and irreversibility. The irreversibility, i.e. the exergy loss, can be seen as the fuel which is lost due to heat transfer and fluid friction. So the rational efficiency is the quantification of the performance from the point of view of fuel saving.

3.2 Theory

3.2.1 Effectiveness

According to Kays and London (1984) the effectiveness is defined as:

$$\varepsilon = \frac{C_h(T_{h,in} - T_{h,out})}{C_{\min}(T_{h,in} - T_{c,in})} = \frac{C_c(T_{c,out} - T_{c,in})}{C_{\min}(T_{h,in} - T_{c,in})} \quad (3.1)$$

where $C_{\min} = C_c$, when $C_h > C_c$ and $C_{\min} = C_h$, when $C_h < C_c$. It is assumed that there are no heat losses to the environment.

In general, it is possible to express

$$\varepsilon = f(N_{tu}, \frac{C_{\min}}{C_{\max}}, \text{flow arrangement}) \quad (3.2)$$

where $N_{tu} = \frac{1}{C_{\min}} \int_0^A U dA = \frac{A\bar{U}}{C_{\min}}$

The non-dimensional parameters in (3.2) are useful in determining the heat exchanger effectiveness (ε). In Kays and London (1984) the effectiveness of different types of heat exchangers is described with these parameters.

3.2.2 Rational efficiency of the heat exchanger

The exergy balance of the incoming and outgoing exergy flows for the heat exchanger is:

$$\dot{E}_{in} = \dot{E}_{out} + \dot{I} \quad (3.3)$$

Substituting exergy flows in terms of the different components

$$\dot{E} = \dot{E}^{ph} + \dot{E}^o = \dot{E}^{\Delta T} + \dot{E}^{\Delta P} + \dot{E}^o \quad (3.4)$$

in (3.3), we obtain

$$\begin{aligned} & (\dot{E}_{c,in}^{\Delta T} + \dot{E}_{c,in}^{\Delta P} + \dot{E}_{c,in}^o) + (\dot{E}_{h,in}^{\Delta T} + \dot{E}_{h,in}^{\Delta P} + \dot{E}_{h,in}^o) = \\ & (\dot{E}_{c,out}^{\Delta T} + \dot{E}_{c,out}^{\Delta P} + \dot{E}_{c,out}^o) + (\dot{E}_{h,out}^{\Delta T} + \dot{E}_{h,out}^{\Delta P} + \dot{E}_{h,out}^o) + \dot{I} \end{aligned} \quad (3.5)$$

From the consideration of the function of the heat exchanger we can say that the desired exergy output is the increase of the thermal component of exergy of the cold stream, i.e.

$$\dot{E}_{desired\ output} = \dot{E}_c^{\Delta T} \quad (3.6a)$$

when the heat exchanger is operating above T_0 , where

$$\dot{E}_c^{\Delta T} = (\dot{E}_{c,out}^{\Delta T} - \dot{E}_{c,in}^{\Delta T})$$

and, when the heat exchanger is operating below T_0 , its function can be seen as cooling and therefore the desired exergy output is the increase of the thermal component of exergy of the hot stream⁴, i.e.

$$\dot{E}_{desired\ output} = -\dot{E}_h^{\Delta T} \quad (3.6b)$$

where $\dot{E}_h^{\Delta T} = (\dot{E}_{h,in}^{\Delta T} - \dot{E}_{h,out}^{\Delta T})$.

However, when the heat exchanger is operating above and below T_0 (3.6a) or (3.6b) has to be chosen depending on the fact which desired output has the greatest value in terms of exergy.

With reference to (2.4), which described the way in which the rational efficiency is formulated, we can now identify the required exergy input (\dot{E}_{used}) with the aid of expression (3.5) and (3.6). The change in the chemical exergy component is zero because no chemical reactions take place. So, the chemical components can be left out. This leads to

$$\dot{E}_{used} = \dot{E}_h^{\Delta T} + \dot{E}_h^{\Delta P} + \dot{E}_c^{\Delta P} \quad (3.7a)$$

when the heat exchanger operates above T_0 and where $\dot{E}_h^{\Delta P} = (\dot{E}_{h,in}^{\Delta P} - \dot{E}_{h,out}^{\Delta P})$ and $\dot{E}_c^{\Delta P} = (\dot{E}_{c,in}^{\Delta P} - \dot{E}_{c,out}^{\Delta P})$. For the heat exchanger operating below T_0 the required exergy input (\dot{E}_{used}) becomes

$$\dot{E}_{used} = -\dot{E}_c^{\Delta T} + \dot{E}_h^{\Delta P} + \dot{E}_c^{\Delta P} \quad (3.7b)$$

By using (2.3) the rational efficiency of the heat exchanger has been determined to be

$$\psi_{heating} = \frac{\dot{E}_c^{\Delta T}}{\dot{E}_h^{\Delta T} + \dot{E}_h^{\Delta P} + \dot{E}_c^{\Delta P}} \quad (3.8a)$$

or using (2.5)

$$\psi_{heating} = \frac{\dot{E}_c^{\Delta T}}{\dot{E}_c^{\Delta T} + \dot{I}}$$

for operating above T_0 and

⁴ At temperatures below T_0 the exergy of the streams becomes greater at lower temperatures, so $\dot{E}_h^{\Delta T}$ becomes negative

$$\Psi_{\text{cooling}} = \frac{-\dot{E}_h^{\Delta T}}{-\dot{E}_c^{\Delta T} + \dot{E}_h^{\Delta P} + \dot{E}_c^{\Delta P}} \quad (3.8b)$$

or using (2.5)

$$\Psi_{\text{cooling}} = \frac{-\dot{E}_h^{\Delta T}}{-\dot{E}_h^{\Delta T} + \dot{I}}$$

for operating below T_0 .

3.2.3 Calculation of exergy

For the exergy change of the hot and cold streams can be written according to (1.3) and with help of the ideal gas relations

$$\begin{aligned} \dot{E}_{\text{out}} - \dot{E}_{\text{in}} &= \dot{H}_{\text{out}} - \dot{H}_{\text{in}} + T_0(\dot{S}_{\text{out}} - \dot{S}_{\text{in}}) = \\ &C_p(T_{\text{out}} - T_{\text{in}}) - T_0 C_p \ln\left(\frac{T_{\text{out}}}{T_{\text{in}}}\right) + \dot{m} T_0 R \ln\left(\frac{P_{\text{out}}}{P_{\text{in}}}\right) \end{aligned} \quad (3.9a)$$

or liquid relations

$$\begin{aligned} \dot{E}_{\text{out}} - \dot{E}_{\text{in}} &= \dot{H}_{\text{out}} - \dot{H}_{\text{in}} + T_0(\dot{S}_{\text{out}} - \dot{S}_{\text{in}}) = \\ &C_p(T_{\text{out}} - T_{\text{in}}) - T_0 C_p \ln\left(\frac{T_{\text{out}}}{T_{\text{in}}}\right) + \dot{m} \frac{(P_{\text{out}} - P_{\text{in}})}{\rho} \end{aligned} \quad (3.9b)$$

Formula (3.1) can be rewritten in

$$T_{\text{h,out}} = T_{\text{h,in}} - \varepsilon \cdot \frac{C_{\text{min}}}{C_{\text{h}}}(T_{\text{h,in}} - T_{\text{c,in}}) \quad (3.10a)$$

and

$$T_{\text{c,out}} = T_{\text{c,in}} + \varepsilon \cdot \frac{C_{\text{min}}}{C_{\text{c}}}(T_{\text{h,in}} - T_{\text{c,in}}) \quad (3.10b)$$

With help of (3.9), (3.10a) and (3.10b) and neglecting the pressure drops (3.8a) can be rewritten in

$$\psi_{heating} = \frac{\dot{E}_c^{\Delta T}}{\dot{E}_h^{\Delta T}} = \frac{\varepsilon C_{\min}(T_{h,in} - T_{c,in}) - T_0 C_c \ln\left(1 + \varepsilon \frac{C_{\min}}{C_c} \left(\frac{T_{h,in}}{T_{c,in}} - 1\right)\right)}{\varepsilon C_{\min}(T_{h,in} - T_{c,in}) + T_0 C_h \ln\left(1 - \varepsilon \frac{C_{\min}}{C_h} \left(1 - \frac{T_{c,in}}{T_{h,in}}\right)\right)} \quad (3.11a)$$

or

$$\psi_{heating} = \frac{\dot{E}_c^{\Delta T}}{\dot{E}_h^{\Delta T}} = \frac{1 - \frac{C_c}{C_{\min}} \frac{1}{\varepsilon(R_h - R_c)} \ln\left(1 + \varepsilon \frac{C_{\min}}{C_c} \left(\frac{R_h}{R_c} - 1\right)\right)}{1 + \frac{C_h}{C_{\min}} \frac{1}{\varepsilon(R_h - R_c)} \ln\left(1 - \varepsilon \frac{C_{\min}}{C_h} \left(1 - \frac{R_c}{R_h}\right)\right)} \quad (3.11b)$$

where $R_h = \frac{T_{h,in}}{T_0}$ and $R_c = \frac{T_{c,in}}{T_0}$. The same rewriting can be repeated for formula 3.8b.

Formulas 3.11a and 3.11b are valid for $0 > \varepsilon \geq 1$.

3.3 Example

The example is a counter flow heat exchanger with negligible pressure drop. The reference temperature (T_0) is taken to be 25°C. For various situations the relation of the rational efficiency (ψ) versus the effectiveness (ε) is shown in Figure 3.1 to 3.3.

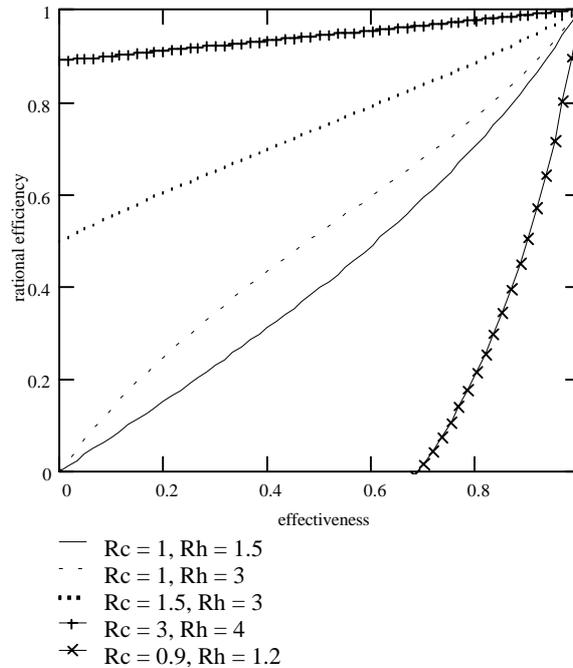


Figure 3.1 Rational efficiency versus effectiveness for various cold and hot temperature ratios for $C_h = C_c$.

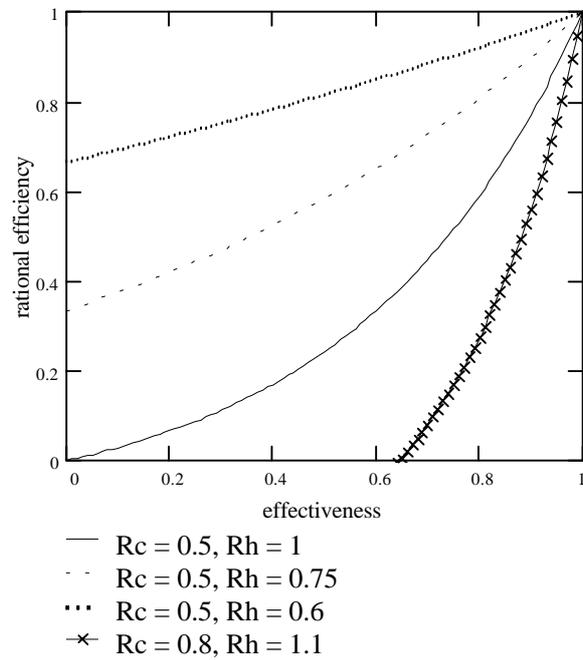


Figure 3.2 Rational efficiency versus effectiveness for various cold and hot temperature ratios for $C_c = C_h$.

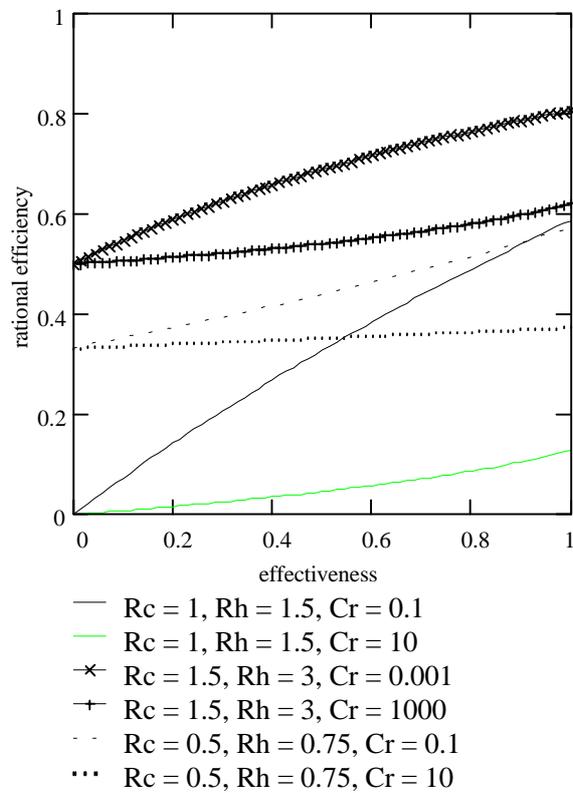


Figure 3.3 Rational efficiency versus effectiveness for different temperature ratios and capacity ratio ($C_r = C_c/C_h$).

The validity of presented data can be extended to every type of heat exchanger with the limitation that the value of ε is bounded to a value lower than 1. For example, in the case of a balanced co-current flow heat exchanger the highest possible value of ε is equal to 0.5.

3.4 Results

A relation between the effectiveness and the rational efficiency has been derived for the countercurrent heat exchanger. For a temperature ratio of the cold flow of one ($R_c = 1$) and a temperature ratio of the hot stream of about 1.5 to 3 the effectiveness and rational efficiency are roughly the same as shown in Figure 3.1. When the temperature ratio of the cold stream increases the rational efficiency becomes greater than the effectiveness, especially at its lower values. This is the case because the exergy transfer at higher temperatures above T_0 increases relatively more than the exergy loss due to the temperature difference between the hot and cold stream (ΔT). When the temperature ratio of the cold stream becomes smaller than one ($R_c < 1$) in combination with the ratio of the hot stream greater than one ($R_h > 1$) the rational efficiency drops substantially compared to the effectiveness, because the exergy of the cold stream due to a temperature below T_0 is destroyed, which leads to a low exergy transfer. This results in a low rational efficiency. However, when both the temperature ratios of the cold and hot stream become smaller than one the rational efficiency becomes greater than the effectiveness as can be seen in Figure 3.2. This can again be explained by the fact that the exergy transfer at lower temperatures below T_0 increases relatively more than the exergy loss due to the ΔT .

When the capacity ratio is not equal to one, an effectiveness of one does not give a rational efficiency of one, which can be explained by the exergy loss due to the capacity mismatch as can be seen in Figure 3.3. Especially, in the case of an evaporator ($C_r \rightarrow \infty$) the rational efficiency becomes rather independent of the effectiveness. This can be explained by the high ΔT when the cold stream evaporates, while there is no condensation at the hot stream.

3.5 Conclusions

Only in the situation of a balanced heat exchanger with an incoming cold temperature close to T_0 and an incoming hot temperature around 1.5 to 3 times T_0 the effectiveness and rational efficiency are similar. In all other situations there is a great difference between the behaviour of the effectiveness and the rational efficiency.

Because there is a direct relation between exergy loss and fuel use the rational efficiency is preferable and the effectiveness should not longer be used as a performance criterion for a heat exchanger.

The rational efficiency can be extended to include other aspects such as exergy loss due to fluid friction and external exergy losses due to the generation of heat and power, see chapter eight for an example.

3.6 Nomenclature

A = heat transfer area
 C = capacity rate
 E = exergy
 I = irreversibility
 \dot{m} = mass flow
 N_{tu} = number of transfer units
 P = pressure
 R = temperature ratio
 T = temperature
 U = heat transfer coefficient

Greek

ϵ = effectiveness
 ρ = density
 ψ = rational efficiency

Superscript

o = environmental state, chemical component
 ph = physical component
 ΔP = pressure component
 ΔT = thermal component

Subscripts

c = cold stream
 h = hot stream
 in = incoming
 min = minimum
 max = maximum
 out = outgoing

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Exergy analyses of distillation processes

Abstract

In this chapter two process units are analysed from an exergetic point of view: the crude oil distillation plant and the cryogenic air separation plant. A conventional exergy analysis is performed on these process units. It is shown that most exergy losses take place in the utilities providing heat or cold. So most attention for improvements is devoted to these installations. The distillations processes were modelled in the software program Aspen Plus and additionally the computer package Advent was used to optimise the heat exchanger network of the crude oil distillation plant. To determine the degree of thermodynamic perfection of the different plant units the rational efficiency was used.

Exergy analysis of the crude oil distillation plant comprising the atmospheric and the vacuum distillation units and the heating section was carried out. The biggest causes of irreversibility are the use of high-quality fuel for the production of middle-quality heat and the dissipation of low quality energy in the environment. The irreversibility rate would be reduced by 16% by using this low quality energy. A significant cause of irreversibility, about one-fourth, is the distillation process itself.

An exergy analysis is performed on the cryogenic distillation process, which is the main method of air separation. It is shown that more than half of the exergy loss takes place in the liquefaction unit and almost one third in the air compression unit. Minor exergy losses are taking place in the distillation unit and the main heat exchanger. The major cause of exergy loss is the use of compressors and to a lesser extent the use of turbines. Especially, the relatively low rational efficiency of the turbines operating in the cryogenic region is striking. Improvements are suggested which save one fourth of the exergy loss. For more substantial reductions of the exergy losses in air separation alternative processes have to be used or developed.

4.1 Introduction

One of the most energy intensive processes in chemical plants is distillation. It accounts for one quarter of the exergy destruction in refineries and chemical industries (Kaibel and

Blass, 1989). Several improvements have been suggested, for example Kaiser et al. (1985) have analysed how to build a reversible distillation column. In this chapter two different distillation plants are analysed and compared.

The computer programs Aspen Plus and Advent were used to assist in the design process. Aspen Plus was most useful in the design and simulation of the distillation plants by calculating the enthalpy and entropy changes of the streams. Advent, a special simulation program to model heat exchanger networks, which uses Pinch Analysis, was used to model the heater and heat exchangers of the crude oil distillation plant. For the crude oil distillation plant only the physical exergy is calculated in Aspen Plus with help of built-in enthalpy and entropy functions. For the calculation of exergy in the case of the air separation plant a special exergy calculation routine has been used, which is called ExerCom and has been developed by Stork Comprimo in co-operation with Delft University of Technology (Lie et al, 1994). More details are described in Hinderink et al. (1996). The data for the modelling of the crude oil distillation plant was obtained from industry and literature, while the data of the air separation plant is based on one specific plant.

The goal of this study is to evaluate the exergy destruction and to suggest improvements in the two distillation plants. Expressions for rational efficiency of the various subsystems analysed were obtained from the basic definition as defined by Kotas (1995).

4.2 An exergy analysis of a crude oil distillation process

4.2.1 Introduction

The fuel use in refineries as found in Heijningen et al. (1992) appears to be quite low when it is compared to the exergy content of the processed crude oil. However, as shown in this chapter, the exergy destruction in distillation processes is big if it is compared with the minimum exergy needed to perform the separation.

Anaya et al. (1990) have made an exergy analysis of a refinery in Mexico. In the latter study the efficiency of the atmospheric and vacuum distillation unit was found to be 8.6%. However, since the method used to calculate the efficiency was described very briefly, it is hard to interpret the results. No absolute values were given. Lee et al. (1989) have used Pinch Analysis to optimise the heat integration system of a crude oil distillation unit.

4.2.2 The distillation plant

Modelling

Two sections, the crude oil distillation unit and vacuum distillation unit, were modelled in Aspen Plus and the heating section in Advent. The crude oil was simulated in Aspen Plus using 34 components (Perry, 1984) in what is known as a pseudo-components representation. The flowsheet is given in Fig 4.1.

The crude oil distillation unit (CDU)

In the CDU the crude oil is separated into different fractions as shown in the flowsheet in Figure 4.1. The feed is 215 tons of crude oil per hour with an API gravity⁵ of 27.6. This can be considered to be a heavy blend. It has only undergone the desaltification process, which is not included in the simulation. The crude oil is supplied at 350°C and brought into the distillation column near its bottom. The column is modelled in 26 stages and has a condenser. A quantity of 10.5 ton steam per hour at 301°C and 8.5 atm and 6.2 MW of energy, supplied with fuel gas, are used for vaporising the oil in the column. Kerosene, light gas oil and heavy gas oil are removed by three side streams and sent to three side-strippers (only one is shown in Fig. 4.1). Three heat exchangers inside the column were used to get a good heat balance in the distillation column.

The vacuum distillation unit (VDU)

The function of this process is to extract gas oil fractions from the atmospheric residue (ATM-RES) of the crude oil distillation by further distillation under low pressure. The ATM-RES enters close to the bottom of the column with a temperature of 407.8°C. The column was modelled in 13 stages and has a condenser. The pressure inside the column is about 0.02 atm and the energy supplied with fuel gas for the column is 17.3 MW. Two heat exchangers are used to get a good heat balance inside the distillation column.

The heating section

The heat exchanger network contains 33 heat exchangers, which heat the crude oil to 313.2°C. A gas fired heater is used to heat the oil further to 350°C. After passing through the heat exchangers all distillate fractions cool to 60°C by the air coolers. With Advent an economic analysis was carried out to estimate a ΔT_{\min} . This has been estimated from the trade-off between the running cost and the capital cost of the heat exchanger network. The results are displayed in Figure 4.2. The ΔT_{\min} was chosen to be 25°C. It has been assumed that the streams have suffered no pressure drops in the heater, heat exchanger network and the air coolers.

⁵ It is equal to a specific mass of $0.8887 \times 10^3 \text{ kg/m}^3$

Figure 4.1 Schematic flowsheet of the oil distillation process

Figure 4.2 The cost function of the heat exchanger network (HEN)

4.2.3 Rational efficiency for the different units

The exergy needed for the reversible separation of a mixture into its N components can be calculated as follows according to formula 1.9:

$$\sum_{i=1}^N x_i \varepsilon_i^o - \varepsilon_{\text{mix}}^o = RT_o \sum_{i=1}^N x_i \ln \gamma_i x_i \quad (4.1)$$

where R is the universal gas constant, x_i and γ_i are the mole fraction and the activity coefficient of the i -th component, respectively. For ideal solutions the activity coefficient is equal to one. T_o and P_o are assumed to be 298.15 K and 1.00 atm, respectively.

Rational efficiency of the CDU

With reference to Fig. 4.1 the exergy balance for the incoming and outgoing exergy flows is set up for the CDU:

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}} + \dot{I} \quad (4.2)$$

Substituting exergy flows in terms of the two components in (4.2)

$$\dot{E} = \dot{E}^{\text{ph}} + \dot{E}^o \quad (4.3)$$

we obtain

$$\dot{E}_{\text{CH}}^{\text{ph}} + \dot{E}_{\text{CH}}^o + \dot{E}_{\text{S}} + \dot{E}_{\text{GAS2}} + \dot{E}_{\text{GAS3}} = \sum_{\text{fr CDU}} (\dot{E}_i^{\text{ph}} + \dot{E}_i^o) + \dot{E}_{\text{CDU}}^Q + \dot{E}_{\text{W}} + \dot{I}_{\text{CDU}} \quad (4.4)$$

where
$$\dot{E}_{\text{S}} = \dot{E}_{\text{STEAM1}} + \dot{E}_{\text{STEAM2}} + \dot{E}_{\text{STEAM3}} \quad (4.5)$$

and $\sum_{\text{fr CDU}}$ is the sum for all the oil fractions leaving the CDU and \dot{E}_{CDU}^Q is the thermal exergy flow leaving the CDU and going to the heat exchanger network. Other symbols used are explained in Nomenclature. From the consideration of the function of the CDU we can say that the desired exergy output is the increase of chemical exergy of the crude oil (the mixture) to that of the fractions as a result of the distillation process, i.e.

$$\left(\sum_{\text{fr CDU}} \dot{E}_i^o - \dot{E}_{\text{CH}}^o \right) \quad (4.6)$$

There is a by-product produced from the CDU in the form of thermal exergy \dot{E}_{CDU}^Q which is not rejected to the environment and is utilised for preheating the crude oil. Therefore

$$\dot{E}_{\text{desired output}} = \left(\sum_{\text{fr CDU}} \dot{E}_i^o - \dot{E}_{\text{CH}}^o \right) + \dot{E}_{\text{CDU}}^Q \quad (4.7)$$

With reference to (2.4), which described the way in which the rational efficiency is formulated, we can now identify the required exergy input (\dot{E}_{used}) in the distillation process with the aid of expression (4.4) and (4.7). This leads to the following expression for the exergy used:

$$\dot{E}_{used} = (\dot{E}_{CH}^{ph} - \sum_{fr\ CDU} \dot{E}_i^{ph}) + (\dot{E}_S - \dot{E}_W) + \dot{E}_{GAS2} + \dot{E}_{GAS3} \quad (4.8)$$

By using (2.3) the rational efficiency of the CDU has been determined to be

$$\psi_{CDU} = \frac{(\sum_{fr\ CDU} \dot{E}_i^o - \dot{E}_{CH}^o) + \dot{E}_{CDU}^Q}{(\dot{E}_{CH}^{ph} - \sum_{fr\ CDU} \dot{E}_i^{ph}) + (\dot{E}_S - \dot{E}_W) + \dot{E}_{GAS2} + \dot{E}_{GAS3}} \quad (4.9)$$

or using (2.5)

$$\psi_{CDU} = \frac{(\sum_{fr\ CDU} \dot{E}_i^o - \dot{E}_{CH}^o) + \dot{E}_{CDU}^Q}{(\sum_{fr\ CDU} \dot{E}_i^o - \dot{E}_{CH}^o) + \dot{E}_{CDU}^Q + \dot{I}_{CDU}} \quad (4.10)$$

Rational efficiency of the VDU

In the same way the rational efficiency for the VDU has been determined. The exergy balance according to (4.5) is:

$$\dot{E}_{AR}^{ph} + \dot{E}_{AR}^o + \dot{E}_{GAS4} = \dot{E}_{VDU}^Q + \sum_{fr\ VDU} (\dot{E}_i^{ph} + \dot{E}_i^o) + \dot{I}_{VDU} \quad (4.11)$$

where $\sum_{fr\ VDU}$ is the sum for all the oil fractions leaving the VDU and \dot{E}_{VDU}^Q is the thermal exergy flow leaving the VDU and going to the heat exchanger network.

The desired exergy output is:

$$\dot{E}_{desired\ output} = (\sum_{fr\ VDU} \dot{E}_{AR}^o - \dot{E}_{AR}^o) + \dot{E}_{VDU}^Q \quad (4.12)$$

This leads with reference to (2.4) and with aid of expression (4.11) and (4.12) to the following expression:

$$\dot{E}_{used} = (\dot{E}_{AR}^{ph} - \sum_{fr\ VDU} \dot{E}_{AR}^{ph}) + \dot{E}_{GAS4} \quad (4.13)$$

By using (2.3) the rational efficiency of the VDU has been determined to be:

$$\Psi_{\text{VDU}} = \frac{(\sum_{fr \text{ VDU}} \dot{E}_{\text{AR}}^{\text{o}} - \dot{E}_{\text{AR}}^{\text{o}}) + \dot{E}_{\text{VDU}}^{\text{Q}}}{(\dot{E}_{\text{AR}}^{\text{ph}} - \sum_{fr \text{ VDU}} \dot{E}_{\text{AR}}^{\text{ph}}) + \dot{E}_{\text{GAS4}}}$$
 (4.14)

Or using (2.5)

$$\Psi_{\text{VDU}} = \frac{(\sum_{fr \text{ VDU}} \dot{E}_{\text{AR}}^{\text{o}} - \dot{E}_{\text{AR}}^{\text{o}}) + \dot{E}_{\text{VDU}}^{\text{Q}}}{(\sum_{fr \text{ VDU}} \dot{E}_{\text{AR}}^{\text{o}} - \dot{E}_{\text{AR}}^{\text{o}}) + \dot{E}_{\text{VDU}}^{\text{Q}} + \dot{I}_{\text{VDU}}}$$
 (4.15)

The rational efficiency of the HEN

The exergy balance for the heat exchanger network is:

$$\sum_{i=1}^n \dot{E}_{i, \text{in}, \text{hot}} + \sum_{i=1}^n \dot{E}_{i, \text{in}, \text{cold}} = \sum_{i=1}^n \dot{E}_{i, \text{out}, \text{hot}} + \sum_{i=1}^n \dot{E}_{i, \text{out}, \text{cold}} + \sum_{i=1}^n \dot{I}_{i, \text{H}}$$
 (4.16)

where n is the number of heat exchangers. Because chemical exergy changes do not take place in the heat exchangers, all exergy flows in (4.16) can be treated as physical exergy flows. The desired output is the exergy increase of the crude oil stream:

$$\dot{E}_{\text{desired output}} = \sum_{i=1}^n \dot{E}_{i, \text{out}, \text{cold}}^{\Delta T} - \sum_{i=1}^n \dot{E}_{i, \text{in}, \text{cold}}^{\Delta T}$$
 (4.17)

which in term of the specific symbols used here is:

$$\dot{E}_{\text{desired output}} = \dot{E}_{\text{out}, \text{C}}^{\Delta T} - \dot{E}_{\text{in}, \text{C}}^{\Delta T}$$
 (4.18)

The last term of the equation is zero, because the incoming crude oil has the environmental temperature. This leads, with reference to (2.4) and with the aid of expressions (4.17) and (4.18) and the further assumption that no pressure drops take place in the HEN, to the following expression for the exergy used:

$$\dot{E}_{\text{used}} = \sum_{i=1}^n \dot{E}_{i, \text{in}, \text{hot}} - \sum_{i=1}^n \dot{E}_{i, \text{out}, \text{hot}}$$
 (4.19)

Using (2.3) the rational efficiency is:

$$\Psi_{\text{HEN}} = \frac{\dot{E}_{\text{out}, \text{C}}^{\Delta T}}{\sum_{i=1}^n \dot{E}_{i, \text{in}, \text{hot}} - \sum_{i=1}^n \dot{E}_{i, \text{out}, \text{hot}}}$$
 (4.20)

or using (2.5)

$$\psi_{HEN} = \frac{\dot{E}_{out,C}^{\Delta T}}{\dot{E}_{out,C}^{\Delta T} + \sum_{i=1}^n \dot{I}_{i,H}} \quad (4.21)$$

The efficiency of the heater

With reference to Fig. 4.1 the exergy balance of the heater is

$$\dot{E}_C + \dot{E}_{GAS1} = \dot{E}_{CH} + \dot{I}_{heater} \quad (4.22)$$

The exergy used is considered to be : $\dot{E}_{used} = \dot{E}_{G1}$ (4.23)

With reference to (2.4), with use of (4.22) and (4.23) and the assumption that no pressure drops take place in the heater the desired output is:

$$\dot{E}_{desired\ output} = \dot{E}_{CH}^{\Delta T} - \dot{E}_C^{\Delta T} \quad (4.24)$$

This according to (2.3) gives the following rational efficiency:

$$\psi_{heater} = \frac{\dot{E}_{CH}^{\Delta T} - \dot{E}_C^{\Delta T}}{\dot{E}_{G1}} \quad (4.25)$$

Rational efficiency of the whole plant

The exergy balance for the whole distillation plant is given by:

$$\dot{E}_C^o + \sum_{all} \dot{E}_{GASi} + \dot{E}_S = \sum_{fr\ all} (\dot{E}_{out,i}^{ph} + \dot{E}_i^o) + \dot{E}_W + \sum_{all} \dot{I}_i \quad (4.26)$$

where $\sum_{all} \dot{E}_{GASi}$ is all the fuel gas streams, $\dot{E}_C^o = \dot{E}_{CH}^o$ and $\sum_{fr\ all} \dot{E}_{out,i}^{ph}$ means the physical exergy of the fractions leaving the distillation plant at 60°C. This physical exergy is used in other parts of the refinery. From (4.26) it is clear that the exergy used is:

$$\dot{E}_{used} = \sum_{all} \dot{E}_{GASi} + (\dot{E}_S - \dot{E}_W) \quad (4.27)$$

This leads with reference to (2.4) and with the aid of expressions (4.26) and (4.27) to the following expression for the desired exergy output:

$$\dot{E}_{desired\ output} = (\sum_{fr\ all} \dot{E}_i^o - \dot{E}_C^o) + \sum_{fr\ all} \dot{E}_{out,i}^{ph} \quad (4.28)$$

Using (2.3), the rational efficiency for the whole distillation plant is given by:

$$\psi_{\text{plant}} = \frac{(\sum_{fr\ all} \dot{E}_i^o - \dot{E}_C^o) + \sum_{fr\ all} \dot{E}_{out,i}^{ph}}{\sum_{all} \dot{E}_{GASi} + (\dot{E}_S - \dot{E}_W)} \quad (4.29)$$

or from (2.5)

$$\psi_{\text{plant}} = \frac{(\sum_{fr\ all} \dot{E}_i^o - \dot{E}_C^o) + \sum_{fr\ all} \dot{E}_{out,i}^{ph}}{(\sum_{fr\ all} \dot{E}_i^o - \dot{E}_C^o) + \sum_{fr\ all} \dot{E}_{out,i}^{ph} + \sum_{all} \dot{I}_i} \quad (4.30)$$

The efficiency calculated in formula 4.29 can not be obtained by multiplying the efficiencies of the preceding units, as would be the case for the simple efficiency.

4.2.4 Results

The temperature, mass flow, enthalpy and exergy of the flows are shown in Table 4.1, 4.2 and 4.3. The used abbreviations for the flows are shown in Figure 4.1. The enthalpy values are calculated according to the reference system of Aspen Plus, which is not discussed here. These are displayed here to show that the energy balance is closed. The irreversibility rate and the rational efficiency of the different units are shown in Table 4.4.

The crude distillation unit (CDU)

The exergy analysis of the CDU showed an exergy destruction of 5881 kW for the distillation column. The irreversibility rate in the reboilers is 2890 kW, leaving an irreversibility rate of 2991 kW in the remaining parts of the column and the side-strippers. An exergy destruction of 4153 kW due to dissipation into the environment of the thermal energy of the condenser cooling water (COND), 3790 kW of exergy destruction, and a part of the first internal heat exchanger (HEAT1), an exergy destruction of 363 kW, has to be added to give the total exergy destruction of 10034 kW. The exergy of the fuel is assumed to equal the enthalpy value. The exergy needed for the reversible separation of the fractions, found to be 1007 kW, has been calculated by using (4.1). The activity coefficients of the fractions were assumed to be one and the distillate fractions were taken to be the components of the mixture. The three internal heat exchangers in the CDU involve heat transfer \dot{Q}_{CDU} to a coolant, which carries with it thermal exergy of 2705 kW (part of HEAT1, HEAT2 and HEAT3). This is used subsequently in the heat exchanger network. Using (4.10) the rational efficiency of the CDU was found to be 0.270, of which 27% of the desired output can be attributed to the separation of the components and 73% to the useful heat given by the three internal heat exchangers.

Table 4.1: The components of the CDU

Components	Temp. (in °C)	Press. (in atm.)	Flow ¹⁾ (kg/sec)	Enthalpy (in kW)	Exergy (in kW)
CRUDEH	350.0	2.04	59.74	31980	17416
REF-GAS	80.5	1.72	- 2.14	4736	-69
NAFTA	80.5	1.72	- 7.60	2507	-61
KEROSINE	225.6	1.80	-4.51	-538	-431
LIGHTGAS	242.7	1.88	-6.56	-1291	-776
HEAVYGAS	291.0	1.96	-6.04	-2083	-1065
ATM-RES	407.8	2.00	-33.07	-22582	-11276
STEAM1	301.1	8.51	2.64	-34124	2385
STEAM2	301.1	8.51	0.12	-1554	109
STEAM3	301.1	8.51	0.10	-1349	96
WATER	80.5	1.72	-2.67	41738	-52
GAS2	25.0			6155	6155
GAS3	25.0			308	308
COND	147.9 - 80.5			-16566	-3790
HEAT1	213.5 - 202.8			-2579	-911
HEAT2	262.8 - 247.2			-2421	-1015
HEAT3	328.2 - 313.2			-2344	-1142
Total				-7	5881

¹⁾ The minus sign means the stream is leaving the CDU

The vacuum distillation unit (VDU)

The irreversibility rate in the VDU is 12422 kW. This figure includes the exergy destruction of 11720 kW in the column as shown in Table 4.2 and the exergy destruction of 702 kW due to dissipation in the environment of a part of the thermal energy of the condenser cooling water (VCOND) and VHEAT1. The exergy of the fuel is assumed to be equal to the enthalpy value. The irreversibility rate in the reboiler is 7003 kW, leaving an irreversibility rate of 4717 in the remaining part of the column. The minimum exergy needed to separate the ATM-RES into the fractions, calculated in the same way as the with the CDU, is found to be 217 kW. As in the case of the CDU, the heat transfer, \dot{Q}_{VDU} , to the coolant from the two internal heat exchangers and a part of the condenser yields an exergy flow of 7184 kW, which is used in the heat exchanger network. Using (4.15) a rational efficiency of 0.373 for the VDU was obtained, of which 3% of the desired output can be attributed to the separation of the components and 97% to the useful heat given by the two internal heat exchangers

Table 4.2: The components of the vacuum distillation column

Components	Temp. (in °C)	Press. (in atm.)	Flow ¹⁾ (kg/sec)	Enthalpy (in kW)	Exergy (in kW)
AR	407.8	2.000	33.07	22582	11276
VACGAS	165.7	0.0132	-1.05	-257	-119
VACGASOL	240.4	0.0175	-8.40	-2167	-1073
VACDEST	322.0	0.0208	-8.40	-4124	-1932
VR	436.4	0.0263	-15.22	-11560	-5829
GAS4	25.0			17291	17291
VCOND	184.1 - 165.7			-14430	-4825
VHEAT1	240.4 - 201.2			- 4103	-1526
VHEAT2	322.0 - 278.9			-3224	-1543
Total				8	11720

¹⁾ The minus sign means the stream is leaving the VDU

The heating section

The heating section consists of the heat exchanger network (HEN), the gas fired heater and the air cooler. The exergy increase of the crude oil through heating in the HEN was found to be 14274 kW with a corresponding irreversibility of 3134 kW, see Table 4.3.

Table 4.3: The components of the heat exchanging network

Components	T_{before} (° C)	T_{after} (°C)	Flow (kg/sec)	Enthalpy (kW)	Exergy (kW)
Δ CRUDE	25	307.8	59.74	-41289	-14274
Δ KEROSENE	225.6	184.4	4.51	367	139
Δ LIGHTGAS	242.7	184.4	6.56	793	325
Δ HEAVYGAS	291.0	184.4	6.04	1413	627
HEAT1	213.5	202.8		1490	548
HEAT2	262.8	247.2		2421	1015
HEAT3	328.2	313.2		2344	1142
Δ VACGASOL	240.4	184.4	8.40	1029	436
Δ VACDEST	322.0	184.4	8.40	2721	1273
Δ VACRES	436.4	184.4	15.22	9135	4719
VCOND	184.1	165.7		13320	4480
VHEAT1	240.4	201.2		3034	1169
VHEAT2	322.0	278.9		3224	1535
Total				-2	1582.0

In this table the exergy changes of the flows in the HEN are displayed. The exergy change of the cold stream (Δ CRUDE) is displayed with a negative sign in Table 4.3 to obtain a positive value for the exergy loss. Hence, using (4.21), the rational efficiency is 0.820. The exergy use of the heater is 6.384 MW, while the exergy increase of the heated stream is 3.459 MW. So the rational efficiency of the heater, calculated with the aid of (4.25), is 0.541. The exergy destruction of the distillate fractions due to the cooling is 1891 kW. The electricity consumption of the fans used in the air cooling of the distillate fractions has been omitted from the total exergy balance. No rational efficiency can be defined for the air coolers, because there is no desired output in terms of exergy.

The complete distillation plant

The whole plant under consideration is made up of the three process units referred to above. The total irreversibility for the plant is 30.40 MW as can be seen in Table 4.4. The physical exergy of the fractions, leaving the distillation plant at 60°C, which as mentioned above is used in other parts of the refinery, is 437 kW. This gives an overall rational efficiency of the distillation plant, using (4.30), of 0.0518.

Table 4.4: The irreversibility rates and rational efficiencies of the units

unit	irreversibility rate (MW)	rational efficiency
CDU	10.03	0.270
VDU	12.42	0.373
HEN	3.13	0.820
heater	2.93	0.541
air cooler	1.89	
Total / Overall	30.40	0.0518

Accuracy of the results

Because of lack of suitable data the calculation of exergy of separation (4.1) has been calculated assuming all activity coefficients of the mixture to be unity. Furthermore, for the sake of simplicity, the electrical power consumption by the pumps has been neglected which, for pumping liquids, should involve only a small error. However, in the case of the VDU the program used, Aspen Plus, is designed to include pump power in the simulation of this process. Also, as has been mentioned earlier, the electrical power used by air cooling fans has been neglected.

4.2.5 Suggestions for improvements

About 90% of the exergy used in the plant has been supplied in the form of fuel gas and the remaining 10% comes from steam generated outside the distillation plant. The combustion of this fuel gas in the heater and in the reboilers of the CDU and VDU leads to large irreversibilities of 3134, 2890 and 7003 kW, respectively. This form of exergy destruction could be reduced by the use of heat from a combined heat and power plant. This has already been applied in some form in a few refineries. Assuming a national power generation efficiency of 50% for the calculation of the saving due to the produced electricity the irreversibility can decrease depending of the system from 19% to 60% (Korobitsyn et al., 1995)

The irreversibility rate in the distillation columns, excluding the reboilers, is 7708 kW. A more detailed analysis of the columns has to be made to suggest improvements to reduce this irreversibility rate. By rejecting heat of a temperature between 80.5°C and 184.4°C of the condensers and internal heat exchangers of the CDU and VDU in the form of the cooling water into the environment, 4855 kW of exergy has been destroyed. If this rejected energy were used in a reversible way the rational efficiency of the CDU and VDU would increase from 0.270 and 0.373 to 0.572 and 0.409, respectively and the total irreversibility would reduce by 16%.

By cooling the fractions of the CDU and VDU from 184.4°C to 60°C, 1891 kW of exergy has been destroyed. This exergy together with the waste energy rejected into the environment could be used e.g. in a district heating system. The heat exchanger network can be improved in terms of rational efficiency by lowering ΔT_{\min} . A ΔT_{\min} of 10 K will lower the heating duty of the heater to 3.9 MW instead of 6.4 MW. But as Figure 4.2 shows, this is not justified on economic grounds.

4.2.6 Conclusions for the crude oil distillation

The exergy analysis of a distillation plant of a refinery has pinpointed and quantified the exergy destruction in the different plant sections. The modelling of the distillation plant in the computer packages Aspen Plus and Advent gave a detailed simulation. The highest irreversibilities are in the vacuum distillation unit (VDU) and the crude distillation unit (CDU). The rational efficiencies of the CDU and the VDU were found to be 0.270 and 0.373, respectively. The overall rational efficiency of the distillation plant, including the heater and the heat exchanger network, is 0.0518. The biggest causes of irreversibility are the use of high-quality fuel for the production of middle-quality heat and the dissipation of low quality energy in the environment. The irreversibility of the plant would reduce by 16% by using this low quality energy. A significant cause of irreversibility, about one-fourth, is the distillation process itself, as can be seen from the exergy destruction in the columns, excluding the reboilers.

4.3 Cryogenic air separation

4.3.1 Air separation

The separation of air into its internal components is carried out for industrial and medical use. The greatest use of nitrogen is as an inert blanketing gas and as a reactant in chemical processes. Oxygen is used both for industrial and medical purposes. There are three methods of air separation commercially available: the cryogenic distillation process, the pressure swing adsorption (PSA) process and the membrane separation process. The cryogenic distillation is used when high purity of the products is needed. The PSA process becomes interesting from a commercial point of view when a nitrogen flow between 10 and 100 m³/hr is needed with a purity of 98 to 99.5 vol. %. Membrane separation is used for small flows, less than 10 m³/hr, and low purity, lower than 98.5 vol. % (Hoek Loos). Cryogenic distillation is required, when the products are needed in a liquid form. In this chapter the most widely used air separation process, the cryogenic process, has been analysed.

4.3.2 Cryogenic air separation plant

The analysed plant uses the principle of two-column separation. This is based on a low and a high pressure distillation column. The condenser of the high pressure lower column (LC) is cooled by the reboiler of the low pressure upper column (UC). This principle is widely used for cryogenic air separation plants. The flowsheet of the plant is shown in Figure 4.3. The incoming air is compressed to 6.2 bar and after cooling in the spray cooler (SC) water and carbon dioxide are removed by adsorption in the molecular filters. Subsequently, the air is cooled in the main heat exchanger (MHE) to -172 °C and enters the lower column (LC), which has 54 stages. Here, the air is separated into the top product nitrogen, which is partly liquefied in the condenser and the bottom product (B), which is a mixture of 38 vol.% oxygen, 60 vol. % nitrogen and 1.6 vol. % argon. The gaseous part of the nitrogen (TN) is liquefied in a special section. Here, the nitrogen is compressed to 46 bar with the aid of compressor 1, 2 and 3. Compressor 3 is driven by the hot expansion turbine (HET) and compressor 2 by the cold expansion turbine (CET). The liquefaction of the stream takes place in heat exchanger 2 (HE 2). Further cooling of the liquid nitrogen takes place in the throttling valve (TV). The nitrogen stays almost completely liquefied after passing this valve and enters the lower column again. The bottom product and the liquid nitrogen leaving the lower column are throttled to ca. 1.5 bar in valve 1 (JT 1) and valve 2 (JT2) and enter the upper column (UC), which has 96 stages. Of this column the top product is gaseous (GAN) and liquid nitrogen (LIN) and the bottom product is gaseous (GOX) and liquid oxygen (LOX). One of the side products is unpurified argon, consisting of argon with oxygen and nitrogen. Most of this oxygen and nitrogen returns from the crude argon distillation column to the upper column.

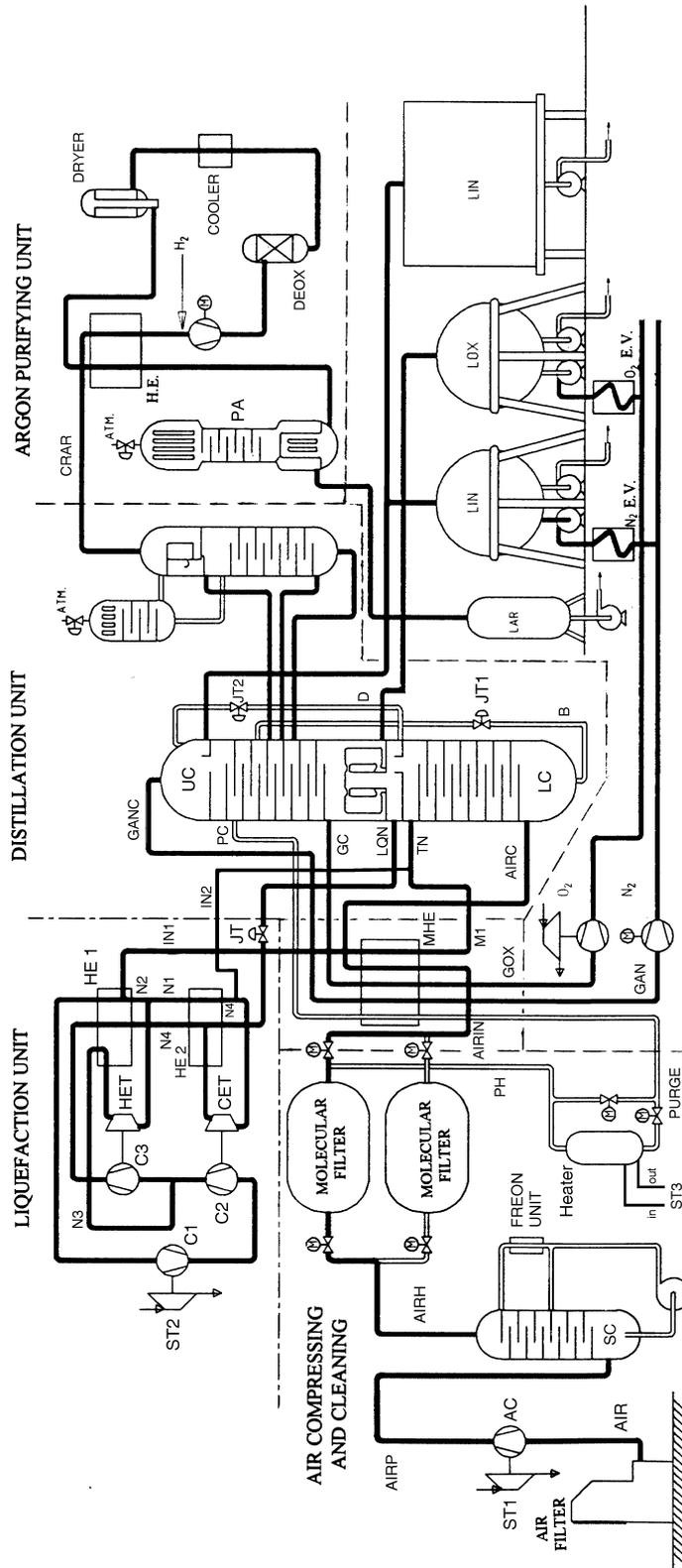


Figure 4.3 Flowsheet of the analysed cryogenic air separation plant

The other side product is purge gas (PC), which is used to clean the molecular filters. It consists mainly of nitrogen and the gas is heated to 170°C before cleaning the filters. The top product of the crude argon distillation column (CRAR) enters the argon purifying unit, where it is compressed to 3 bar and deoxygenated by burning the oxygen with hydrogen. After cooling, separating the water and further cooling to -180°C the argon is further purified in the pure argon distillation column (PA). Compression of the products to the desired pressure is left out in this analysis. The flow and product specification are displayed in Table 4.5. Because of the flexibility of the plant variations of the flow are possible. The analysis has been performed with the flow given in Table 4.5.

Table 4.5. Flow and product specification

Components	flow (in kg/s)	purity (in vol. %)
Air	16.39	--
Gaseous Oxygen (GOX)	2.82	99.5
Liquid Oxygen (LOX)	0.70	99.6
Gaseous Nitrogen (GAN)	6.18	99.95
Liquid Nitrogen (LIN)	2.86	99.999
Liquid Argon (LAR)	0.13	99

4.3.3 Rational efficiency of the different units

For the compressors the following formula is used

$$\psi_{\text{compressor}} = \frac{\dot{E}_{\text{out}} - \dot{E}_{\text{in}}}{\dot{W}_{\text{in}}} \quad (4.31)$$

where W is the work input of the compressor. For turbines the rational efficiency is defined as the work output divided by the exergy decrease of the incoming and outgoing flows, in formula form

$$\psi_{\text{turbines}} = \frac{\dot{W}_{\text{out}}}{\dot{E}_{\text{out}} - \dot{E}_{\text{in}}} \quad (4.32)$$

For the sake of simplicity no distinction is made in the rational efficiency of the heat exchanger for the thermal and mechanical component as is suggested in (3.8). The rational efficiency is then defined as

$$\psi_{\text{HE,heating}} = \frac{\dot{E}_{\text{cold,out}} - \dot{E}_{\text{cold,in}}}{\dot{E}_{\text{hot,in}} - \dot{E}_{\text{hot,out}}} \quad (4.33a)$$

for operating above T_0 and

$$\psi_{\text{HE,cooling}} = \frac{\dot{E}_{\text{hot,in}} - \dot{E}_{\text{hot,out}}}{\dot{E}_{\text{cold,out}} - \dot{E}_{\text{cold,in}}} \quad (4.33b)$$

for operating below T_0 .

The difference between formula 3.8 and 4.33 is small in the case of the heat exchangers operating in the air separation unit, because the mechanical losses are small. No chemical reactions occur in the heat exchangers, so the difference in exergy of the streams are taken, because the chemical component will have no influence on the rational efficiency.

Because no distinction is made between the thermal and mechanical component in the case of the throttling valve it was not possible to define a desired product, so no efficiency could be calculated. No desired product can be defined in the case of mixing and also no rational efficiency can be obtained.

For the air separation and cleaning unit the exergy balance⁶ is

$$\dot{E}_{\text{AIR}} + \dot{E}_{\text{ST1,in}} + \dot{W}_{\text{freon}} + \dot{E}_{\text{PURGE}} + \dot{E}_{\text{ST3,in}} = \dot{E}_{\text{AIRIN}} + \dot{E}_{\text{ST1,out}} + \dot{E}_{\text{ST3,out}} + \dot{I}_{\text{ACC}} \quad (4.34)$$

The desired product is

$$\dot{E}_{\text{desired output}} = \dot{E}_{\text{AIRIN}} - \dot{E}_{\text{AIR}} \quad (4.35)$$

which leads to the following expression for the exergy used:

$$\dot{E}_{\text{used}} = \dot{E}_{\text{ST1,in}} - \dot{E}_{\text{ST2,out}} + \dot{E}_{\text{ST3,in}} - \dot{E}_{\text{ST3,out}} + \dot{W}_{\text{freon}} + \dot{E}_{\text{PURGE}} \quad (4.36)$$

The rational efficiency is then

$$\psi_{\text{ACC}} = \frac{\dot{E}_{\text{AIRIN}} - \dot{E}_{\text{AIR}}}{\dot{E}_{\text{ST1,in}} - \dot{E}_{\text{ST1,out}} + \dot{E}_{\text{ST3,in}} - \dot{E}_{\text{ST3,out}} + \dot{W}_{\text{freon}} + \dot{E}_{\text{PURGE}}} \quad (4.37)$$

or in the rewritten form by (2.5)

$$\psi_{\text{ACC}} = \frac{\dot{E}_{\text{AIRIN}} - \dot{E}_{\text{AIR}}}{\dot{E}_{\text{AIRIN}} - \dot{E}_{\text{AIR}} + \dot{I}_{\text{ACC}}}$$

For the lower distillation column the exergy balance is

$$\dot{E}_{\text{AIRC}} + \dot{E}_{\text{LQN}} + \dot{E}_{\text{cond}}^Q = \dot{E}_{\text{B}} + \dot{E}_{\text{D}} + \dot{E}_{\text{TN}} + \dot{I}_{\text{LC}} \quad (4.38)$$

⁶ Stream ST1_{in} and ST1_{out} refer to the incoming and outgoing stream of steam turbine 1 (ST1). This notation is used further on for more components displayed in Figure 4.3

The desired product is as discussed in 4.2.3

$$\begin{aligned}\dot{E}_{desired\ output} &= (\sum_{fr} \dot{E}_i^o - \dot{E}_{mix}^o) \\ &= \dot{E}_B^o + \dot{E}_D^o + \dot{E}_{TN}^o - \dot{E}_{AIRC}^o - \dot{E}_{LQN}^o\end{aligned}\quad (4.39)$$

where the mixture is the incoming stream(s) and the fractions are the leaving streams. \dot{E}_{LQN}^o and \dot{E}_{TN}^o are equal and can be left out for further analysis. This leads to the following expression for the exergy used:

$$\begin{aligned}\dot{E}_{used} &= (\dot{E}_{mix}^{ph} - \sum_{fr} \dot{E}_i^{ph}) + \dot{E}_{cond}^Q \\ &= (\dot{E}_{AIRC}^{ph} + \dot{E}_{LQN}^{ph} - \dot{E}_B^{ph} - \dot{E}_D^{ph} - \dot{E}_{TN}^{ph}) + \dot{E}_{cond}^Q\end{aligned}\quad (4.40)$$

The rational efficiency for the lower distillation column is then

$$\psi_{LC} = \frac{\dot{E}_B^o + \dot{E}_D^o + \dot{E}_{TN}^o - \dot{E}_{AIRC}^o - \dot{E}_{LQN}^o}{(\dot{E}_{AIRC}^{ph} + \dot{E}_{LQN}^{ph} - \dot{E}_B^{ph} - \dot{E}_D^{ph} - \dot{E}_{TN}^{ph}) + \dot{E}_{cond}^Q}\quad (4.41)$$

For the upper distillation column the exergy balance is

$$\dot{E}_{JT1,in} + \dot{E}_{JT2,out} = \sum_{fr} \dot{E}_i + \dot{E}_{cond}^Q + \dot{I}_{LC}\quad (4.42)$$

where the fractions (fr) are GANC, LIN, CRAR, PC, GC and LOX. For the upper column, including the crude argon distillation column, the desired product is

$$\dot{E}_{desired\ output} = (\sum_{fr} \dot{E}_i^o - \dot{E}_{mix}^o) + \dot{E}_{cond}^Q\quad (4.43)$$

where mix is JT1, in and JT2, out and this results to the following expression for the exergy used:

$$\dot{E}_{used} = (\dot{E}_{mix}^{ph} - \sum_{fr} \dot{E}_i^{ph})\quad (4.44)$$

The rational efficiency for the upper distillation column is then

$$\psi_{UC} = \frac{(\sum_{fr} \dot{E}_i^o - \dot{E}_{mix}^o) + \dot{E}_{cond}^Q}{(\dot{E}_{mix}^{ph} - \sum_{fr} \dot{E}_i^{ph})}\quad (4.45)$$

Because there is no heat exchange with the environment, the rational efficiency for the distillation section is

$$\Psi_{\text{DIST}} = \frac{(\sum_{fr} \dot{E}_i^o - \dot{E}_{\text{AIRC}}^o)}{(\dot{E}_{\text{AIRC}}^{\text{ph}} - \sum_{fr} \dot{E}_i^{\text{ph}})} \quad (4.46)$$

as can be derived by setting up the exergy balance. The fractions are the flows leaving the section. The same formula can be applied for the argon purifying unit by replacing AIRC by CRAR in (4.46). The outgoing fraction is then only the LAR. The top product is released and dissipated in the environment.

The desired output of liquefaction unit is seen as the exergy increase of the outgoing flow compared to the incoming flow because of the cooling and liquefying of the nitrogen. By setting up the exergy balance and defining the desired exergy output and thus the exergy used as just described the following formula is obtained

$$\Psi_{\text{LIQ}} = \frac{\dot{E}_{\text{LQN}} - (\dot{E}_{\text{IN1}} + \dot{E}_{\text{IN2}})}{(\dot{E}_{\text{ST2,in}} - \dot{E}_{\text{ST2,out}})} \quad (4.47)$$

The exergy balance for the whole air separation plant is given by:

$$\dot{E}_{\text{air}} + \sum_{\text{all}} \dot{E}_{\text{Steam,in}} + \dot{W}_{\text{freon}} = \sum_{\text{fr all}} (\dot{E}_{\text{out,i}}^{\text{ph}} + \dot{E}_i^o) + \sum_{\text{all}} \dot{E}_{\text{Steam,out}} + \sum_{\text{all}} \dot{I}_i \quad (4.48)$$

where $\sum_{\text{all}} \dot{E}_{\text{Steam,in}}$ and $\sum_{\text{all}} \dot{E}_{\text{Steam,out}}$ are all steam in- and outputs of the plant, respectively, \dot{E}_{air} is the exergy of the incoming air and $\sum_{\text{fr all}} \dot{E}_{\text{out,i}}^{\text{ph}}$ means the physical exergy of the fractions leaving the distillation plant, which are GAN, LIN, GOX, LOX and LAR as displayed in Figure 4.3. This physical exergy can be used in other processes. From (4.46) it is clear that the exergy used is:

$$\dot{E}_{\text{used}} = \sum_{\text{all}} \dot{E}_{\text{Steam,in}} - \sum_{\text{all}} \dot{E}_{\text{Steam,out}} + \dot{W}_{\text{freon}} \quad (4.49)$$

This leads with reference to (2.4) and with the aid of expressions (4.48) and (4.49) to the following expression for the desired exergy output:

$$\dot{E}_{\text{desired output}} = \sum_{\text{fr all}} (\dot{E}_{\text{out,i}}^{\text{ph}} + \dot{E}_i^o) - \dot{E}_{\text{air}} \quad (4.50)$$

Using (2.3), the rational efficiency for the whole air separation plant is given by:

$$\Psi_{\text{plant}} = \frac{\sum_{\text{fr all}} (\dot{E}_{\text{out,i}}^{\text{ph}} + \dot{E}_i^o) - \dot{E}_{\text{air}}}{\sum_{\text{all}} \dot{E}_{\text{Steam,in}} - \sum_{\text{all}} \dot{E}_{\text{Steam,out}} + \dot{W}_{\text{freon}}} \quad (4.51)$$

or from (2.5)

$$\Psi_{\text{plant}} = \frac{\sum_{fr\ all} (\dot{E}_{\text{out},i}^{\text{ph}} + \dot{E}_i^{\text{o}}) - \dot{E}_{\text{air}}}{\sum_{fr\ all} (\dot{E}_{\text{out},i}^{\text{ph}} + \dot{E}_i^{\text{o}}) - \dot{E}_{\text{air}} + \sum_{all} \dot{I}_i} \quad (4.52)$$

It could be argued that the desired product is the chemical exergy increase only as discussed in chapter 2.3.2. This would lead to the following formula for the whole plant

$$\Psi_{\text{plant}} = \frac{\sum_{fr\ all} \dot{E}_i^{\text{o}} - \dot{E}_{\text{air}}^{\text{o}}}{\sum_{fr\ all} \dot{E}_{\text{out},i}^{\text{ph}} - \dot{E}_{\text{air}}^{\text{ph}} + \sum_{all} \dot{E}_{\text{Steam,in}} - \sum_{all} \dot{E}_{\text{Steam,out}} + \dot{W}_{\text{freon}}} \quad (4.53)$$

4.3.4 Exergy analysis

Air compressor and cleaning unit

The main exergy loss, 1708 kW, is caused by the compression of the air to 6.2 bar as can be seen in Table 4.6. The exergy loss of steam turbine 1 (ST1), which delivers the required 4.3 MW of power for the air compressor (AC), is 708 kW (see Table 4.7). The consumption of steam of 400°C and 40 bar is 15.5 ton a hour. The rational efficiency of steam turbine 1 as calculated with (4.32) is 0.86. For cooling the air to 7°C the electricity use for the freon unit is 110 kW (\dot{W}_{freon}). Most of the exergy increase due to the cooling of the air is lost in the molecular filter, because the air is here heated to just below the environmental temperature. The molecular filters are cleaned by the hot purge gas. By the heating of the purge gas to 170°C with steam of 210°C and 11 bar and dumping the gas into the environment after the cleaning of the filters 225 kW of exergy is lost. The rational efficiency of the air compressor and cleaning unit is calculated to be 0.48 with formula 4.37.

Table 4.6 The in- and outgoing flows of the air compressor (AC)

Components	Temp. (in °C)	Press. (in bar)	Flow ¹⁾ (kg/sec)	Enthalpy (in kW)	Exergy (in kW)
AIR	20.0	1.01	16.39	-2638.9	0.7
Work input				4334.9	4334.9
Heat transfer to envir.				-3362.6	0
AIRP	79.4	6.20	-16.39	1666.7	-2627.8
Total				0.1	1707.8

¹⁾ The minus sign means the stream is leaving the AC

Table 4.7 The in- and outgoing flows of steam turbine 1 (ST1)

Components	Temp. (in °C)	Press. (in bar)	Flow ¹⁾ (kg/sec)	Enthalpy (in kW)	Exergy (in kW)
Steam in (ST1in)	400.0	40.00	4.31	-54921	5497.3
Steam out (STout)	29.6	0.04	-4.31	59256	-454.1
Work output				-4335	-4334.9
Total				0	708.3

¹⁾ The minus sign means the stream is leaving the ST1

Main heat exchanger (MHE)

The exergy losses are caused by temperature difference between the hot and cold streams and pressure losses. The mean temperature difference is 4.2 K. The rational efficiency is 0.86 as calculated by (4.33b), where AIRIN is the hot stream. In Figure 4.3 INA and INB are displayed as one stream (IN). In reality M1 is split in the main stream INA and a small stream INB in the main heat exchanger (MHE) and combined at constant temperature after heat exchanger 1 (HE 1). The exergy calculations take into account this split as can be seen in Table 4.8.

Table 4.8 The in- and outgoing flows of the main heat exchanger

Components	Temp. (in °C)	Press. (in bar)	Flow ¹⁾ (kg/sec)	Enthalpy (in kW)	Exergy (in kW)
AIRIN	20.0	6.20	16.20	-110.1	2567.6
GOXC	-178.1	1.67	2.82	-527.4	81.3
Purge cold (PC)	-180.6	1.43	3.49	-743.8	665.9
M1	-177.5	5.85	3.60	-795.1	1195.1
GANC (GC)	-182.0	1.40	6.20	-1349.6	1292.2
AIRC	-172.6	5.85	-16.20	3373.5	-4688.0
GOX	17.0	1.47	-2.82	21.9	-426.5
INA (IN)	17.0	5.75	-3.50	34.9	-626.3
INB (IN)	-84.0	5.75	-0.10	11.7	-20.7
GAN	17.0	1.30	-6.20	53.9	-293.4
PURGE	17.0	1.43	-3.49	30.1	-145.6
Total				-0.3	333.2

¹⁾ The minus sign means the stream is leaving the MHE

Distillation unit

Because of its complex nature no complete flow scheme, including all stream data, has been given in this chapter. The incoming and outgoing flows of the different components

are given in Appendix 2. In Table 4.9 it can be seen how the different components operate by showing the exergy losses and rational efficiencies. The heat transfer between the reboiler of the upper column and the condenser of the lower column cause an exergy loss of 82 kW due to the temperature difference of 1.1 K. This results according to (4.33b) in a rational efficiency of 0.98. The throttling valves (JT1 and JT2) between the columns causes an exergy loss of 125 kW. The exergy loss in the lower column (LC) is 62 kW and the exergy loss in the upper column (UC) and crude argon column is 487 kW. Of the exergy loss in the crude argon column 108 kW is caused in the condenser by the temperature difference of 2.8 K between the coolant and the top of the column. The chemical exergy increase of the products of the lower and upper distillation column is 242.7 kJ and 428.5 kJ, respectively. Using (4.41) and (4.45) the rational efficiency of the lower and upper column were found to be 0.80 and 0.92, respectively. The high efficiency of the upper column is caused by the high exergy transfer from its reboiler to the condenser of the lower column. An exergy loss of 32 kW takes place in the heat exchanger (not displayed in Figure 4.3), which cools the products of the lower column, B and D, before going into the valves which lead to the upper column. Using (4.33b) a rational efficiency of 0.88 is obtained for this heat exchanger. The rational efficiency of the distillation section is 0.46 as calculated by (4.46) with a chemical exergy increase of 671.2 kJ.

Table 4.9 Results of the exergy analysis in the distillation unit

Unit	exergy losses (in kW)	rational efficiency
heat exchange reboiler/condenser	82	0.98
Throttling valves columns	125	--
Lower distillation column	62	0.80
Upper distillation column	487	0.91
Heat exchanger	32	0.88
Total	788	0.46

Liquefaction unit

Because of its complex nature the incoming and outgoing flows of the different components are given in Appendix 2. In Table 4.10 it can be seen how the different components operate by showing the exergy losses and rational efficiencies. For the compression of the nitrogen to 30 bar a five stage compressor with intercooling (C1) is used. Steam turbine 2 (ST2), which delivers the required 6.0 MW for the compressor, uses 20 ton steam a hour of 420°C and 40 bar. A capacity mismatch between compressor 3 (C3) , which compresses to 46 bar, and the hot expansion turbine (HET) results in a exergy loss of 344 kW. The relative low efficiency of the turbines is caused by the fact that the expansion of the flows takes place below the environmental temperature. The temperature difference between the isentropic temperature and the real outlet temperature leads to a great exergy loss due to the low temperature. Especially, in the

case of the cold expansion turbine (CET), where the outlet temperature is $-177\text{ }^{\circ}\text{C}$. The throttling process in the valve (TV) causes a temperature decrease of 3.4 K . The low exergy loss in the valve is achieved because the flow enters the valve in the liquid phase and leaves the valve still almost completely liquid.

Table 4.10 Results of the exergy analysis in the liquefaction unit

Unit	exergy losses (in kW)	rational efficiency
Five stages compressor (C1)	1965	0.67
Steam turbine	1021	0.86
Compressor 2 (C2)	205	0.70
Compressor 3 (C3)	141	0.69
Capacity mismatch	344	----
Hot expansion turbine (HET)	241	0.77
Cold expansion turbine (CET)	428	0.61
Heat exchanger 1 (HE 1)	256	0.72
Heat exchanger 2 (HE 2)	173	0.90
Mixers	5	----
Throttling valve (TV)	74	----
Total	8810	0.25

Argon purifying unit

The exergy loss in the heat exchanger (H.E.) and compressor is 1.2 kW and 2.9 kW , respectively. The deoxygenation of the stream with hydrogen with an exergy of 65 kW causes an irreversibility of 27 kW . The exergy loss taking place due to the cooling of argon flow from 950°C to 15°C leads to an exergy loss of 43 kW . The heat is not used somewhere else. The exergy loss in the pure argon distillation column is 10 kW . A purge stream of 0.8 kW leaves the column at the top. The rational efficiency of this unit is 0.017 according to (4.46). No rational efficiencies are calculated for the separate components due to the small exergy losses of these components.

Total

The overall exergy loss is 8810 kW as can be seen in Table 4.11. The total exergy of the products is 3514 kW , resulting in an overall rational efficiency of 0.28 according to (4.51). The efficiency is so high due to the great physical exergy of the products of 2765 kW , mainly the physical exergy of the liquid products, which is 2602 kW . If only the chemical exergy of the products is seen as the desired output, the rational efficiency becomes 0.071 by using (4.53).

Table 4.11 Results of the exergy analysis

Unit	exergy losses (in kW)	rational efficiency
Air compressor and cleaning	2751	0.48
Main heat exchanger	333	0.86
Distillation unit	788	0.52
Liquefaction unit	4853	0.25
Argon purifying unit	85	0.02
Total	8810	0.28

4.3.5 Results

More than half of the exergy loss is taking place in the liquefaction unit, while almost one third is lost in the air compression unit. Minor exergy losses are taking place in the distillation unit and heat exchangers. The greatest exergy losses are caused by the compressors and to a lesser extent by the turbines. The relatively low rational efficiency of the turbines operating in the cryogenic region is striking, while the efficiency of the heat exchangers is very high.

4.3.6 Improvements

By using the excess work in the case of the capacity mismatch 344 kW can be saved. The increase of the polytropic efficiency of the air compressor (AC) from 0.70 to 0.85 will decrease the power use by 880 kW. This will give an associated exergy saving in steam turbine 1 (ST1) of 139 kW. The increase of the polytropic efficiency of the nitrogen compressor (C1) from 0.75 to 0.85 leads to a decrease of the power use by 759 kW. It gives also an associated exergy saving of 108 kW in steam turbine 2 (ST2). The rational efficiencies of both compressors (AC and C1) becomes 0.77. The increase of the polytropic efficiency of 0.83 of the cold expansion turbine (CET) to the same value as the hot expansion turbine (HET), will give an exergy saving of 75 kW and results in a rational efficiency of 0.66 for the cold expansion turbine.

The rational efficiency of the heat exchangers is high to very high. So the possibilities for exergy saving are very limited. Only the exergy loss in heat exchanger 1 (HE 1) of the liquefaction unit could be halved by improving the heat integration. This would lead to an exergy saving of about 125 kW.

4.3.7 Conclusions for the cryogenic air separation

The exergy analysis of the cryogenic air distillation plant has pinpointed and quantified the exergy loss in the different plant sections. The greatest exergy loss is caused by the compressors. This exergy loss can be reduced by almost one half by using better

compressors. Including other suggested improvements the exergy loss can be reduced by 25%. However, the cryogenic air separation unit itself is well designed from an exergetic point of view and for a further improvement of the rational efficiency of air separation alternative processes have to be used or developed.

4.4 Comparison

Comparing the rational efficiency of the two distillation plants, it is shown that the air separation plant has a much higher efficiency, 29%, than the crude oil distillation plant (5.2%). However, the inputs are not comparable. The irreversibility due to the steam production is not taken into account. This will lead, assuming an exergetic efficiency of 50% for the steam production by a combined heat and power plant, to an additional irreversibility of 1.27 MW and 6.16 MW for the crude oil distillation plant and the cryogenic air separation plant, respectively. The rational efficiency of the two plants becomes then 5.0% and 19%, respectively. In the following chapters it will be shown how to deal with the comparison of processes with different types of inputs and outputs. From the comparison made it can be concluded that the cryogenic air separation plant is operating more efficiently than the crude oil distillation plant.

4.5 Conclusions

With the aid of exergy analysis the exergy destruction can be pinpointed and the degree of thermodynamic perfection of the processes can be determined. The mayor exergy losses are taking place in the utilities providing the required amount of heat and the cold. The rational efficiency of cryogenic air separation is higher than crude oil distillation. The possibilities of improvements in the case of crude oil separation are substantial, by using the heat of a combined heat and power plant and the low quality heat of the process the irreversibilities can be minimised. In the case of cryogenic air separation they are more limited. Alternative concepts have to be developed for more improvement in the case of air separation.

4.6 Nomenclature

E, ε	= exergy, molar exergy
GAN	= gaseous nitrogen
GOX	= gaseous oxygen
LAR	= liquid argon
LIN	= liquid nitrogen
I	= irreversibility
P	= pressure
R	= universal gas constant
T	= temperature

ψ = rational efficiency

Subscripts

AC = air compressor
ACC = air compressing and cleaning unit
AR = ATM-RES = atmospheric residue
C = crude oil
CDU = crude distillation unit
CET = cold expansion turbine
CH = CRUDEH (crude heated)
cold = cold stream
CRAR = crude argon
cold = cold stream
cond = condenser
DIST = distillation section
GAS = fuel gas
HE = heat exchanger
HEN = heat exchanger network
HET = hot expansion turbine
hot = hot stream
fr = fractions
hot = hot stream
i = i-th item
in = incoming
JT = Joule-Thomson valve
LC = lower distillation column
LIQ = liquefaction unit
min = minimum
mix = mixture
out = outgoing
PA = pure argon distillation column
PC = purge cold
PH = purge hot
S = STEAM
Scf = spray cooler
ST = steam turbine
TN = top nitrogen
UC = upper distillation column
VDU = vacuum distillation unit
W = WATER

Superscripts

0 = environmental state, chemical component

ph	= physical component
Q	= heat
ΔT	= thermal component
•	= rate with respect to time

4.7 References

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Exergy analysis and Life Cycle Assessment

Abstract

Environmentally orientated Life Cycle Assessment (LCA) has been developed in the last decade to analyse environmental problems associated with the production, use and disposal or recycling of products or product systems. A comprehensive frame work has been established to enable the execution of the analysis. In this framework the concept of exergy or similar thermodynamic concepts have not yet been introduced. It is shown in this chapter that for sustainable development the depletion of exergy reservoirs and the exergetic emissions to the environment have to be minimised. To accomplish this it is proposed to extend the LCA with an Exergetic Life cycle Analysis (ELCA). The method of how to perform an ELCA is presented. An extension to the method of ELCA, the Zero-ELCA, is presented in which emissions are translated in terms of exergy. The concepts of a zero-exergy and a reduced exergy emission process are introduced. It is shown that in the case of a zero exergy emission the LCA can be replaced by an ELCA.

5.1 Introduction

To determine the thermodynamic perfection of a system, not only processes that occur within the system should be considered to be taken into account, as is the case in the preceding chapters, but also all kinds of interaction between energy and material flows outside the system's boundaries. By doing so the actual performance of the system and its impact on the environment can be evaluated. The exergy consumption during the complete life cycle allows to evaluate the degree of thermodynamic imperfection of the production processes and to conduct the assessment of the whole process chain associated with the product.

The first step of analysing the life cycle on the basis of exergy has been taken by Szargut (1987). It is called the analysis of Cumulative Exergy Consumption (CExC), which is the cumulative exergy use⁷ from the extraction of natural resources to the final product. Several studies have been performed with this method (Szargut et al., 1988b). This method has a strong resemblance with the method of calculating the Gross Energy

⁷ Exergy consumption is synonym to exergy use in the concept of CExC, so the exergy content of the product is included in the CExC

Requirement (GER) as proposed by Slesser (1974). It also uses the four levels for the analysis. Mostly, only level one and two, the direct or process energy use and energy for the production of materials, are taken into account. Instead of energy exergy is used in the CExC. In the CExC only the production and use of the product or material are taken into account. The exergy consumption is accumulated to a defined point in the life cycle. However, in this method the exergy destruction associated with the disposal of the product and the influence of recycling which cause changes in the exergy destruction are not taken into account.

Finnveden and Östlund (1997) proposed to use exergy analysis in the life cycle assessment. To work out their proposal they have calculated the exergy used to produce one kilogram of product. However, instead of taking the exergy destruction, as they originally proposed, they have calculated the exergy content of the required ores for the production of the products. Stougie et al. (1994) have performed a study to relate exergy loss with different environmental effects. Their conclusion is that there is a qualitative relation, but that more research is needed to determine a quantitative relation. Blonk et al. (1997) have analysed the feasibility of making operational the depletion of abiotic resources in LCA via the key resources energy and land. They investigate the use of exergy loss as an indicator. However, one of their criteria for the characterisation was that the indicator should be independent of the location and the way of refining of materials. This is not the case with exergy analysis and they concluded that the exergy loss approach was not useful as an indicator within this framework.

An excellent bibliography about the use exergy as a measure of resource flows and reserves is written by Ayres et al. (1996). In general we can state that exergy analysis and life cycle analysis have been developed separately. Life cycle assessment (LCA) has been described by Consoli et al. (1993) and Heijungs et al. (1992). They give a detailed methodology how to perform an LCA. The methodology includes the effects of all the phases (production, use and recycling) on the environment. With the LCA an improvement analysis is connected to minimise the environmental burden of the analysed products. In this thesis it will be shown how the tool LCA can be made more powerful if the LCA is extended with an exergetic life cycle analysis (ELCA) and a Zero-ELCA.

5.2 Environmental Life Cycle Assessment

The LCA uses the cradle to grave approach. The complete life cycle of the product is taken into account. According to the frame work established by the Society of Environmental Toxicology and Chemistry as described by Consoli et al. (1993) and Guinée (1995) four stages in the LCA can be distinguished.

1. In the goal definition and scoping the subject of study is determined in relation to the application intended. The functional unit of the product, the spatial scale and time horizon have to be determined.

2. In the inventory analysis the complete life cycle of the product is analysed leading to the inventory table, a list of inputs from and outputs to the environment.
3. The impact assessment is a stepwise process. In the first step, the classification, it is determined which environmental effects are considered and which extractions and emissions contribute to it. In the second step, the characterisation, the contribution of all extractions and emissions to the selected environmental problems is estimated. For a better interpretation the impact of the environmental effects can be normalised to the actual impact in some given area. The final step is the valuation of the selected environmental problems.
4. The last stage of the LCA is the improvement analysis.

In the present LCA framework the depletion of abiotic resources is calculated according to Heijungs et al. (1992) by the use of material i (M_i) divided by an estimate of the recoverable reserve of material i (R_i) or in formula form

$$abiotic\ depletion = \sum_i \frac{M_i}{R_i} \quad (5.1)$$

In the case of minerals its usefulness is determined by its atomic structure. So the material reserve will always stay constant, because the atomic structure of the minerals can not be lost⁸. Only the concentration and chemical structure of the minerals can change. Hence, with the use of sufficient exergy it is always possible to obtain the desired concentration and chemical structure for the minerals. Because the material reserve stays constant, i.e. it can never be depleted, there is no relation between material use and reserve. In formula 5.1 the recoverable material reserve is used. However, this reserve is dependent of the market price, because at a higher price more financial efforts can be taken to recover the material. This makes the meaning of formula (5.1) as a criterion for depletion of minerals less relevant.

In the case of (fossil) fuels its usefulness is determined by work potential or better said exergy. This is lost after its use in the most common case of burning. So the material reserve of fuels will decrease by its use in the case of burning. However, the material reserve will also decrease in the case of non-energetic use of fossil fuels, for example the production of plastics. The work potential is then lost only partly. However, according to formula (5.1) this gives this same depletion as in the case of burning, which is not justified because after the plastic production, the plastics can still be used as a fuel. It can be concluded that in the case of (fossil) fuels formula 5.1 is not an appropriate way to describe depletion. A new criterion for the depletion of natural resources is introduced, which displays the relation between the use of minerals and the depletion of fossil fuels and is based on the work potential of materials.

The impact assessment of the LCA results in a list of environmental effects, which are completely different. This leads to the fact that in the improvement analysis often a trade

⁸ Except for the special case of nuclear reactions. These reactions are not taken into consideration.

off has to be made between the different environmental effects. To solve this question different approaches are used, like the Eco-indicator 95 (Goedkoop, 1995), which lead to one indicator for all environmental effects. The problem associated with the use of this single indicator is that the valuation of the weight of the different environmental effects in this single indicator is a subjective process. In this chapter will be discussed how exergy can be part of the solution to this problem.

5.3 Exergy and sustainability

Up until now the exergy concept has not been used in the LCA and is not mentioned in the framework suggested by Consoli et al. (1993). This can be seen as an omission in the LCA, because there is a direct relation between exergy and the environmental problem of depletion of natural resources. The exergy concept can improve the traditional LCA.

For sustainable development the destruction of the exergy reservoirs of natural resources has to be minimised to a level at which there is no damage to the environment and at which the supply of exergy to further generations is secured. It is not necessary to distinguish between the different kinds of materials as shown in (5.1). For example, if the exergy reservoir of high concentration lead ore is depleted the exergy reservoir of fuels and other materials required can be used to explore the lower concentration lead ore. One could argue that a distinction between abiotic and biotic resources should be made. However, in many situations in the present society abiotic resources are used to obtain biotic resources, for example the use of fertilisers and green houses to produce food. So to make a distinction between these two types of resources becomes less meaningful. To clarify this point more research has to be carried. However, for the sake of simplicity no distinction is made between abiotic and biotic resources in this thesis. Exergy destruction is used as single criterion for the depletion of natural resources.

To minimise the use of natural resources an exergy analysis can be used as shown by Szargut (1978). In an exergy analysis it is shown where the work potential of natural resources in relation to the surrounding environment is lost, i.e. where the irreversibility or exergy destruction takes places. The maximal obtainable potential of work of a flow can only be obtained via reversible processes as shown by process 1 in Figure 5.1. The figure illustrates that there is no emission to the environment and no depletion of resources in a reversible process. In the real world, processes are irreversible and this is shown in process 2 in Figure 5.1. The process shows irreversibility and emission in terms of exergy.

Exergy can also be used as a more general criterion in a world, where more emissions can be prevented or separated and transformed to harmless waste or useful products. To carry out these prevention or separation and transformation processes, which like all real processes are irreversible, an exergy destruction or irreversibility will be created. When all exergetic emissions are handled by these processes the environmental burden of

products can be compared by only the criterion of irreversibility as shown in process 3 in Figure 5.1.

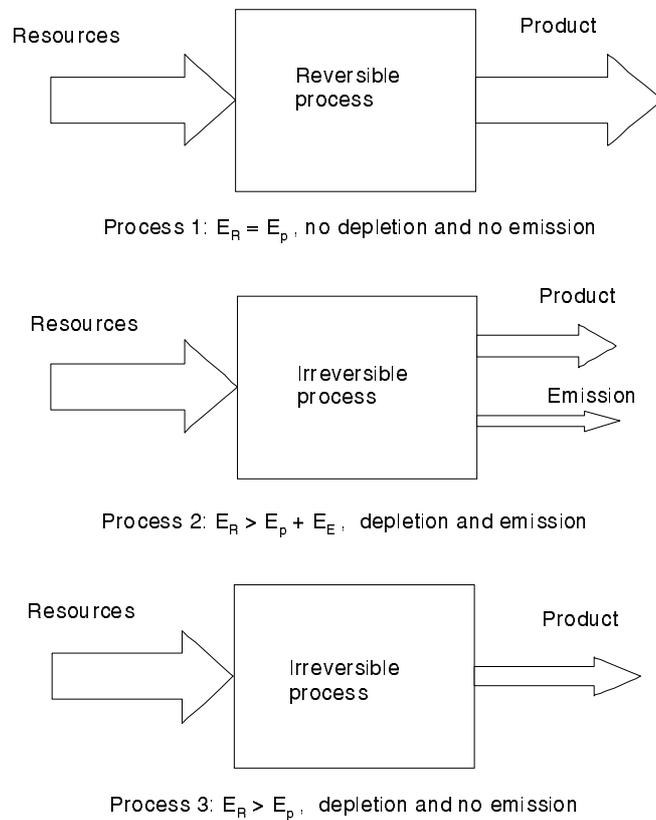


Figure 5.1 Reversible and irreversible processes

Zero-exergy emission processes have no emissions or emissions containing zero exergy, which are harmless to the environment, because their chemical composition and physical properties are identical to the environment itself. So, in the case of a zero-exergy emission process the LCA can be completely replaced by an ELCA.

5.4 Exergetic life cycle analysis (ELCA)

The framework of the ELCA and the LCA are similar. The goal definition and scoping of the LCA and ELCA are completely identical. The inventory analysis of the ELCA is more extensive. A complete flowsheet of the mass and energy streams of the different production steps is required. The material and energy balances have to be closed. This is not always the case in a LCA. Also the more simplified black box approach can be used of which only the inputs and outputs of the production processes are taking into account. The impact assessment is limited to calculation of the exergy of the flows and the determination of the exergy destruction in the different processes. There is no classification in the ELCA. For the calculation of exergy the conditions and composition of the environment have to be specified. For processes where no location is specified it is

recommended to use the standard state established by Szargut et al. (1988a). The cumulation of all exergy destruction in the life cycle gives the life cycle irreversibility of the product. The improvement analysis is the minimisation of the life cycle irreversibility.

In the case of production of co-products, combined waste processing, and open-loop recycling an allocation method has to be used. Three methods are suggested. The most straight forward method is the distribution of exergy destruction on the basis of the exergy of the flows. The second method is the distribution of the exergy destruction on basis of the exergy destruction in the case of separate production of the by-products or handling of the by-waste as suggested by Szargut (1988). However, the limitation to the use of the last-mentioned method is that there always has to be a distinction between the main product and the by-product or waste and that separate production of the by-product or handling of the by-waste is taking place. Furthermore, the allocation of exergy destruction to products on basis of their separate production can lead to negative exergy losses or otherwise stated exergy savings. Therefore, it is suggested to use separate production only as a basis for comparing combined and separate production. The third method, the marginal cost method, is the distribution of exergy destruction on basis of changes in the exergy values of the flows. Starting from zero input, if possible, each addition to the exergy destruction is allocated to the marginal increase in the exergy value of the flows.

The exergy analysis pinpoints the places where the exergy destruction takes place and in the improvement analysis different possibilities can be presented to minimise the life cycle irreversibility, i.e. the irreversibility during the life cycle of the product or product system. The improvement analysis can be extended with an exergo-economics analysis in which also the monetary costs are taken into account. Examples of this type of analysis are given by Lozano and Valero (1993). When in the inventory analysis a black box approach has been used only different alternatives for the black boxes can be compared in the improvement analysis.

To perform an ELCA the assistance of advanced flow sheeting computer programs with the possibility of calculating the exergy of flows, like Aspen Plus in connection with Exercom (Lie et al., 1994) can facilitate the work greatly. The most ideal situation is to combine the design of the process plant with an exergy analysis. Then the irreversibility of the process plant can be taken into account immediately.

5.5 Zero-exergy emissions processes

In the special case of zero-exergy emission production processes the LCA can be completely replaced by the ELCA as discussed in 5.3. However, not always a zero-exergy emission is necessary for sustainable development. A process is considered to be a sustainable process when the emission rate of the process is below the acceptance level

of the environment or below the allocated⁹ emission rates based on these levels. For the Dutch situation these levels are estimated by Weterings et al. (1992) for the main harmful emissions. The present emission rates analysed were 50% to 90% above these acceptance levels. So instead of zero-exergy emission processes the sustainable emission levels could be taken, which lead to reduced exergy emission processes, i.e. processes where the exergy emission is reduced to the sustainable emission levels. However, for the sake of simplicity we limit ourselves to zero-exergy emissions processes as the finite goal in this thesis.

There is still one special point to be dealt with the reduced exergy emission. Can heat transfer with nearly environmental temperature and a very low exergy content increase the average temperature of the earth? Rough calculations with a mean radiation input from the sun of 185 W/m^2 (ODUM, 1979) shows that in one month more energy of the sun reaches the earth surface than the total reservoir of fossil fuels of $2.6 \cdot 10^{23}$ Joules (Weterings et al., 1992). So, it is assumed that the increase of the mean temperature of the environment by the release of waste heat is negligible. This leads to the conclusion that heat transfer to the environment is no threat to sustainable development in general and there is no need to pay special attention to it. However, for the influence of heat transfer to the local environment exergy can be considered as the best criterion. A higher exergy content of heat means a higher temperature difference with the environment, which will result in a greater impact on the local environment.

An example of a zero-exergy emission process of electricity production is an oil fired power plant of which the exergy containing exhaust gases of the plant are injected into the oil wells to enhance the oil production as shown by Wall et al. (1995). Of this example the environmental effects associated with the building and disposal of the power plant are left out and the oil fields are assumed to be inside the system boundaries.

5.6 Zero-exergy emission ELCA (Zero-ELCA)

The ELCA can be extended with the abatement exergy of emissions¹⁰. In principle the non zero-exergy emission processes can be adjusted to zero-exergy emission processes by separation and transformation of the emissions. This separation and transformation will consume exergy. So an amount of exergy use can be assigned to the different emissions, which we call the abatement exergy. By this method the ELCA is extended to take into account environmental effects associated with emissions and not only depletion of natural resources.

The Zero-ELCA can also be adjusted to the reduced exergy emission ELCA. This concept can be used to equalise the emissions per functional unit for different production

⁹ The allocation of emission rates is a political or economical process. If and in which way the allocation should take place is beyond the scope of this thesis

¹⁰ In normal practise, abatement will be preceded by prevention so that only those emissions that could not be prevented will be abated.

routes and designs. So, the life cycle irreversibility can be used as the sole criterion for comparing production routes and designs, because all other environmental effects will be similar. Zero-exergy emission rates can be exchanged for emission rates which can be neutralised by the environment, i.e. emission rates under which sustainability can be reached (Weterings et al., 1992) as discussed before.

A different method has been proposed by Szargut (1985). In this method coefficients have been defined which relate the exergy of harmful emissions to the exergy destruction of useful products and natural resources. This method is preferable from a theoretical point of view, because there is a direct relation between exergy emission and the damage in terms of exergy. However, the practical difficulties to obtain such a database of coefficients are expected to be great and so this method is not developed further in this thesis.

5.7 Conclusions

The present criterion for the depletion of natural resources in the environmentally orientated LCA is not appropriate. As criterion for the depletion of natural resources exergy destruction should be used. In order to achieve this the present LCA has to be extended with an ELCA. In the case of zero-exergy emission processes the LCA can be omitted and replaced by the ELCA. The ELCA can be extended with the Zero-ELCA to take into account also all environmental problems associated with emissions by accounting for the abatement exergy of emissions. This approach can also be used for the comparison of different emission processes on the basis of exergy.

5.8 Nomenclature

E = exergy
 M_i = use of material i
 R_i = recoverable reserve of material i

Subscripts

E = emission
P = products
R = resources

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Exergetic Life Cycle Analysis of the Conventional Textile Washing Process

Abstract

In this chapter an ELCA for a production process is performed: the conventional textile washing process. Special attention is devoted to the improvement analysis: the minimisation of the life cycle irreversibility of the washing process. In order to minimise the life cycle irreversibility some experiments have been performed to quantify the exergy consumption of the different process steps that occur during the washing process. These experiments show that the electrical heating process consumes most exergy. Hence, different alternatives of this process are proposed.

The washing performance is related to the washing temperature and the detergent quantity, by performing a number of experiments. With this relation an optimum combination of washing temperature and detergent quantity can be found for the different heating systems, that results in an acceptable washing performance with a minimal life cycle irreversibility. It is shown that replacing the conventional electrical heating system by heating by natural gas or district heating will reduce the life cycle irreversibility by 1/3 and more than a 1/2, respectively.

6.1 Introduction

In the Netherlands washing machines account for 6.3 percent of the electricity consumption in households. Their market penetration is 97% (Weegink, 1996). It is, therefore, of interest to analyse the possibilities of more efficient washing machines. In order to examine the textile washing process in a complete way, an ELCA is performed. This is done by the approach in which the different processes are described by block boxes. No detailed inventory analysis is performed, only the exergy flows are taken into consideration. In this manner the total exergy use for the processes is obtained and the life cycle irreversibility can be determined. The optimised process with electrical heating is compared with the proposed alternatives with the same washing performance. A picture of the process steps of the washing process is given in Figure 6.1.

The process steps within the marked area of this figure are examined and optimised to obtain the lowest life cycle irreversibility. Thus, the performance of the washing process and the inputs and outputs associated with it are studied. The exergy use of most inputs is determined by help of the Cumulative Exergy Consumption (CExC). The irreversibilities due to the manufacture of the washing machine has been left out in this study, but can be expected to be negligible.

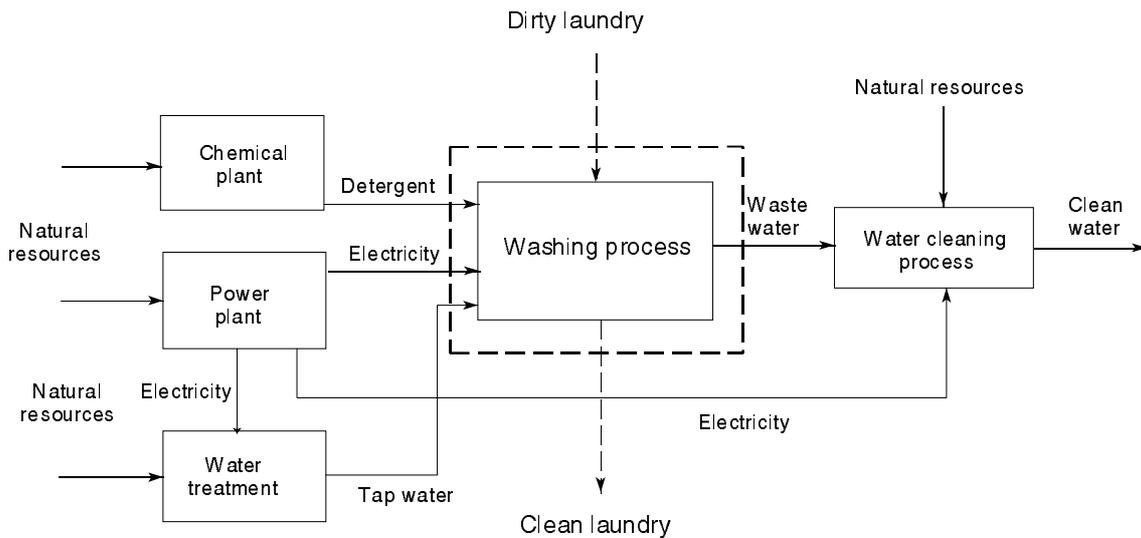


Figure 6.1 System diagram of the washing process

It is emphasised that the exergy rise due to soil removal from textiles is extremely low, due to the small quantity of soil to be removed. The exergy rise is, therefore, neglected in this study and no exergetic efficiencies of the washing process have been calculated. So no minimum exergy use for the ideal situation, i.e. reversible processes, could be determined.

6.2 Experiments

6.2.1 Set-up

Five variables that influence the washing performance can be distinguished. These are: temperature, detergent quantity, washing time, agitation and water input. From the viewpoint of minimising the life cycle irreversibility at a required washing performance only the temperature and detergent quantity are analysed. It can be argued that a lower water input is preferable for minimising the life cycle irreversibility, however, the water input is already quite low and the possibilities of reducing it are limited. Therefore, the water input has been kept constant. Higher agitation than used in the normal washing program is assumed not to have a large influence, because the mixing is already complete. From thermodynamic considerations a longer washing time is preferable. However, from the users viewpoint the washing time must not take longer than it already

is. So this variable is also kept constant. The relation of the washing performance depending only on the washing temperature and the detergent quantity is used to obtain the optimal combination of temperature and detergent with a minimal life cycle irreversibility at the required washing performance.

The experiments were carried out with a Philips Whirlpool 1200 Silent. This is a front loading drum type washing machine. During the entire washing process measurements were made of the electric power input, the temperature of the washing fluid and the water input. The exergy used by each process i.e. heating, agitation and pumping + centrifugation, were determined at the washing programs of 40, 60 and 80°C. The actual washing performance was tested in another set of experiments. These experiments were performed at 30, 40 and 60°C, with detergent quantities of 18, 36, 50, 72, 90 and 108 grams. The washing time was set at 35 minutes for each experiment. The incoming tap water temperature was 15°C. In this set of experiments six different test pieces were washed in duplo, together with 3.4 kg of dummy towels. Sensitivity of the test pieces for the different components in the detergent are given in Table 6.1.

Table 6.1 Sensitivity data of the different test pieces

Test piece	Contains	Sensitive for
1	Particulate soil ¹⁾	Builder / actives
2	Same as 1+ casein, milk	Same as 1 + enzymes
3	Tea	Bleach
4	Olive oil + ink	Actives / enzymes
5	Sebum ²⁾ + ink	Actives / non-ionics
6	Pig blood	Enzymes

1) Iron oxide, silicate, cotton oil and some emulsifier

2) Sweat

6.2.2 Results

For all washing programs the water input for the washing cycle was equal to 17 litres. The results of the electrical power consumption of each separate process will be presented in the following figures and tables. The electrical power input for a washing program of 60°C is shown in Figure 6.2. The electrical power input used by the heating process can clearly be distinguished in this figure as large blocks. The power fluctuations during the entire washing process are caused by the agitation of the drum. The smaller blocks on the right hand side of Figure 6.3 are the electrical power used for pumping and centrifugation. By integrating the power over time, the electricity or exergy used by each process can be calculated. This is presented for each washing program in Table 6.2.

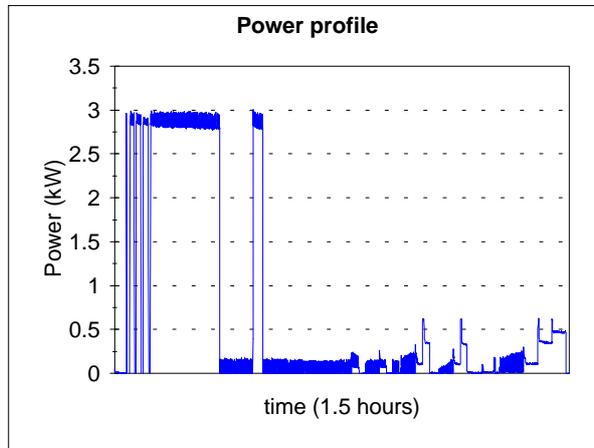


Figure 6.2 Power input profile at a 60°C washing program

Table 6.2 shows that the exergy use due to the heating process ranges from 78 to 94 percent of the total amount of exergy use for the different programs. The increase in exergy use of the agitation process at the 80°C program is explained by the longer rinsing program.

Table 6.2 Exergy use of the different processes for the washing programs

Washing program	40°C (kJ)	60°C (kJ)	80°C (kJ)
Heating	1450	3440	6400
Pumping + centrifugation	300	290	270
Agitation	100	100	140

From different experiments including those presented in Table 6.2 a linear relation between the exergy use by the heating process and the washing temperature has been obtained, which is

$$E_{hp}(T) = -1.6 + 0.09 \cdot T \quad (6.1)$$

where E_{hp} is expressed in MJ and T lies between 30 and 80°C.

To study the cleaning performance of the washing process, a reflection meter was used. This meter measures the reflection of a beam of light aimed on the test piece. Data of the reflection before and after the washing process can be converted into a dimensionless factor called detergency. This factor can be considered as an indicator of the percentage of soil removal. By performing the experiments at the washing temperatures and detergent quantities as stated in the experimental set-up, a relation of the detergency for each of the test pieces depending on washing temperature and detergent quantity could be found. The relations for the different test pieces are shown in Appendix 3.

6.3 Optimisation

To minimise the life cycle irreversibility of the heating process, the present electrical heating system and different alternatives are analysed. These are a gas fired heating system and a district heating system.

6.3.1 Heating systems

Electrical heating system

In order to obtain the CExC of the electrical heating system, the exergetic efficiency of electricity production has to be determined. As an example the data of the ‘Amercentrale’ power plant are used, which has an electrical efficiency of 41.3% on the lower heating value. Taking into account that the exergy content of the coal is 1.09 times higher than the lower heating value (Szargut et al., 1988), the exergetic efficiency of this power plant is equal to 37.9%. This is approximately equal to the average exergetic efficiency of the Dutch power plants including transportation losses.

By dividing equation 6.1 by this exergetic efficiency, equation 6.2 is obtained, which expresses the cumulative exergy consumption in MJ as a function of the installed washing temperature,

$$CExC_{hp}^{ch}(T) = -4.3 + 0.24 \cdot T \quad (6.2)$$

Gas fired heating system

The energetic efficiency of the boiler on natural gas is assumed to be 85 percent on the lower heating value. Again the ratio between exergy content and lower heating value has to be known to obtain the CExC. For natural gas this value is equal to 1.04 (Szargut et al. 1988). Using this data in equation 6.1 results in the following relation between the CExC (in MJ) and installed washing temperature for the gas fired heating system.

$$CExC_{hp}^{ch}(T) = -2.0 + 0.11 \cdot T \quad (6.3)$$

Equation 6.3 shows that the CExC of the gas fired heating system is more than half of the value of the electrical heating system. This is because the heat of combustion is directly used to heat up the water.

District heating system

In order to obtain the CExC relation of the district heating system, we use the data of the before mentioned ‘Amercentrale’. This power plant also provides heat to greenhouses and buildings in its surroundings. It is able to supply 350 MW of thermal power at 80 to

130°C at the cost of 50 MW electrical power (EPZ, 1989). The heat loss due to the distribution is taken to be 50 MW or 14.3% of the total heat supply, which is about the average heat loss in heat distribution systems as shown in chapter 8. So the exergy use for 300 MW of heat supplied to the consumers is equal to 132 MW. This exergy use is calculated by determining the exergy use for the production of 50 MW electrical power with an exergetic efficiency of 37.9%. By multiplying equation 6.1 with the CExC of the supplied heat as stated above, equation 6.4 is obtained

$$CExC_{hp}^{dh}(T) = -0.7 + 0.04 \cdot T \quad (6.4)$$

This is six times better than the electrical heating system and three times better than the gas fired heating system.

6.3.2 Determination of the life cycle irreversibility

In order to determine the life cycle irreversibility of the washing process two factors have yet to be determined i.e. the CExC per unit mass of detergent and the CExC of water supply and water treatment per unit volume.

The CExC per unit mass of detergent is taken from Zhao et al. (1996). This value is equal to 46.8 MJ/kg. The CExC of water supply ($V_{w,tot}$) is taken to be 5.98 kJ/l according to Boom (1994). This value is almost equal to 6.25 kJ/l, which is estimated by Zhao et al. (1996). The CExC of treatment of waste water ($V_{w,ws}$) is $1.09 \cdot c + 0.57$ kJ/l, where c is the concentration of the detergent in g/l (Zhao et al., 1996). This gives the following irreversibility for the water treatment

$$I_{wt}(V_{w,tot}, D) = 5.98 \cdot V_{w,tot} + \left(1.09 \cdot \frac{D}{V_{w,ws}} + 0.57 \right) \cdot V_{w,ws} \quad (6.5)$$

where c is replaced by $D/V_{w,ws}$. Experiments show a water consumption of 85 litres ($V_{w,tot}$) for the complete washing process. The water consumption of the washing cycle only, i.e. without rinsing, is equal to 17 litres ($V_{w,ws}$). It is assumed that all detergent is removed after the washing cycle, so only this quantity of water has to be cleaned in the water cleaning process¹¹. Inserting these values in (6.5) results in the following relation for the irreversibility (in kJ) associated with water treatment.

$$I_{wt}(D) = 518 + 1.09 \cdot D \quad (6.6)$$

¹¹ The ideal situation has been assumed that the clean water leaving the washing process is not mixed with the dirty water.

From formula 6.6 it can be seen that the second term of the irreversibility of waste water treatment is of a lower order than that of the production of detergent, namely 1.09 MJ/kg against 46.8 MJ/kg. So this term of formula 6.6 is neglected.

With the average value of Table 6.2 for agitation, pumping and centrifugation to be 400 kJ per washing program and the exergetic efficiency of electricity production 37.9%, the CExC for these processes is calculated to be 1055 kJ per washing program. For the determination of the life cycle irreversibility in the case of an electrical heating spiral, equations 6.2 and 6.6, the CExC for agitation, pumping and centrifugation and the CExC for the detergent must be added. This results in formula 6.7.

$$I_{LC}^{eh}(T, D) = -2.8 + 0.24 \cdot T + 0.0468 \cdot D \quad (6.7)$$

where I_{LC}^{eh} is expressed in MJ. The formula is valid for T between 30 and 80°C and D between 18 and 108 gram.

6.3.3 Determination of the minimum life cycle irreversibility

Now that the relation for the life cycle irreversibility is known (equation 6.7) and also the relations for washing performance depending on temperature and detergent quantity as displayed in Appendix 3, a minimum life cycle irreversibility for the washing process can be found for each test piece separately at an acceptable detergency. Usually, the detergency is considered to be 80 percent. However, a detergency of 80 percent could not be reached for the test pieces 3, 4 and 5, within the range of the used washing temperature and detergent quantities. The detergency is therefore set to the maximum value of detergency that could be reached. The used detergency for the six test pieces in the minimisation is given in Table 6.3.

Table 6.3 Used detergency for the different test pieces

Test Pieces	Detergency
1	80%
2	80%
3	58%
4	58%
5	75%
6	80%

Substituting the detergency of each test piece into formula A3.1 to A3.6 of Appendix 3, results in a relation between the washing temperature and detergent quantity. Substituting this relation into equation 6.7, a relation of the life cycle irreversibility and

temperature or detergent quantity only is obtained. With this relation a minimum value of the life cycle irreversibility can be found. The minimum value of life cycle irreversibility and the values of both variables are given in Table 6.4, for the case of the conventional electrical heating system. The variations in the optimal washing temperature, detergent quantity and minimum life cycle irreversibility are due to variations in the function parameters as shown in Appendix 3.

Table 6.4 Optimal washing temperatures and detergent quantities for each test piece for a minimal life cycle irreversibility

Test pieces	T_{opt} (°C)	D_{opt} (gr)	I_{LC}^{eh} (MJ)
1	62 ± 12	89 ± 18	17.4 ± 3.7
2	46 ± 12	72 ± 15	11.4 ± 3.5
3	60	108	16.7
4	15	108	6.5
5	60	72	14.6
6	27 ± 11	62 ± 6	6.0 ± 2.8

To reach the maximum detergency of test piece 3 as listed in Table 6.3, the highest temperature and detergent quantity within the validity of the formulas given in Appendix 3 are chosen. The same applies to test piece 5, with the exception that for quantities greater than 72 grams the detergency does not depend on the detergent quantity anymore. The detergency relation of test piece 4 does not depend on the washing temperature, hence the washing temperature has been set at the tap water temperature, which is 15°C. Because the values of test pieces 3,4 and 5 are fixed and not calculated there are no variations associated with these as can be seen in Table 6.4.

Table 6.4 shows that a low optimum washing temperature immediately results in a low life cycle irreversibility. This was expected, since the heating process is the largest life cycle exergy consumer of the washing process. A similar table can be drafted for the other heating systems that were mentioned, i.e. the gas fired heating system and the district heating system. Instead of representing these data for each separate heating system, Tables 6.5 and 6.6 have been drafted that show the average optimum temperature and detergent quantity for each heating system. For this comparison only test piece 1,2 and 6 are considered here, because these can reach a detergency of 80%, as can be seen in Table 6.3. The average detergency of these three test pieces is set to 80 percent for each heating system. However, the detergency of the separate test pieces deviates within a range of 10 percent in detergency. The average detergency of all test pieces, including test pieces 3,4 and 5, is three percent lower for the gas fired heating and three percent higher for the district heating compared to the electrical heating system.

Table 6.5 Average optimum temperatures for the various heating systems

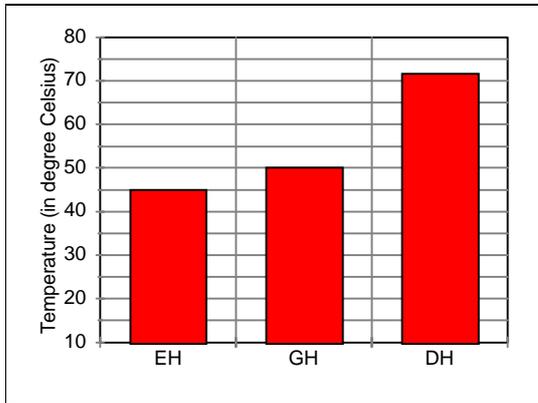


Table 6.6 Average optimum detergent quantities for the various heating systems

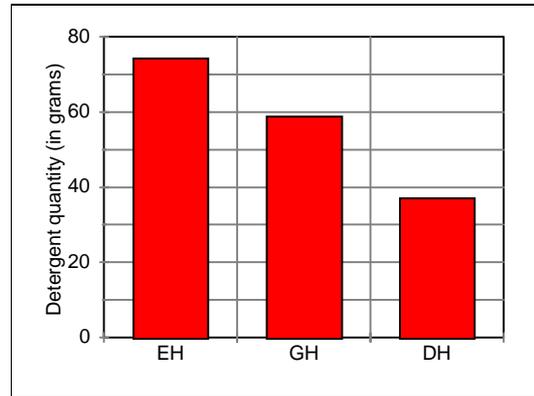
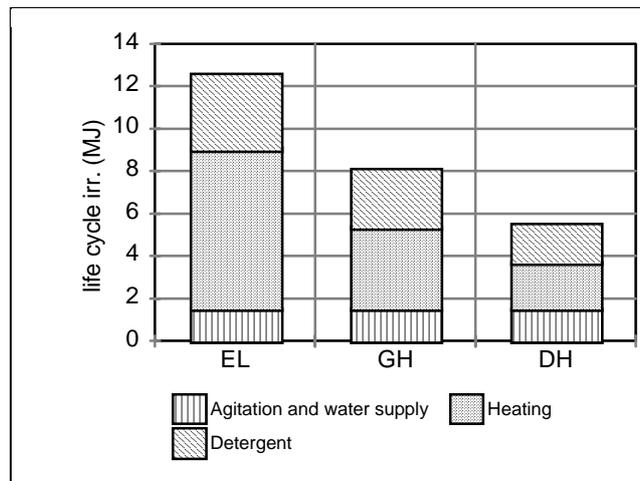


Table 6.7 Average minimum life cycle irreversibility for the various heating systems



These tables show that, as the heating system operates more efficiently, the optimum washing temperature rises. This again results in a lower demand of detergent and thus a lower optimum detergent quantity. More important, however, is that the life cycle irreversibility decreases to a considerable extent; i.e. from 12.6 MJ in the case of the electrical heating spiral to 8.2 and 5.4 MJ in the case of the gas fired heating system and the district heating system respectively.

6.4 Discussion

In the conventional textile washing process large quantities of exergy are used to remove just a few grams of soil. Most of this exergy is consumed by the heating process, since it takes a large quantity of energy to heat up 17 litres of water. Although the life cycle

irreversibility can be reduced to a considerable extent by using a district or a gas fired heating system, the life cycle irreversibility remains relatively large, compared to the actual exergy needed for soil removal in case of a reversible process. Research should be carried out for alternative washing processes.

Since the washing temperature and detergent quantity cause the major irreversibilities, it seems reasonable to further exploit the other variables that influence the washing performance i.e. the washing time, the agitation input and water input. These variables consume only a relatively small part of the life cycle irreversibility. Furthermore, attention should be given to better insulation of the textile washing machine.

It is indeed possible to determine a minimum life cycle irreversibility that results in a certain detergency of the separate test pieces. The optimum values of the separate test pieces, however, differ to quite some extent. The average optimum temperature and detergent quantity lead to variation in the detergency of the different test pieces, despite the fact that the average detergency is nearly equal for the three test pieces in the comparison. This leads to the fact that in the comparison of the different types of heating systems some test pieces are better and some test pieces are less well cleaned.

The average exergetic efficiency of power plants in the Netherlands will increase in the near future to about 45 - 50%. This will decrease the life cycle irreversibility of the electrical heating system. The listed efficiency on the LHV of 85% for the gas-fired heating can only be reached and even become higher, when special arrangements in or outside the washing machine are taken, for example a gas-fired boiler inside the washing machine. When the already existing tap water heating system is used, the efficiency will be around 50% on the LHV, while in the case of a new high efficiency boiler for combined central and tap water heating the efficiency will be 60% on the LHV.

A limit for the use of the district heating system can be the maximum obtainable temperature. It may not always be possible to obtain 80°C, so additional electrical heating may be necessary. This will lead to a higher life cycle irreversibility. A washing machine with a hot fill, i.e. where hot water is directly inserted, may be disadvantageous to the removal of enzyme sensitive soil, because enzymes do not work at temperatures higher than 40°C. The used relation for detergency as listed in Appendix 3 are only valid in the temperature range of 30 to 60°C. However, it is believed that extrapolating this formulas to 70°C, as is the case for the optimum temperature for the district heating system, is justified. However, at this high washing temperatures an older detergent is preferable, because its washing performance is then satisfactory without damaging the textile.

6.5 Conclusions

It is shown that the ELCA is a useful method to optimise washing cycles and compare different types of heating systems. An optimum function is established for the life cycle

irreversibility, which depends on the washing temperature and detergent quantity for three types of heating systems.

It has been found that the conventional electrical heating spiral is a very inefficient heating system in terms of life cycle irreversibility. It is, therefore, recommended to replace this system by a system based on gas fired heating or, if possible, on a district heating system. This will reduce the life cycle irreversibility by 1/3 and more than 1/2, respectively.

6.6 Nomenclature

c	= Concentration of the detergent, [g/l]
CExC	= Cumulative exergy consumption, [MJ]
C_p	= Heat capacity, [J/kgK]
D	= Detergent quantity, [g]
Det	= Detergency
de	= Infinitesimal exergy quantity, [J]
dq	= Infinitesimal heat quantity, [J]
DS	= District heating system
EH	= Electrical heating system
E	= Exergy, [MJ]
GH	= Gas fired heating system
I	= Irreversibility, [J]
M	= Mass, [kg]
LHV	= Lower heating value, [MJ]
η	= Efficiency
T	= Temperature, [°C]
V	= Volume, [m ³]

Superscripts

dh	= District heating system
eh	= Electrical heating system
gh	= Gas fired heating system

Subscripts

det	= Detergent
hp	= Heating process
lc	= Life cycle
o	= Environmental state
opt	= Optimal
tot	= Total
w	= Water

ws = Washing
wt = Water treatment

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Exergetic Life Cycle Analysis of different types of district heating systems

Abstract

The objective of this chapter is to perform an ELCA and to define the exergetic pay back period. This period can be used for energy systems as part of a more general assessment also including financial aspects. For the conservation of the exergy reservoirs of natural resources investments with a shorter exergetic pay back period will be preferable. To determine exergy savings of fuel efficient systems the ELCA has been used.

As an example a district heating system in combination with a gas fired cogeneration plant has been selected. The irreversibility, taking place during the life cycle of the system due to the construction, operation, dismantling and waste processing, is determined and compared with domestic central heating also based on natural gas. The analysis has been performed for four alternative district heating systems. The exergetic pay back period, defined as additional exergy invested divided by exergy saving, is shown to be between 0.07 and 1.2 years. The life cycle exergy saving varies from 24% to 49%, mainly depending on the density of the heat distribution system. In addition the differences in exergy saving are due to the fact that the small scale alternatives have a smaller gas turbine, which has a lower exergetic efficiency when compared with the big scale projects.

There is no direct relation between the exergetic and financial pay back period in this study. However, the financial pay back period is roughly one order of magnitude higher than the exergetic pay back period. From this it can be concluded that district heating systems with exergetic pay back periods of more than one year have unacceptable financial pay back periods.

7.1 Introduction

Investments are mostly judged on their financial benefit. The financial benefit can be measured with the financial pay back period (T_{fi}). For the assessment of investments a maximum period for the financial pay back period is fixed. This maximum period is based on the risks of investment, inflation, impact on the society, etc. and differs from less than

one year to more than ten years. However, to include the environmental benefit of energy efficient systems in the assessment a pay back period based on the use of natural resources has to be defined. In this chapter it will be shown how and how meaningful it is to define such a pay back period.

With an ELCA different systems, having the same functions, can be compared for its use of natural resources. For sustainable development the depletion of the exergy reservoirs of natural resources has to be reduced and exergy emissions have to be minimised to a sustainable level. So a pay back period based on the exergy spent on investment and exergy saved by this investment during operation can be defined.

For the thermodynamic and financial optimisation of the separate alternatives a thermo-economic analysis can be performed as for example suggested by Lozano and Valero (1993). In this chapter such an approach has been adopted in a simplified form for comparing different alternatives of district heating systems. The results of the ELCA and the financial analysis are used as data for the assessment, which leads to a preference order for the alternatives from the viewpoint of cost effective exergy savings.

7.2 Method

7.2.1. Pay back periods

The financial pay back period is defined as:

$$T_{fi} = \frac{C_{inv}}{CF} \quad (7.1)$$

where C_{inv} is the amount of money invested in the conversion of exergy, e.g. a power plant, and CF is the cash flow, which is the yearly income minus the yearly cost, associated with the production of electricity. The exergetic pay back period is defined as:

$$T_{ex} = \frac{I_{inv}}{\dot{E}_{pro}} \quad (7.2)$$

where I_{inv} is the irreversibility connected with the construction, maintenance and dismantling, which includes waste processing and \dot{E}_{LC} is the exergy of the electricity produced in the power plant during the life cycle of the plant (E_{pro}) minus I_{inv} divided by the operating time (t) or in formula form

$$\dot{E}_{LC} = \frac{E_{pro} - I_{inv}}{t} \quad (7.3)$$

However, this method can also be applied to compare more fuel efficient systems. A reference situation has to be chosen, which is mostly the situation of the conventional system. In this case the financial pay back period is defined as:

$$T_{fi} = \frac{C_{inv}}{CF} \quad (7.4)$$

where C_{inv} is the additional amount of money invested in the more fuel efficient system, and CF is the cash flow, which is the yearly income minus the yearly cost, associated with the saving of exergy due to this additional investment. C_{inv} can be calculated with

$$C_{inv} = C_{inv}^{alt} - C_{inv}^{ref} \quad (7.5)$$

where C_{inv}^{alt} and C_{inv}^{ref} are the cost of the alternative and reference system, respectively. The exergetic pay back period is then defined as:

$$T_{ex} = \frac{I_{inv}}{\dot{I}_{LCS}} \quad (7.6)$$

where I_{inv} is the additional irreversibility connected with the construction, maintenance and dismantling, which includes waste processing, for the exergy saving alternative system compared to the reference situation. In formula form

$$I_{inv} = I_{inv}^{alt} - I_{inv}^{ref} = I_{build}^{alt} + I_{dis}^{alt} - I_{build}^{ref} - I_{dis}^{ref} \quad (7.7)$$

and \dot{I}_{LCS} is the life cycle exergy saving divided by the operating time. In formula form

$$\dot{I}_{LCS} = \frac{I_{LC}^{ref} - I_{LC}^{alt}}{t} \quad (7.8)$$

where I_{LC}^{ref} and I_{LC}^{alt} are the life cycle irreversibilities of the reference and alternative situation, respectively, and t is the operating time. Generally speaking $I_{inv}^{alt} > I_{inv}^{ref}$ in an exergy saving situation and $I_{LC}^{alt} < I_{LC}^{ref}$, so that exergy will be saved. These life cycle irreversibilities are determined with the ELCA.

7.2.2 Assessment

For the general assessment of more fuel efficient systems the financial and the exergetic pay back period will be taken into account. This investment has to be made in such a way that the highest minimisation of irreversibility with the greatest profitability or lowest cost will be accomplished during the life cycle. This means that an investment with a shorter exergetic and financial pay back period is always preferred.

An investment function has been defined, which includes both exergetic and financial data. It is based on the saved life cycle exergy for the additional investment carried out and in formula form

$$O = \frac{\dot{I}_{LCS}}{C_{inv}} \quad (7.8)$$

where O has to be maximised.

7.3 Exergetic life cycle assessment (ELCA)

The goal of the ELCA is to compare the life cycle irreversibility of different systems for space heating and tap water heating of houses, office buildings and greenhouses for different locations in the Dutch situation. The reference situation is based on the now common new building situation with high efficiency boilers. The alternatives are based on a combined heat and power plant consisting of a steam and gas turbine plant (STAG) with a heat distribution network. The functional unit is the space and tap water heating in buildings.

7.3.1 Life cycles

Reference situation

The reference situation is based on the high efficiency boiler (HE boiler) on natural gas for space and tap water heating. In the Netherlands almost every house and office building is connected to the gas distribution network. The HE boiler has been introduced several years ago and is widely used in newly built houses. It is the most energy efficient heating system for houses and other buildings, when only the first law of thermodynamics is taken into account. The thermal efficiency is 99% based on the lower heating value. The water for the radiators is heated to 65°C, while the return temperature of the water is 45°C. Also the tap water is heated to 65°C with the HE boiler. The tap water use is taken to be 100 litres a day per household. The cooking takes place on natural gas with an average use of 85 m³ a year. The lower heating value of this gas is 31.7 MJ/Nm³. The natural gas consumption of the HE boiler per house is given in Table 7.1 for the different reference situations. Reference situation 1 is an existing situation with apartments in a high building. Reference situation 2 and 4 are new detached houses in a residential area. The different values are due to the difference in insulation of the houses. Reference situation 3 consists of green houses, so no value could be given.

Table 7.1 Gas consumption per house in m³

Reference situation	gas consumption
1	1719
2	1271
3	-
4	1271

Alternatives

For the alternatives the hot water for the heat distribution network is provided by a steam and gas turbine (STAG) at a temperature of 130°C and the return temperature of the water is 75°C. The heat distribution network consists of a collective part which transports hot water heated by the STAG, and a direct part which is heated by the collective net and delivers the heat to the consumers. The peak load is delivered by auxiliary boilers on natural gas. The cooking takes places with electricity and an equivalent heat production during cooking is assumed as with gas. Four different situations for district heating systems are analysed as shown in Table 7.2. Alternative 1 is a big scale project in an existing situation with high buildings. Because of the high buildings the distribution net is dense, while the insulation of the flats is bad compared to new buildings. More information about the alternatives is summarised in Table 7.4. Alternative 2 is a big scale project in a residential area with detached houses. Alternative 3 is a small scale project with greenhouses, which are located close to each other. Alternative 4 is comparable to alternative 2. However, it is now a small scale project. Alternative 1 is under construction, alternative 2 has been built already and several projects like alternative 3 have also been built¹.

Table 7.2 The four alternatives

Alt.	big or small scale	dense or widespread distribution system	existing or new buildings
1	big	dense	existing
2	big	widespread	new
3	small	dense	existing
4	small	widespread	new

¹ Typical examples of projects like alternative 1 to 4 are in the Netherlands: de Bijlmer, Almere, Klazinaveen and Almere small.

7.3.2 Exergy analysis

The irreversibility during construction

The irreversibility associated with the construction inside the houses is for both types of heating systems quite similar: steel tubes and radiators. So they can be left out in the analysis. The irreversibility associated with the building of a STAG is estimated to be 1% of the total life cycle irreversibility of the STAG according to Dwarshuis et al. (1995) and is neglected. So only the gas and heat distribution network outside the buildings have to be taken into account.

The gas distribution net is made of PVC for the main tubes and HDPE for the smaller tubes. The heat distribution net is constructed with steel tubes which are insulated with PUR-foam and the outside of the tube is protected by a HDPE tube. The tube inner diameter of the heat distribution net varies from 32 to 600 mm, while the thickness of the insulation layer of PUR-foam varies from 31 to 72 mm.

The PUR and HDPE used in the heat distribution network are assumed to be primary materials, i.e. materials produced from natural resources. The steel, PVC and HDPE used in the gas distribution network are recycled after the life cycle of the networks, so these are secondary materials. The total exergy consumption of the production is given by the cumulative exergy consumption (CExC) as defined by Szargut (1988). The irreversibility associated with the material production is the CExC minus the exergy of the material as stated in Szargut (1988). Because no recent data about the CExC is available we use the cumulative energy consumption (CEC) for the calculation of the CExC. The CExC is determined to the second level, i.e. the irreversibility associated with the production of the plants themselves are not taken into account. The error introduced by this will be small.

The CEC of the primary production of HDPE and PUR with a density of 30 kg/m^3 are found in van Heijningen et al. (1992) and in Kindler and Nikles (1980), respectively. The irreversibility associated with the recycling of steel to steel slabs is about 8 MJ per kg according to Wall (1986), including the irreversibility associated with the power generation of 2.4 MJ per kg steel. The energy use of the secondary production, i.e. recycling, of PVC and HDPE is estimated to be 5 MJ/kg (Heijningen et al., 1992). The energy use for bending or forming to tubes is estimated to be 2.2 MJ/kg for steel and 9 MJ/kg for HDPE and PVC. Because most inputs are fossil fuels for the materials production and the exergy and enthalpy content are roughly equal, the CExC is assumed to be equal to the CEC. The results are listed in Table 7.3.

Table 7.3 The CExC, exergy and irreversibilities associated with the material production in MJ/kg

Material	CExC	exergy	Irreversibility ²
Steel tube (secondary)	16.9	6.7	10
PVC tube (secondary)	32.5	18.5 ³	14
HDPE tube (primary)	80.9	48.5	32
HDPE tube (secondary)	62.5	48.5	14
PUR (primary)	98	27 ⁴	71

The removal of the soil for the trenches is estimated to cause an irreversibility of 66 MJ/m³, based on the fuel use of the excavator. The irreversibility associated with the use of fresh stone-free sand for the building of the heat distribution net is 537 MJ/m³ (Heijningen et al., 1992). It is assumed that this sand has been transported over 100 km. The irreversibility connected with the investment for the different alternatives is calculated as 134, 401, 16 and 118 TJ for alternatives 1 to 4, respectively, as is showed in Appendix 4.

The irreversibility during operation

The exergetic efficiency of the HE boiler is 8.7%, where the product is the heating of the water from 45°C to 65°C. The reference temperature is taken to be 0°C, because the heating systems are mostly used during the winter.

The STAG for the big scale alternatives uses an ABB GT 13E gas turbine of 145 MW of electricity and has an exergetic efficiency of 50%, which hardly changes when electricity production is reduced and heat is delivered to the heat distribution net. The small scale alternatives use a STAG with an ABB GT 10B gas turbine, which delivers a power of 24 MW and has an exergetic efficiency of 46.5%, which is also almost independent of the heat to power ratio. The exergetic efficiency of the auxiliary boilers, which only operate during peak hours, is 13%, based on the heating of water from 70°C to 90°C. The exergetic efficiency of the heat transport, defined as the exergy output of heat divided by the exergy input of heat added with the fuel input for the power generation used by the pumps, is 42%, 32%, 40% and 33% for the alternatives 1, 2, 3 and 4, respectively. The exergetic efficiency of power generation for the pumps is taken to be 40%, which is about the mean value in the Netherlands. The exergy and fuel savings during operation of the alternatives compared to the reference systems are listed in Table 7.3. For the reference situation and alternative 1 Grassmann diagrams are given in Figure 7.1 and 7.2. The numbers in the figures are based on an exergy input of natural gas of 100% in Figure 7.1. It can be seen that the irreversibility during operation can be almost halved by

² Because of the uncertainty in the used data all irreversibility values are rounded to MJ

³ Based on the exergy of vinylchloride of 19.8 MJ/kg and an estimate of the ΔG^0 of polymerisation with the group contribution method of Anderson, Beyer and Watson (Wilson, 1964)

⁴ For lack of better data the exergy is assumed to be equal to the lower heating value

replacing the HE boiler by a district heating system, where the heat is delivered by a STAG.

The irreversibility during dismantling and disposal

The steel tubes are recycled. The irreversibility associated with the recycling is taken into account in the paragraph dealing with construction. The PUR isolation material and HDPE outer tubes are burned in a waste incineration plant with power generation. The exergetic efficiency of the power generation is 25% (Korobitsyn et al., 1996). The power generation by 1 MJ exergy of disposed PUR and HDPE material saves the use of 0.63 MJ exergy of primary material for power production, when an exergetic efficiency of 40% is taken for power plants. The power generation causes an irreversibility of 18, 47, 2.0 and 131 TJ for alt. 1 to 4, respectively, as shown in Appendix 4. The HDPE and PVC tubes of the gas distribution net are recycled. The secondary PVC is used together with primary PVC in sewage tubes. The operating time of the networks is assumed to be 25 years.

7.3.3 Results of the ELCA and suggestions for improvement

Results are presented in Table 7.4. The life cycle exergy saving by replacing the domestic HE boilers by a district heating system with a STAG is 24% to 49%. The difference in exergy savings of the alternatives occurs mainly because of more heat transfer to the environment in the widespread distribution net and to a lesser extent by the lower efficiency of the STAG in the small scale projects. As shown in the Grassman diagrams, i.e. exergy flow diagrams, there is still an improvement potential by lowering the ΔT in the heat exchangers in the distribution network or combining the collective and direct parts of the net, which will result in a lower hot water temperature at the STAG and, associated with this, a lower exergy input. Also the heat transfer of the distribution network tubes to the environment can be decreased, e.g. by replacing the separate hot and cold tubes by one annular design.

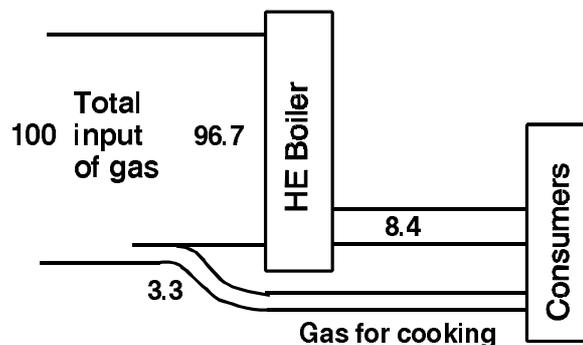


Figure 7.1 Grassman diagram of reference situation 1

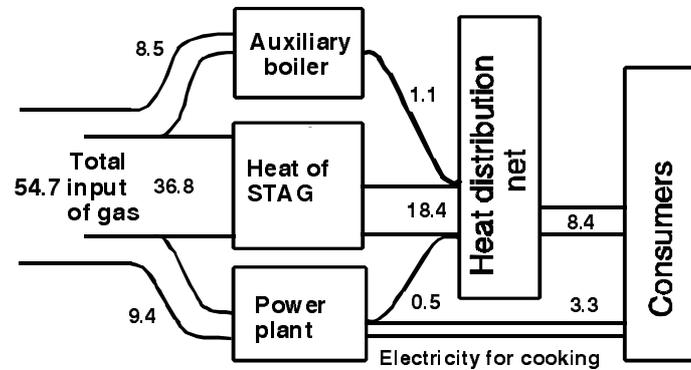


Figure 7.2 Grassman diagram of alternative 1

7.4 Economics

The additional investment costs for the more exergy efficient system are the investment of the heat distribution net including pumps and auxiliary boilers and the adjustment cost of the STAG for heat delivery. The cost of the gas distribution net in the case of new building areas has to be subtracted, because instead of a gas distribution net a heat distribution net has been built. The costs are estimated according to the guide of the Centre for Energy Conservation and Environmental Technology (1982) and are valid for the Dutch situation⁵. The interest rate and inflation are assumed to be 7% and 2%, respectively. The energy price is \$ 0.32 per m³ for natural gas and \$ 10.20 per GJ for heat for private consumers and small companies. For the green houses in alternative 3 and for the gas consumption of the STAG the energy price is \$ 0.12 per m³ for natural gas and \$ 3.71 per GJ heat. The exergy of the natural gas is 32.7 MJ/Nm³. The construction of the gas distribution network is \$ 1000 per house in a new building area. With this data the additional investment for the more exergy efficient systems is calculated as 55, 104, 9.2 and 29.6 M\$ for alternative 1 to 4, respectively, and listed in Table 7.4.

7.5 Results

The results are presented in Table 7.4. The exergetic pay back period (T_{ex}) is low and differs because of the density of the heat distribution system. The irreversibility associated with the investment (I_{inv}) can be neglected with respect to the life cycle exergy saving. There is no direct relation between the exergetic and financial pay back period. This is caused by the fact that the “industrial” consumers in alternative 3 pay less than half the gas price than the private consumers. Leaving out option 3 it can be concluded

⁵ An exchange rate of 1.7 guilder for 1 US dollar has been assumed

that district heating systems with exergetic pay back periods of more than one year have unacceptable financial pay back periods of more than ten years.

According to the optimum investment function alternative 3 is preferred most, followed by 1 and alternative 4 and 2 are preferred least. Alternative 3 gives the greatest exergy saving for every invested dollar. However, the low prices of natural gas for the greenhouses causes a relative high financial pay back period.

Table 7.4 The data and results for the different alternatives

Alternative	1	2	3	4
Number of houses	18200	30500	0	11000
Number of companies	167	0	35	0
Tube length (km)	61	1063	11.7	380
Mean diameter (mm)	400	65	250	40
Heat demand (TJ/y)	1408	1230	504	444
Heat demand per house (GJ/y)	54.5	40.3	-	40.3
Heat production STAG (TJ/y)	1365	1487	477	528
Heat production help boiler (TJ/y)	114	98	40	36
Heat transfer to environment (TJ/y)	71 (5%)	355 (24%)	13 (3%)	120 (23%)
Pump power (TJ/y)	18	68	5	18
Exergetic efficiency heat transport	42%	32%	41%	31%
Direct fuel saving	45%	27%	49%	25%
Direct exergy saving (TJ/y)	700	373	263	127
Additional investment	152	447	18	130
irreversibility I_{inv} (TJ)				
Life cycle exergy saving \dot{I}_{LCS} (TJ/y)	694	355	262	122
relative saving (%)	(44%)	(26%)	(49%)	(24%)
Add financial invest. C_{inv} (M\$)	55	109	9.2	29.6
Exergetic pbp T_{ex} (y)	0.22	1.26	0.07	1.07
Financial pbp T_{fi} (y)	7.1	56	31	21.6
Optimum investment function O (MJ/(\$·y))	12.6	3.26	28.5	4.12

7.6 Conclusions

An ELCA is needed to calculate the exergy saving of new options of thermal energy delivery. From a point of view of saving natural resources and the environmental effects associated with their use an ELCA should be carried out for each alternative energy system. The exergetic pay back period of each alternative shows which investment gives the greatest exergy saving per unit irreversibility associated with the investment.

As shown by the low exergetic pay back period the replacement of domestic gas heating by a district heating system connected with a STAG leads to a substantial exergy saving. However, the financial pay back period can place a limit to this replacement. In the case of big scale projects in a residential area of detached houses the financial pay back period is too high.

To get cost effective savings of natural resources an investment function based on life cycle exergy saving and financial investment has been defined. It gives the exergy saving per year for every invested dollar. Because of different gas prices for the users there is no uniform relation between a low financial pay back period and a high optimum investment function.

Using the exergetic pay back period and the investment function in addition to the financial pay back period will lead to more fuel efficient investment decisions.

7.7 Nomenclature

Alt	=	alternative
C	=	amount of money
CEC	=	cumulative energy consumption
CExC	=	cumulative exergy consumption
CF	=	cash flow
ELCA	=	exergetic life cycle analysis
\$	=	US dollar
HDPE	=	high density polyethene
HE	=	high efficiency
I	=	irreversibility
O	=	optimum investment function
pbp	=	pay back period
PVC	=	polyvinylchloride
PUR	=	polyurethane foam
t	=	operating time
T	=	pay back period
STAG	=	steam and gas turbine plant
y	=	year

Subscripts

build	=	building
dis	=	dismantling
ex	=	exergetic
fi	=	financial
inv	=	investment

LC = life cycle
LCS = life cycle saving

Superscripts

alt = alternative
ref = reference
tot = total

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Exergetic optimisation of a single component: the heat exchanger

Abstract

The objective of this chapter is to show that the concepts of an ELCA, the minimisation of life cycle irreversibility, can be applied for the optimal design of a component, where there is a trade-off between exergy saving during operation and irreversibility during construction. The analysis has been carried out for the optimal design of a the heat exchanger in a district heating system. The optimisation takes into account irreversibilities due to frictional pressure drops and the temperature difference between the hot and cold stream and to irreversibilities due to the production of the materials and the construction of the heat exchanger. As an example of this type of heat exchanger a water to water heat exchanger in a district heating system has been selected. The influence of the configuration of the heating system, including the energy conversion, on the optimisation of the heat exchangers has been shown. The analysis of the heat exchanger in which the concepts of the ELCA are used gives the design conditions of the heat exchangers which lead to the lowest life cycle irreversibility.

8.1 Introduction

Because of its widespread use the heat exchanger has been selected as an example. Bejan (1982) studied extensively the optimisation of a heat exchanger, excluding exergy destruction associated with use of materials and the generation of heat and power. His approach uses the concept of entropy generation minimisation. An extension to his approach to include material use has been made by Aceves-Saborio et al. (1989). They took into account the irreversibility associated with the use of the materials, but did not include the irreversibility due to the pressure drops.

Tondeur and Kvaalen (1987) have shown that in the case of heat exchangers or separation devices involving a given heat transfer and achieving a specified transfer duty, the total entropy produced is minimal when the local rate of entropy production is uniformly distributed along space variables and time. De Oliveira et al. (1991), (1994) have shown that in the case of an optimal heat exchanger the thermal and viscous

contributions to the entropy generation should be equal when the heat flux is optimised. The ratio of thermal and viscous contribution to the exergy destruction is between one and three when the Reynolds number or hydraulic diameter are optimised. However it is not shown that this is the case when both, Reynolds number and hydraulic diameter, are together optimised.

Lozano and Valero (1993) have developed a theory to allocate exergetic and monetary cost. In this theory they define a matrix containing all irreversibilities, including irreversibilities associated with the building of installations and the disposal of waste materials. No extensive examples of this theory, including irreversibilities associated with the building of installations and the disposal of waste materials, are available.

In this chapter the optimisation of the heat exchanger has been performed using the criterion of the minimisation of the life cycle irreversibility for the delivery of domestic hot water. The methodology is based on the minimisation of the life cycle irreversibility. It uses the marginal life cycle irreversibilities of the different components.

8.2 Optimisation of a heat exchanger

8.2.1 The heat exchanger

The heat exchanger analysed is a balanced counter flow heat exchanger, which is used in a district heating system to heat the domestic tap water. The inner tubes carry the cold stream and the surrounding outer tube carries the hot stream. An equal mass flow of the hot and cold water has been assumed. The inner and outer tube have been constructed from copper and steel, respectively. The combined annular heat exchanger is helically wound as shown in Figure 8.1. The thermal insulation of the heat exchanger has been assumed to be perfect.

The following formula can be derived for the irreversibility in the heat exchanger due to the stream to stream heat transfer and pressure drops:

$$\dot{I} = \dot{I}^{\Delta T} + \dot{I}^{\Delta P} \quad (8.1)$$

with

$$\dot{I}^{\Delta T} = T_0 \left[\dot{m} c_p \ln \frac{T_{1,out}}{T_{1,in}} + \dot{m} c_p \ln \frac{T_{2,out}}{T_{2,in}} \right] \quad (8.1a)$$

and

$$\dot{I}^{\Delta P} = \left[\frac{\dot{m}}{\rho} (P_{1,in} - P_{1,out}) + \frac{\dot{m}}{\rho} (P_{2,in} - P_{2,out}) \right] \quad (8.1b)$$

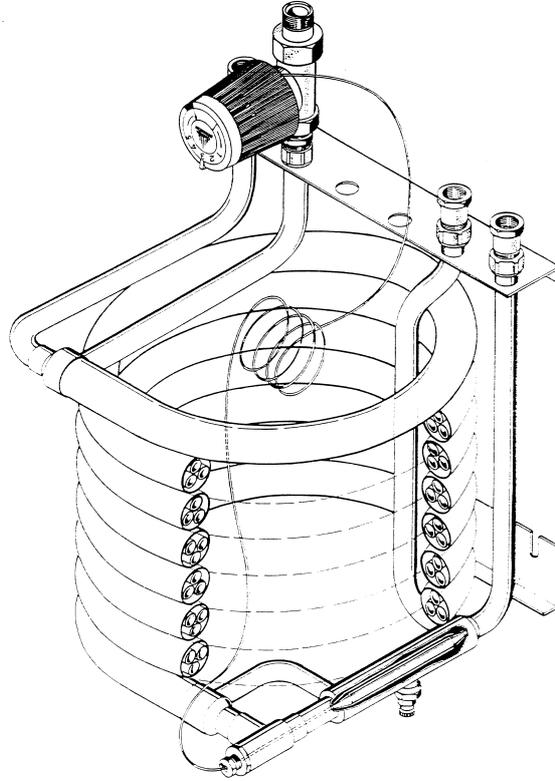


Figure 8.1 The analysed heat exchanger

Formula (1a) can be rewritten in

$$\dot{I}^{\Delta T} = T_0 \cdot \dot{m} c_p \left[\ln \frac{T_{1,\text{out}}}{T_{1,\text{in}}} + \ln \frac{T_{1,\text{in}} + \Delta T}{T_{1,\text{out}} + \Delta T} \right] \quad (8.2)$$

because $T_{2,\text{out}} = T_{1,\text{in}} + \Delta T$ and $T_{2,\text{in}} = T_{1,\text{out}} + \Delta T$ in a balanced heat exchanger. With the heat balance of the n inner tubes

$$n \cdot \dot{m}_n c_p (T_{1,\text{out}} - T_{1,\text{in}}) = \alpha A \Delta T \quad (8.3)$$

with $\dot{m}_n = \frac{\dot{m}}{n}$, ΔT in (8.2) can be substituted. By using the force balance we can write for the pressure drops in the tubes:

$$\Delta P_1 = P_{1,\text{in}} - P_{1,\text{out}} = 2f_1(Re) \cdot \rho \cdot \bar{u}_1^2 \frac{L}{D_1} \quad (8.4a)$$

and

$$\Delta P_2 = P_{2,\text{in}} - P_{2,\text{out}} = 2f_2(Re) \cdot \rho \cdot \bar{u}_2^2 \frac{L \cdot (D_2 + n \cdot (D_1 + 2d_1))}{(D_2^2 - n \cdot (D_1 + 2d_1)^2)} \quad (8.4b)$$

Using

$$\bar{u}_1 = \frac{4\dot{m}_n}{\rho\pi D_1^2} \quad (8.5a)$$

and

$$\bar{u}_2 = \frac{4\dot{m}}{\rho\pi [D_2^2 - n \cdot (D_1 + 2d_1)^2]} \quad (8.5b)$$

for the mean velocities of the fluid in the tubes and substituting (8.4a) and (8.4b) in (8.1b) yields

$$\dot{j}^{\Delta P} = \frac{32}{\pi^2} \left[f_1(Re) \frac{n \cdot \dot{m}_n^3}{\rho^2} \frac{L}{D_1^5} + f_2(Re) \frac{\dot{m}^3}{\rho^2} \frac{L \cdot (D_2 + n \cdot (D_1 + 2d_1))}{[D_2^2 - n \cdot (D_1 + 2d_1)^2]^3} \right] \quad (8.6)$$

8.2.2 Turbulent region

In the turbulent flow region in tubes and annular spaces with a limited temperature difference of 5 K for liquids between the bulk fluid and pipe surface temperature we have for the heat transfer coefficient (α) according to Chapman (1974)

$$Nu = \frac{\alpha D_h}{\lambda} = 0.023 Re^{0.8} Pr^n = 0.023 \cdot \left(\frac{\rho \bar{u} D_h}{\mu} \right)^{0.8} \cdot \left(\frac{c_p \mu}{\lambda} \right)^n \quad (8.7)$$

with $n = 0.3$ or 0.4 for cooling or heating, respectively. Experimental data of Kays and London (1984) gives a similar relation.

The friction factor in tubes according to the friction law of Blasius is given in Rogers and Mayhew (1967) as

$$f(Re) = \frac{0.0791}{Re^{0.25}} = 0.0791 \cdot \left(\frac{\mu}{\rho \bar{u} D_h} \right)^{0.25} \quad (8.8)$$

However, to include the curving of the tube the following correction factor according to Ito (1959) has to be included in (8.8)

$$C = 0.962 + 0.092 \cdot Re^{0.25} \left(\frac{D}{R} \right)^{0.5} \quad (8.9)$$

where R is the diameter of the curve. Substituting (8.3) and (8.7) in (8.2) and (8.8) and (8.9) in (8.6), using $A = n \cdot \pi \cdot L \cdot D_1$ for the inner tubes and $A = n \cdot \pi \cdot L \cdot (D_1 + 2d_1)$ for the outer tubes and neglecting the heat resistance of the tube wall we obtain

$$\dot{i}^{\Delta T} = T_0 \cdot \dot{m} \cdot c_p \left[\ln \frac{T_{1,out}}{T_{1,in}} + \ln \left(\frac{T_{1,in} + \Delta T}{T_{1,out} + \Delta T} \right) \right] \quad (8.10a)$$

with

$$\Delta T = \frac{(T_{1,out} - T_{1,in})}{0.023 \cdot n \cdot 4^{0.8} \cdot \pi^{0.2}} \left(\mu^{0.4} \left(\frac{c_p}{\lambda} \right)^{0.6} \frac{\dot{m}}{\dot{m}_n^{0.8}} \frac{D_1^{0.8}}{L} + \mu^{0.5} \left(\frac{c_p}{\lambda} \right)^{0.7} \frac{m^{0.2} (D_2^2 - n \cdot (D_1 + 2d_1)^2)}{L \cdot (D_1 + 2 \cdot d_1) \cdot (D_2 + n \cdot (D_1 + 2 \cdot d_1))^{0.2}} \right)$$

and

$$\dot{i}^{\Delta P} = \frac{1.79}{\pi^{1.75}} \frac{\mu^{0.25}}{\rho^2} \left[\frac{n \cdot \dot{m}_n^{2.75} L}{D_1^{4.75}} \left(0.962 + 0.092 \cdot \left(\frac{4 \cdot m_n}{\pi \mu D_1} \right)^{0.25} \left(\frac{D_1}{R} \right)^{0.5} \right) + \frac{\dot{m}^{2.75} L \cdot (D_2 + n(D_1 + 2d_1))^{1.25}}{[D_2^2 - n(D_1 + 2d_1)]^3} \left(0.962 + 0.092 \cdot \left(\frac{4 \cdot m}{\pi \mu (D_2 - n \cdot (D_1 - 2d_1))} \right)^{0.25} \left(\frac{D_2}{R} \right)^{0.5} \right) \right] \quad (8.10b)$$

In Appendix 5 the formulas used are compared with experimental results of the manufacturer for the heat exchanger SP 15 N (KVM, 1993), which is a heat exchanger with three inner tubes. As can be seen in the appendix for the thermal component the experimental and thermal results match perfectly, while in the case of the mechanical losses there is a big difference. This big difference is probably caused by the irregular position of the inner tubes in the outer tube. The theoretical values are used further on in this chapter.

8.2.3 Cumulative losses due to power and heat generation

Irreversibilities in heat exchangers are associated with external exergy destruction, for example, to overcome the frictional losses in the heat exchanger a pressure difference is needed. The generation of heat and power is associated with exergy destruction, which has to be taken into account in the optimisation, according to the life cycle approach. We assume the following situation, based on district heating. In this system a combined heat and power plant, consisting of a steam and gas turbine and a provision to extract steam, is used, which has an exergetic efficiency of 50%, when there is no useful heat production. The exergetic efficiency of such a plant stays nearly constant when part of the steam is used for the district heating system. The operator is free to a certain extent

to provide exergy in the form of either electricity or heat. Irreversibilities associated with the building of the combined heat and power plant and the transport of the fuel, natural gas, are neglected in this chapter. In chapter 7 it is demonstrated that this can be neglected.

The exergetic efficiency of the heat transport to the houses is estimated to be 0.5 for a widespread distribution net⁶ (wide net) and 0.75 for a very dense distribution net (dense net). This can be compared with alternative 1 and 2 in the preceding chapter, respectively⁷. The exergy destruction in the wide net is greater because of more power needed to overcome the frictional pressure drops and more heat transfer to the environment. A great part of the exergy destruction in the heat transport takes place because of the temperature difference in the heat exchanger between the main transport tube and the local distribution net. It has been assumed that the exergy destruction associated with the heat transport is independent of the district heating water temperature to ease the calculations. The exergetic efficiency of the pumps is assumed to be 0.7.

Hence the exergetic cost, which is the amount of exergy which is used for the production of one exergy unit, can be calculated for the analysed heat exchanger of pressure rise, k_p , and heat, k_T . The exergetic cost for the pressure rise (k_p) is $k_{\text{power}} \cdot k_{\text{pumps}} = 2.85$, where $k_{\text{power}} = 2$ and $k_{\text{pumps}} = 1.43$. These are calculated by taken the inverse of the exergetic efficiency of power generation and pumping. The exergetic cost for the heat (k_T) can be calculated to be 4 for the wide net and 2.66 for the dense net.

For the exergy destruction associated with the operation of the heat exchangers we have

$$\dot{I}_{\text{oper}} = k_T \dot{I}^{\Delta T} + k_p \dot{I}^{\Delta P} \quad (8.11)$$

8.2.4 Irreversibilities associated with the use of the material

The life cycle flow diagram of the heat exchanger is displayed in Figure 8.2. A detailed exergy analysis of the primary anode copper process is presented by Kolenda et al. (1992). The cumulative irreversibility of the flash smelting process, one of the main copper production processes, is shown to be 45.8 MJ per kg anode copper. Including the exergy destruction due to discharge and dissipation of materials into the environment, like slag and combustion gasses, gives a total cumulative exergy destruction of 55.6 MJ per kg primary anode copper.

⁶ This is a hypothetical situation. In reality the exergy losses are higher because the peak in heat demand is supplied by auxiliary heating boilers using natural gas. Their exergetic efficiency is very poor.

⁷ This listed exergetic efficiency of these systems is lower, because the exergy destruction of the heat exchanger optimised in this chapter is already included.

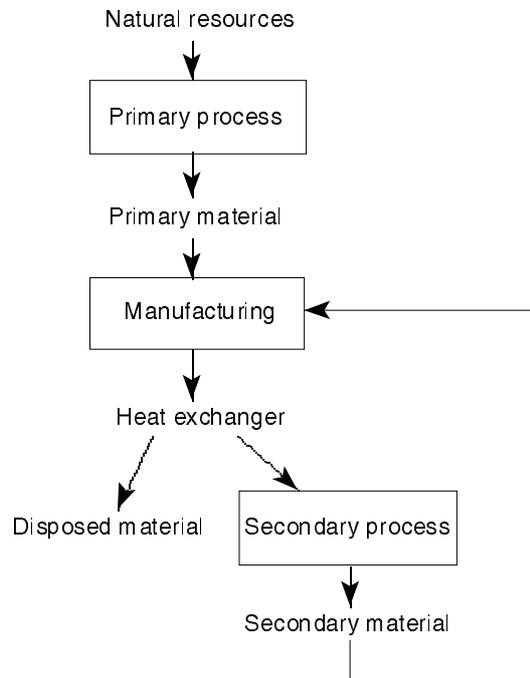


Figure 8.2 Life cycle of the heat exchanger

No recent publication about the exergy destruction of the production of primary steel, the production of copper out of anode copper, the production of secondary copper and manufacturing of steel and copper tubes is available, so due to lack of sufficient data the energy consumption instead of the exergy destruction has been taken. To use the energy consumption instead of the exergy destruction, the following conditions have to be fulfilled. Most inputs ought to be fossil fuels, because the exergy and enthalpy content are then roughly equal or electricity, because exergy and energy are then equal. The exergy increase of the product during the processing has to be subtracted from the energy use to get the exergy destruction.

The cumulative energy use of the production of solid copper out of anode copper is around 4.5 MJ/kg copper according to Boustead and Hancock (1979). The exergy increase is almost zero and we assume the enthalpy values of the inputs to equal the exergy values, so this value is taken for the exergy destruction. The cumulative exergy destruction for primary copper becomes $55.6 + 4.5 = 60.1$ MJ/kg ($C_{pri,Cu}$).

The energy use for the production of primary steel slabs is 16.9 MJ/kg⁸ according to Worrell (1992). Because most inputs are raw materials of which the exergy content roughly equals the energy content the cumulative exergy use is estimated to be equal to

⁸ The energy use is calculated for the steel factory of Hoogovens IJmuiden in the Netherlands, which can be considered to be one of the most energy efficient steel factories in the world.

the energy use. The exergy destruction is the exergy input reduced by the exergy increase due to the change of iron oxide to steel. The exergy increase is estimated to be 6.4 MJ/kg, so the exergy destruction is 10.5 MJ/kg ($C_{pri,s}$). The cumulative exergy destruction of secondary steel is calculated from Wall (1986) by assuming the efficiency of the electricity production to be 0.5. The exergy destruction associated with the production of the alloying materials and lime has been neglected. Boustead and Hancock (1979) give two totally different values for the energy consumption of the secondary copper production, namely 7.2 and 48.8 MJ/kg. The exergy destruction of this process is taken to be 20 MJ/kg ($C_{sec,Cu}$).

The manufacturing process of the steel tubes includes hot and cold rolling and the bending of the tube. The energy consumption of hot and cold rolling is 5.3 MJ/kg according to Worell (1992). The exergy destruction is assumed to be equal to the energy consumption, because the exergy of the material is hardly changed. The exergy destruction of the bending of steel has been estimated by assuming that metal is heated to 900°C by a gas heater. The exergy destruction associated with the force required to bent the steel has been neglected. The exergy destruction associated with welding of steel tubes has been estimated to be 0.260 MJ per meter ($C_{w,s}$) according to the Dutch steel maker, Hoogovens IJmuiden. The energy consumption associated with the manufacturing of the copper tubes, which includes the welding of copper tubes, is 14.7 MJ/kg ($C_{man,Cu}$) according to Alvarado-Grandi (1989). The exergy destruction is assumed to be equal to the energy consumption, because the exergy of the material is hardly changed during the manufacturing. The exergy destruction of the production of each material is shown in Table 8.1.

Table 8.1 Exergy destruction associated with the production of material and manufacturing of tubes

Process	I_s (MJ/kg)	I_{Cu} (MJ/kg)
primary process	10.5	60
secondary process	4.4	20
manufacturing process	5.7	15

The heat exchanger outer tube is insulated by polyurethane (PUR) foam with a thickness of 0.025 m. The cumulative energy consumption for the production of PUR with a density of 30 kg/m³ is 98 MJ/kg according to Kindler et al. (1980). The exergy content is estimated to be 27 MJ/kg on basis of the lower heating value according to Kindler et al. (1980). So the cumulative exergy destruction is 71 MJ/kg (C_{PUR}). No recycling of the PUR has been assumed.

The exergy destruction associated with the manufacture of the heat exchangers is due to the production of copper tube and steel tube, welding and the production of the PUR-foam.

$$\begin{aligned}
\dot{I}_{\text{man}} = & \frac{M_{\text{Cu}} C_{\text{Cu}} + M_{\text{S}} C_{\text{S}} + LC_{\text{w,S}} + M_{\text{PUR}} C_{\text{PUR}}}{t} = \\
& \frac{\pi L}{t} \left[(D_1 + 0.5d_1)d_1 \rho_{\text{Cu}} (C_{\text{man,Cu}} + x_{\text{Cu}} C_{\text{sec,Cu}} + (1-x_{\text{Cu}})C_{\text{pri,Cu}}) \right] \\
& + \frac{\pi L}{t} \left[(D_2 + 0.5d_2)d_2 \rho_{\text{S}} (C_{\text{man,S}} + x_{\text{S}} C_{\text{sec,S}} + (1-x_{\text{S}})C_{\text{pri,S}}) + C_{\text{w,S}} \right] \\
& + \frac{1}{t} \left[(L \cdot (D_2 + 2 \cdot d_2 + 0.025) \cdot \pi) \cdot 0.025 \rho_{\text{PUR}} C_{\text{PUR}} \right] \quad (8.12)
\end{aligned}$$

in which t is the operating time of the heat exchanger during its life cycle and x is the recycling ratio which is the proportion of secondary material, i.e. material which is recycled. For the calculations of the amount of material the mean diameters of the tubes has been taken.

8.3 Results

From the above considerations we obtain an expression for the total life cycle irreversibility, which has to be minimised.

$$\dot{I}_{\text{LC}} = \dot{I}_{\text{oper}} + \dot{I}_{\text{man}} \quad (8.13)$$

Where \dot{I}_{oper} and \dot{I}_{man} are stated in (8.11) and (8.12), respectively.

The following operating parameters have been assumed for the heat exchanger. The mass flow of the district heating water as the domestic tap water is 0.1 kg/s. The incoming temperature of the cold domestic tap water is 15°C. The domestic tap water is heated to 65°C. So, the exergy increase of the domestic tap water is 1628 W. The temperature of the incoming district heating water is variable. The environmental temperature, T_0 , is 25°C. The operating time for the heat exchanger is 30 minutes a day on full load for 10 years. The mean temperature of the inlet and outlet streams is used for the heat capacity, viscosity⁹ and thermal conductivity of water. The recycling ratio (x) is set to be 0.9 for the copper and steel parts of the tube.

8.3.1 Reference configuration

As a reference situation the SP 15 N discussed before is taken (KVM, 1993). This domestic water heat exchanger is commercially available. It has three inner tubes with an inner diameter of $8.4 \cdot 10^{-3}$ m and a outer tube with an inner diameter of $2.77 \cdot 10^{-2}$ m. The length is 6.4 m. The wall thickness of the inner and outer tube are $0.8 \cdot 10^{-3}$ and $2 \cdot 10^{-3}$ m,

⁹ The value of the viscosity is strongly temperature dependent. The viscosity at 15°C, 70 °C and 80°C is $1.14 \cdot 10^{-3}$ Pa·s, $0.406 \cdot 10^{-3}$ Pa·s and $0.357 \cdot 10^{-3}$ Pa·s, respectively. So the assumption of the mean temperature will cause a deviation from the real situation.

respectively. The life cycle irreversibility of the heat exchanger in full operation is $6.68 \cdot 10^3$ W for the wide net and $4.47 \cdot 10^3$ W for the dense net. Resulting from a ΔT , ΔP_1 and ΔP_2 of 28.2 K, 0.049 bar and 0.014 bar.

8.3.2 Results of optimisation

The minimisation of \dot{I}_{LC} for the 4 variables, D_1 , D_2 , L and n is given in Table 8.2 for the wide and dense net. The number of inner tubes is limited to require the condition of turbulent operation.

The value of the minimum life cycle irreversibility for the heat exchanger in the wide net of the district heating system is $7.24 \cdot 10^2$ W. The optimum geometrical parameters were found to be $D_1 = 2.88 \cdot 10^{-3}$ m, $D_2 = 2.15 \cdot 10^{-2}$ m, $L = 25.5$ m and $n = 15$. The number of inner tubes (n) has been limited to 15, because otherwise there is no turbulent flow anymore. The ΔT , ΔP_1 and ΔP_2 of this optimised heat exchanger are 1.42 K, 1.8 bar and 2.1 bar, respectively. Where ΔT is the temperature difference between the hot and cold stream, which is constant, because we have a balanced counter flow heat exchanger. The dense net life cycle irreversibility is $5.90 \cdot 10^2$ W for $D_1 = 2.88 \cdot 10^{-3}$ m, $D_2 = 2.15 \cdot 10^{-2}$ m, $L = 20.8$ m and $n = 15$. The ΔT , ΔP_1 and ΔP_2 of the optimised heat exchanger are 1.75 K, 1.5 bar and 1.7 bar, respectively.

The optimal geometrical parameters of the tube diameters are independent of the efficiency of the distribution net, the dense or wide net. However, the length of the tube is strongly dependent on the type of distribution net. The components of the life cycle irreversibility in the heat exchanger are displayed in Table 8.2.

Table 8.2 Components of life cycle irreversibility in Watts

component	wide net (KVM)	dense net (KVM)	wide net (minimised)	dense net (minimised)
thermal	6604	4392	363	296
mechanical	2	2	112	91
manufacture	47	47	249	203
Total	6653	4441	724	590

From the table it can be seen that the irreversibility due to the thermal component is reduced by increasing the irreversibilities due to the mechanical and manufacture component. The contribution of the use of copper, steel and PUR-foam to the irreversibility associated with the manufacture is 67%, 20% and 13% for the wide and dense net optimisation and 47%, 33% and 20% for the both heat exchangers with a length of 6.4 meter, respectively.

8.3.3 Discussion

The effect of the tube diameters and the length on life cycle irreversibility for the optimal geometrical parameters is shown in Figure 8.3 and 8.4. The figures show combinations of length and diameter for which the life cycle irreversibilities obtains values of 1000, 1500, 2000 and 2500 Watts. The optimums of 724 and 887 W of Figure 8.3 and 8.4, respectively, are marked.

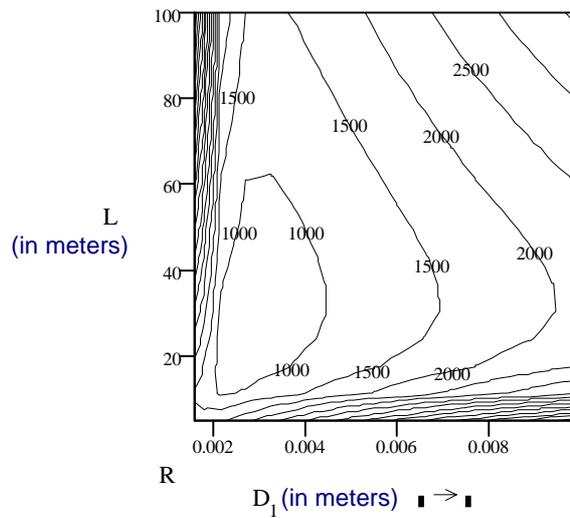


Figure 8.3 Life cycle irreversibility (in Watts) against length and diameter of a heat exchanger with 15 inner tubes ($n = 15$)

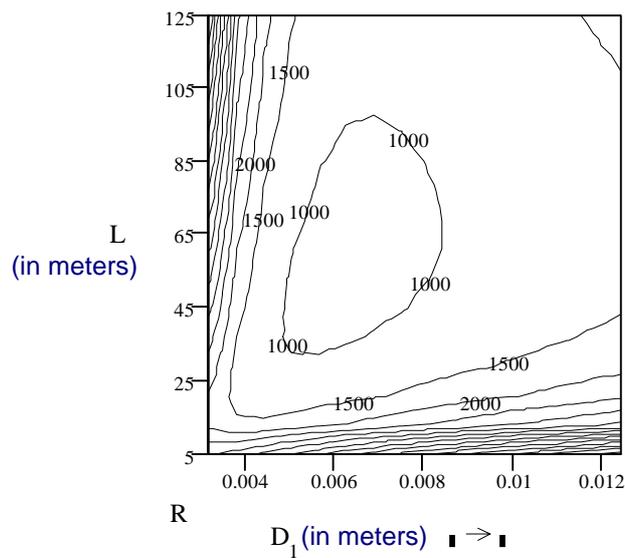


Figure 8.4 Life cycle irreversibility (in Watts) against length and diameter of a heat exchanger with three inner tubes ($n = 3$)

The cross sections of flow areas of the inner and outer passages are set fixed in the ratio of 1: 0.764 and 1 : 0.821, respectively, to get a 3 dimensional figure. This ratio has been obtained for the optimisation of the wide net with $n = 15$ and $n = 3$, respectively.

At zero length or at zero tube diameter the life cycle irreversibility rate becomes of course infinite. When the length or inner diameters are reduced to save materials cost it is shown that there is an optimal relation between these two geometrical parameters to do so. A decrease of the number of inner tubes gives the following optimal geometrical parameters for the minimal irreversibility, see Table 8.3.

Table 8.3 Optimal geometrical parameters and life cycle irreversibility for the wide net for different number of tubes

n (tubes)	i_{lc} (W)	D_1 (10^{-3} m)	D_2 (10^{-2} m)	L (m)
1	1059	10.5	1.69	90.6
2	943	7.58	1.75	67.4
3	887	6.25	1.81	56.6
5	825	4.90	1.89	44.0
10	757	3.50	2.04	31.6
15	724	2.88	2.16	25.6

An increase of the operating time of the heat exchanger leads to optimal geometrical parameters with greater diameters and a longer heat exchanger as shown in Table 8.4. It can be seen that an increase of the operating time leads to a relatively small increase of the tube diameters D_1 , D_2 and a bigger increase of the length. The life cycle irreversibility rate decreases steadily by an increase of the operating time. There are no significant differences between the wide and dense net for the relation between operating time and optimal geometrical parameters.

Table 8.4 Optimal geometrical parameters and life cycle irreversibility for the wide net with 15 inner tubes for different times

t (years)	i_{lc} (W)	D_1 (10^{-3} m)	D_2 (10^{-2} m)	L (m)
5	920	2.53	1.97	18.0
10	724	2.88	2.16	25.6
15	630	3.10	2.28	36.1

De Oliveira et al. (1991) gives for the exergetic optimisation a ratio for the exergy losses due the pressure drop and due to the heat transfer of 3 to 3.5. This ratio is 3.3 for the dense net optimisation, which has roughly the same exergetic cost factor as in the case of De Oliveira et al. (1991).

The number of inner tubes is limited so that the flow stays in the turbulent flow regime. The flow in the inner tubes leaves the turbulent flow regime above $n = 20$. The outer tube leaves the turbulent at $n = 15$, which is set as the upper limit. The entrance and exit pressure drops are neglected in this study. In the case of the wide net optimisation the entrance pressure drop is calculated to be lower than $6 \cdot 10^{-3}$ bar.

Zero heat transfer from the heat exchanger to the environment has been assumed. For the wide net optimisation the heat transfer to the environment during operation of the heat exchanger has been calculated to be 0.42% of the total amount of heat exchanged, when for the heat exchanger box an outside and a mean inside temperature of 20°C and 40°C are assumed, respectively and the coefficient of thermal conduction is taken to be 0.024 W/(m·K) for the polyurethane foam (PUR) and the heat transfer coefficient for the dry air is taken to be 10 W/(m²·K). So the neglect of the heat transfer from the heat exchanger box to the environment during operation is justified. However, if a constant temperature of 40°C outside the operation time is assumed the heat transfer to the environment becomes 20.2%. If the heat exchanger is located in a closed space, the temperature will increase, which will decrease the heat transfer. More attention should be devoted to this subject. A constant temperature in the heat exchanger during all times is desirable because of comfort conditions.

The transformation of mechanical energy into thermal energy has been neglected in this study. As can be calculated for the optimised situation for the wide net, which results in 15 inner tubes, this transformation would cause an temperature increase of 0.043 K for the cold side and 0.050 K for the hot side. Assuming an temperature increase of 0.045 K due to this transformation gives an associated exergy increase of 2.7 W. This would decrease the irreversibility due to the heat transfer with 11.0 W as the effect of the decrease of the required thermal energy.

8.4 Conclusions

With the use of the concepts of an ELCA the optimal design of a heat exchanger can be obtained. For all systems where there is a trade-off between exergy saving during operation and exergy destruction during construction of the system the minimisation of life cycle irreversibility should be used to get the true optimum from the point of view of conservation of exergy reservoir of natural resources.

In the case under study the optimal design parameters of the heat exchangers are obtained under the specified conditions. The dense net, which is a more energy efficient heat supply system than the wide net, has the same inner tubes diameters as the wide net whilst the length of the heat exchanger is smaller for the former than for the latter. The decrease of the number of inner tubes leads to a relatively small increase of the irreversibility. The increase of the operating time leads to a slight increase in the inner tubes diameters and a greater increase in the length of the heat exchanger. In the

optimised situation the life cycle irreversibility is more uniformly distributed between the component irreversibilities than in the fixed length of 6.4 meters situation.

In general we can conclude that the thermodynamic optimisation of the design parameters of a subsystem is dependent of the thermodynamic efficiency of the whole system and that the different components of the life cycle irreversibility of heat exchangers are more uniformly distributed when there are fewer restrictions on the design parameters for the optimisation.

8.5 Nomenclature

A	= heat transfer area
c_p	= heat capacity
C	= cumulative exergy destruction
D	= inner diameter of the tube
d	= thickness of the tube wall
I	= irreversibility or exergy destruction
k_p	= exergetic cost for the pressure rise
k_{power}	= exergetic cost for power generation
k_{pumps}	= exergetic cost for pumping
k_T	= exergetic cost for the heat
L	= length of the tubes
M	= mass
\dot{m}	= mass flow
Nu	= Nusselt number
N_{tu}	= number of heat transfer units
P	= pressure
Pr	= Prandtl number
Re	= Reynolds' number
R	= diameter of the tube curving
T	= temperature
T_1	= inner tube temperature
T_2	= outer tube temperature
\bar{u}	= mean velocity of the fluid in the tube
x	= recycling ratio of the material

Greek letters

α	= heat transfer coefficient
e	= effectiveness
ρ	= density
λ	= coefficient of thermal conduction
μ	= dynamic viscosity

Subscripts

0	= environmental
1	= inner tube of the heat exchanger
2	= outer tube of the heat exchanger
Cu	= copper
h	= hydraulic
S	= steel
in	= inlet
LC	= life cycle
man	= manufacturing
mat	= material
n	= number of inner tubes
oper	= operating
out	= outlet
pri	= primary
PUR	= polyurethane foam
sec	= secondary
tot	= total
w	= welding

Superscripts

ΔP	= mechanical component
ΔT	= thermal component

8.6 References

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Life cycle assessment (LCA) and exergetic LCA of the disposable cup and the porcelain mug

Abstract

As a final example the exergetic life cycle assessment (ELCA) of consumer products, the disposable polystyrene cup and the porcelain mug, has been performed. The ELCA, including a Zero-ELCA, is compared with the LCA, which is carried out with the aid of the software package SimaPro 3.1. The LCA shows that for the nine analysed environmental effects the porcelain mug has a lower impact on all, except for eutrofication, than the disposable cups.

The ELCA shows that the disposable polystyrene cups make two times greater use of the exergy reservoir of natural resources than the porcelain mug. With the Zero-ELCA is displayed that only a relatively small increase in the use of natural resources is needed to avoid harmful emissions. It is shown that the Zero-ELCA is a valuable approach because it gives a good indication of the effort in terms of natural resources and process development needed to prevent emissions and that other indicators, such as the Eco-indicator 95, are more arbitrary as far as combining different environmental effects is concerned.

9.1 Introduction

In recent years much discussion has taken place on the question which cup is preferable from an environmental point of view: the disposable polystyrene cup or the porcelain mug. A study has been performed for the Dutch Government to give an answer to this question. The outcome of this study was that the air pollution, energy use and amount of solid waste was lower for the porcelain mugs than for the polystyrene and paper disposables, while the water pollution was higher for the porcelain mug (Eijk et al., 1991). No split up of specific environmental effects has been made in this study. So, in this chapter first a complete environmentally orientated LCA is performed. In addition an ELCA has been carried out to include the effect of depletion of natural resources and a Zero-ELCA is performed to demonstrate the consequences of a zero emission process.

9.2 Method

The LCA was performed with the aid of SimaPro 3.1. This is a computer program to support the performing of LCA's. Its database contains data of use of natural resources and emissions of harmful materials for many kinds of production processes, energy conversion processes, transportation and disposal handling. The program offers the ability to add data (Goedkoop, 1994). The program does not use a closed material and energy balance. SimaPro 3.1 takes the following environmental problems into account as listed in Table 9.1.

Table 9.1 The normalisation and Eco-indicator 95 weight factors

environmental problem	emission per European individual (kg/year)	Evaluation weight factor
green house effect	1310 GWP	2.5
depletion of the ozone layer	0.926 ODP	100
acidification	113 AP	10
eutrofaction	38.2 NP	5
heavy metals	0.0543 Pb-equiv.	5
carcinogenic	0.0109 PAH-equiv.	10
winter smog	94.6 SO ₂ -equiv.	5
summer smog	17.9 POCP	2.5
pesticides	0.966 useful substance	25

These nine environmental effects are normalised by looking at the average value, which each European individual add to the specific environmental problem on a yearly basis. The Eco-indicator 95 evaluation method of SimaPro 3.1 as developed by PRé (Goedkoop, 1995) has been used to express the environmental problems in one indicator by multiplying the normalised values with a specific factor as listed in Table 9.1, which is determined by how far the environmental effect is away from the desired level. The problem associated with this evaluation method is that the determination of the desired level for the different environmental effects is a subjective process. The death of 1 human being per million has been attributed the same weight in the evaluation method as damaging 5% of an ecosystem. More details are given in Goedkoop (1995). Depletion is not taken into account in SimaPro 3.1, but the underlying LCA method as described in chapter 5 would make it possible. The method of ELCA and Zero-ELCA has been extensively discussed in chapter 5.

9.3 Life cycles

The functional unit is taken to be the use of 3000 cups. A repeated use of 3000 times is taken for the porcelain mug, which is the average life time, while the polystyrene cups (PS cups) are used only once. For most processes the Dutch situation is assumed.

The PS cups consist for about 49%¹⁰ of High Impact Polystyrene (HIPS) and 49% of General Purpose Polystyrene (GPPS). An amount of 2% of TiO₂ is added for the colouring of the cup. The mass of a PS cup is 4 gram. The transport of the different materials is taken into account, the packing material of the PS cups and porcelain mugs is not taken into account in this analysis.

Polystyrene is made out of crude oil. The last steps in the production process are the forming of ethylbenzene out of ethylene and benzene, and the forming of styrene by dehydrogenating the ethylbenzene. Subsequently, polystyrene is created by the polymerisation process. The disposable polystyrene cups are formed by a foil-extrusion process. The inner and outer layer, which are together 30%, are made of primary material, while the intermediate layer is produced from production waste. The electricity use for the foil-extrusion process is 4 MJ per kg of cups. After usage the cups end up in a municipal waste flow of which 48% is incinerated and 52% is dumped at a waste site, as is the present situation (1995) in the Netherlands.

Porcelain is made out of a mixture of minerals, which consists mainly out of kaolin (50%), feldspar (25%) and quartz (25%). Kaolin and feldspar consists mainly of Al₂O₃ and SiO₂ in different amounts. Quartz consists mainly of SiO₂ and is used as a melting material. The mixture of minerals is brought in the desired form and subsequently heated in an industrial furnace to 900 °C to obtain a porcelain mug. Glazing is added to the mug and the porcelain mug is finished with a heat treatment at 1400°C. During this treatment the mug shrinks 15%. The mass for the porcelain mug is 250 gram. For the production of 1 kg of porcelain mugs 1 m³ of natural gas is used. After each use, the porcelain mug is washed in an industrial dishwasher. In this type of dishwasher a detergent containing 30% of phosphates is used. The electricity use is 45 kJ and detergent use is 0.25 gram for each time the mug is washed (van Eijk et al., 1991). After usage this mug ends up in a municipal waste flow which is dumped at a waste site.

9.4 LCA

9.4.1 Inventory analysis, classification and evaluation

The inventory analysis, made with aid of SimaPro 3.1, results in a list of natural resources taken from the environment and emissions brought into the environment during the life cycles of the products, as can be seen in Table 9.2, 9.3, 9.4 and 9.5. The

¹⁰ All percentages are mass percentages

material balance has been checked and corrected for hydrocarbons by comparing the estimated carbon content of the used fossil fuels with the carbon content of the emissions, which consist mainly of CO₂, as can be seen in the Table 9.3 and 9.4. It is assumed that flue gas cleaning is not taking place in the LCA and ELCA. It will be introduced in the Zero-ELCA to emphasise the difference between the ELCA and the Zero-ELCA. Avoided use of fossil fuels by the incineration of waste in a power plant has been taken into account. All transportation of materials during the life cycles has been taken into account.

Table 9.2 Raw materials inputs of both life cycles in kg

inputs	LC 3000 PS cups	LC porcelain mug
bauxite (ore)	2.23e-02	0
clay	2.35e-04	2.53e-01
coal	3.78e-01	4.38e+00
ilmenite	4.80e-01	0
iron (ore)	4.25e-03	0
limestone mineral	6.31e-03	0
natural gas	4.30e-01	6.73e+00
crude oil	2.36e+01	1.44e-03
rock salt	4.28e-01	0

Table 9.3 Emissions to air and water for life cycle of the porcelain mug

emitted substance	emitted to	Mass (kg)	Eco-indicator 95	% of total	cumulative %
NO	air	1.72e-01	1.36e-02	37.269	37.269
SO ₂	air	6.58e-02	9.33e-03	25.568	62.837
phosphate	water	4.50e-02	5.89e-03	16.141	78.978
CO ₂	air	2.96e+01	5.67e-03	15.538	94.517
C _x H _y	air	2.34e-02	1.30e-03	3.563	98.079
dust (SPM)	air	5.63e-03	2.98e-04	0.817	98.896
N ₂ O	air	4.55e-03	2.35e-04	0.644	99.540
methane	air	4.29e-02	1.32e-04	0.362	99.901
ammonia	air	1.20e-04	2.53e-05	0.069	99.971
Pb	water	7.50e-08	6.90e-06	0.019	99.990
aldehydes	air	4.04e-05	2.50e-06	0.007	99.996
Ba	water	1.00e-07	1.28e-06	0.004	100.000
CO	air	2.10e-02			
Total			3.649e-02		

Table 9.4 Emissions to air and water for life cycle of 3000 PS cups

emitted substance	emitted to	Mass [kg]	Eco- indicator 95	% of total	cumulative
SO ₂	air	4.28e-01	6.07e-02	60.320	60.320
NO	air	3.04e-01	2.41e-02	23.973	84.293
CO ₂	air	5.51e+01	1.05e-02	10.445	94.738
C _x H _y	air	8.10e-02	4.49e-03	4.466	99.204
Cr	water	1.05e-05	1.94e-04	0.193	99.397
Pb	water	1.68e-06	1.55e-04	0.154	99.552
dust (SPM)	air	2.36e-03	1.25e-04	0.124	99.676
COD	water	1.88e-02	5.42e-05	0.054	99.730
heavy metals	air	5.77e-07	5.31e-05	0.053	99.783
HCl	air	6.15e-04	4.80e-05	0.048	99.830
toluene	air	4.78e-04	3.75e-05	0.037	99.868
NH ₄ ⁺	water	7.05e-04	3.05e-05	0.030	99.898
benzene	air	8.09e-04	2.95e-05	0.029	99.927
fluoranthene	air	2.88e-08	2.64e-05	0.026	99.954
N ₂ O	air	2.48e-04	1.28e-05	0.013	99.966
methane	air	2.37e-03	7.30e-06	0.007	99.974
Cd	water	2.16e-08	5.96e-06	0.006	99.980
Hg	water	5.23e-09	4.81e-06	0.005	99.984
ethylbenzene	air	6.62e-05	4.15e-06	0.004	99.988
benzo[a]pyrene	air	2.88e-09	2.64e-06	0.003	99.991
styrene	air	2.42e-05	2.57e-06	0.003	99.994
HF	air	1.76e-05	2.50e-06	0.002	99.996
ammonia	air	8.90e-06	1.87e-06	0.002	99.998
carbon black	air	1.55e-05	8.24e-07	0.001	99.999
Cu	water	1.68e-06	7.74e-07	0.001	100.000
aldehydes	air	6.85e-06	4.24e-07	0.000	100.000
C _x H _y aromatic	air	3.88e-07			
CO	air	2.30e-02			
Total			1.0031e-01		

Table 9.5 Emissions to the soil for both life cycles in kg

Emission	LC 3000 PS cups	LC porcelain mug
chemical waste	2.40e-02	6.00e-04
final waste (inert)	6.24e+00	1.30e-01
production waste (not inert)	1.04e+00	2.37e+00
slag	1.15e-01	1.20e-01

9.4.2. Results

The results are displayed in Tables 9.6 and 9.7 and Figure 9.1. Figure 9.1 is another form of representation of the data of Table 9.6. The abbreviations used in Table 9.6 and 9.7 are listed in the Nomenclature. The LCA shows that for the nine analysed environmental problems the porcelain mug has a lower impact on all, except for the problem of eutrofication, than the disposables. In Appendix 6 it is shown which emissions cause the different environmental effects. Using the Eco-indicator 95 the value for the PS cups is 2.75 times higher than for the porcelain mug as can be seen from Table 9.3 and 9.4.

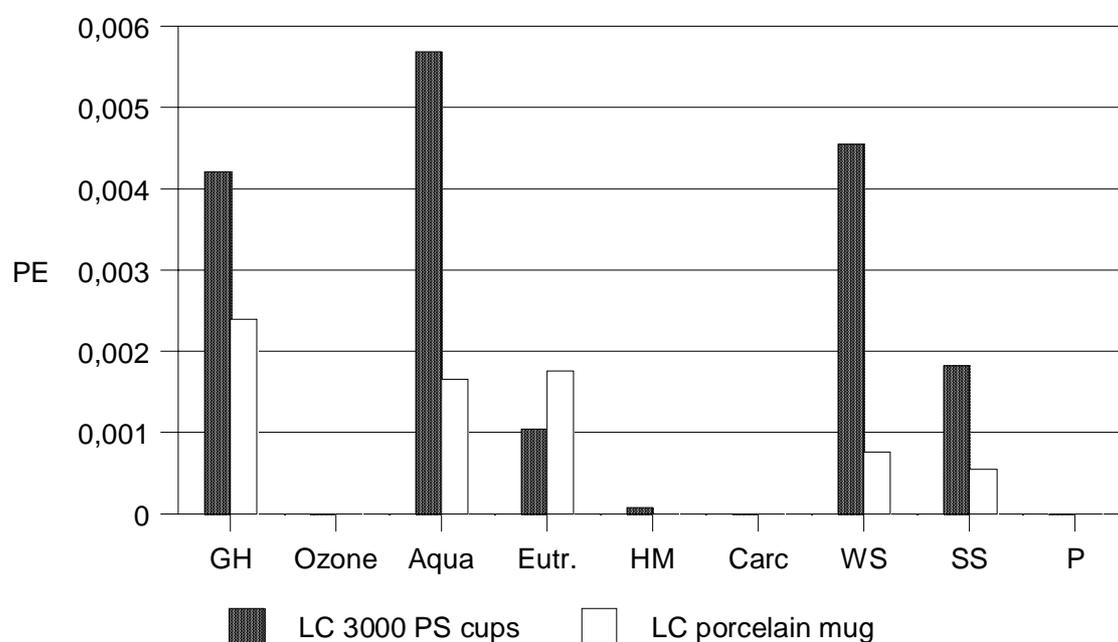


Table 9.6 The normalised values for the different environmental effects

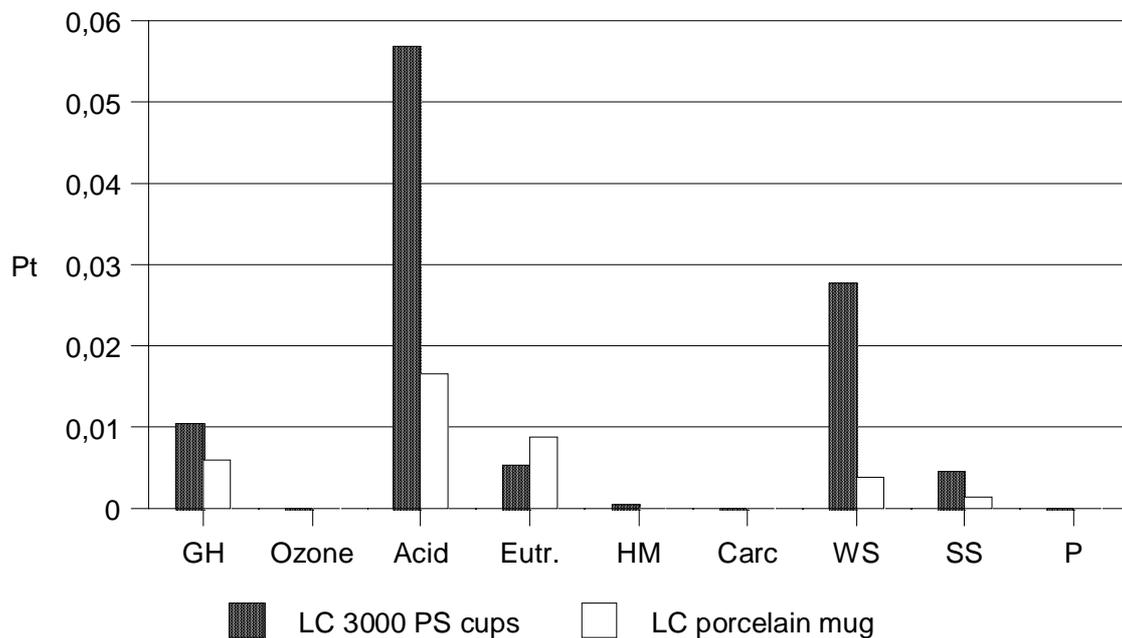


Table 9.7 The evaluated environmental effects according to the Eco-indicator 95

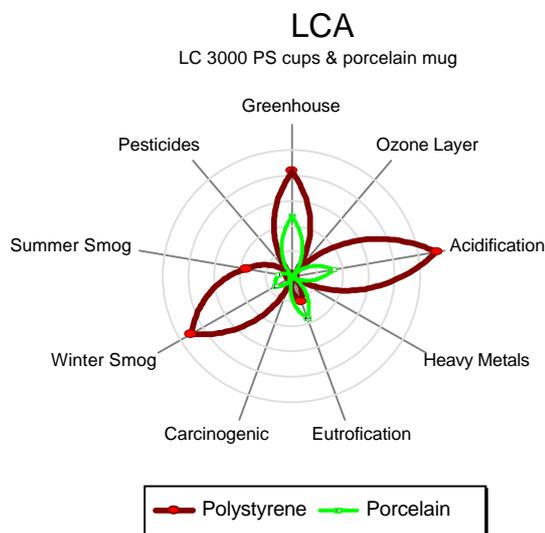


Figure 9.1 The normalised environmental effects, each circle is, starting from the center, an increase of 0.001 PE

From the normalisation in Table 9.6 it can be seen that the environmental effects of green house effect, acidification, eutrofication, winter and summer smog have a significant impact. The first four environmental effects are mainly caused by SO₂, NO_x and CO₂ in the case of PS cups as can be seen in Appendix 6. Most of these emissions take place in the polystyrene production, namely 98% of the SO₂, 95% of the NO_x and 62% of the CO₂. Especially, the distillation of crude oil has a high SO₂ emission. Most of the

remaining part of the CO₂ is emitted during the waste incineration process, but it replaces electricity generation and its associated emissions. For the porcelain mug, it can be seen in Appendix 6 that the main environmental effects are caused by SO₂, NO_x, phosphate and CO₂. Almost all SO₂, NO_x, and CO₂ are emitted in the electricity production process. The emission of phosphate is caused by the dish washer. The environmental effects of the production of the porcelain mug are very small (less than 1% of the value of the Eco-indicator 95).

9.5 ELCA

For the ELCA the black-box approach has been used. For the sake of simplicity a simplified mass balance has been used, which take into account only the main flows as can be seen in Tables 9.8 and 9.9.

Table 9.8 Main incoming flows of the two life cycles in kg

Components	PS cup	porcelain mug
coal	0.378	4.700
crude oil	23.600	
natural gas	0.685	7.478
oxygen	63.961	34.615
phosphate		0.045
total	88.624	46.793

Table 9.9 Main outgoing flows of the two life cycles in kg

Components	PS cup	porcelain mug
CO ₂	55.100	29.600
H ₂ O	26.520	14.773
SO ₂	0.428	0.066
PS waste	6.115	
NO	0.304	0.172
N ₂	0.020	1.701
ash	0.032	0.395
CO	0.023	0.021
C _x H _y	0.081	0.023
CH ₄		0.043
phosphate		0.045
total	88.624	46.793

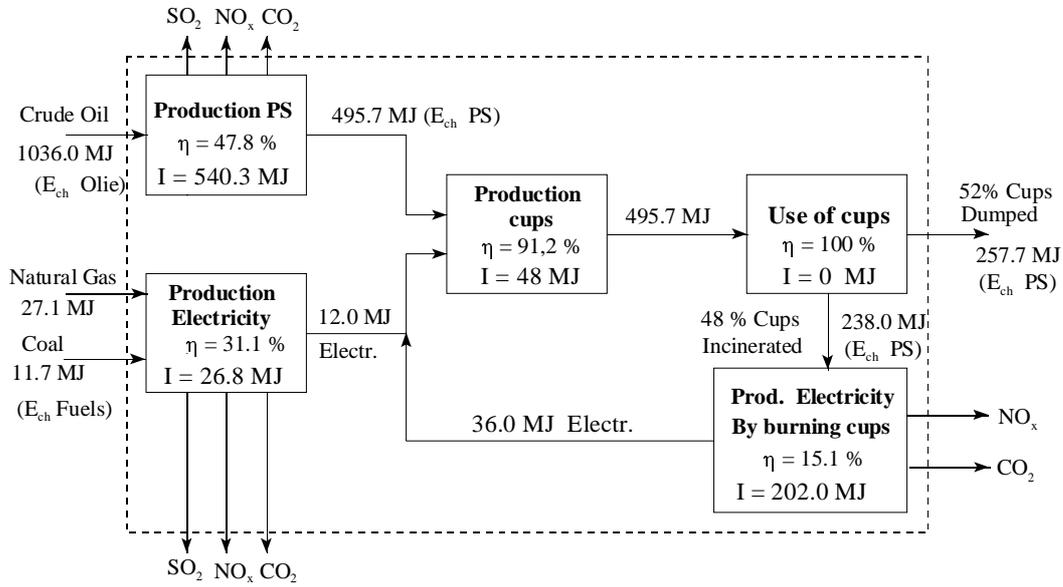


Figure 9.2 Exergy diagram of the life cycle of the 3000 disposable PS cups

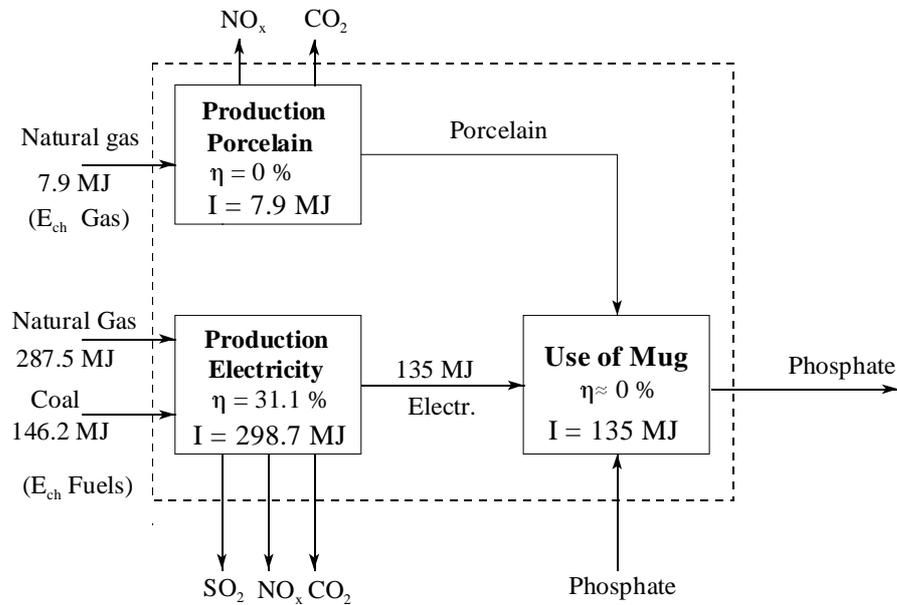


Figure 9.3 Exergy diagram of the life cycle of the porcelain mug

Other flows with a small mass flow or a low exergy content, like clay, are neglected. The exergy of the incoming natural resources has been determined. The exergy of the emissions, except one, was considered to be negligible. It is assumed these emissions diffuse into the environment and dissipate. Only the polystyrene flow that ends up at the waste site was taken into account, because this plastic waste forms an exergy reservoir that might be used in the future. The life cycle irreversibility for the 3000 polystyrene cups is 817 MJ, while the life cycle irreversibility for the porcelain mug is 442 MJ, a factor 1.85 less, as can be seen in Figure 9.2 and 9.3. The main causes of irreversibility for the PS cups are the production of polystyrene and the burning of the waste in a waste incineration plant. The main irreversibilities in the case of the porcelain mug take place in the electricity generation and the dish washer.

9.6 Zero-ELCA

To make a Zero-ELCA of the two products the production processes have to be transformed into zero-exergy emission processes. As discussed before the four major contributors to the pollution of these two products are SO₂, NO_x, CO₂ and phosphate. These substances form together 95% of the environmental impact of the emissions caused by the life cycle of the two products according to the Eco-indicator 95. The abatement exergy for these emissions has been determined. For SO₂ the abatement exergy was found to be 57 MJ/kg based on 90% SO₂ removal with limestone in a flue gas desulphurisation unit (FGD) in a coal fired power plant. The limestone is converted into gypsum in the FGD. For NO_x the abatement exergy was found to be 16 MJ/kg, based on a 80% removal in a DeNO_x unit in a coal fired power plant. In this unit the NO_x react with NH₃ to N₂ and H₂O. It is well known that these abatements techniques are now applied everywhere in Western Europe. For CO₂ a figure of 3 MJ/kg is calculated for the case of separation of 90% CO₂ out of the flue gases, compression and storage in empty gas fields according to Göttlicher et al. (1996). This technique would decrease the exergetic efficiency of the power plant with 15 percentage points and is not yet in use. Finally, for phosphate a figure of 18 MJ/kg was found for 99% removal (STOWA, 1996). The total abatement exergy for each emission is given in Table 9.10.

The zero-exergy emission life cycle irreversibility for the 3000 polystyrene cups is 992 MJ. This is an increase of 21% compared to the situation without emission reduction. The zero-exergy emission life cycle irreversibility of the porcelain mug is 528 MJ. This is an increase of 20% compared to the situation without emission reduction and a factor 1.88 less than zero-exergy emission life cycle irreversibility for the 3000 polystyrene cups. The results of the reduction of emissions on the environmental effects are graphically shown in Figure 9.4 for the case of no emission reduction and in Figure 9.5 and Table 9.11 with emission reduction. Figure 9.4 displays the results of the LCA, including an ELCA, while Figure 9.5 represents a LCA, which include a Zero-ELCA. The normalised value for the irreversibility rate in Europe is calculated to be 160 GJ per European a year according to the normalisation method of Goedkoop et al. (1995). The

calculation is based on the determined irreversibility rates in Italy and Sweden¹¹ according to Wall (1987) and Wall et al. (1994). The normalised value for the depletion should be considered as an estimate, because the irreversibility rates of Italy and Sweden are obtained in 1990 and 1980, respectively, while Goedkoop et al. (1995) use values of 1988 for the extrapolation on basis of energy use. This gives a normalised value of 0.00511 and 0.00276 PE for the depletion of natural resources for the PS cups and porcelain mug, respectively, as displayed in Figure 9.4. The value of the Eco-indicator 95 (excluding depletion of natural resources) is reduced to 0.0104 and 0.00365 for the PS cups and the porcelain mug, respectively, as can be seen in Table 9.11. The ratio of the Eco-indicator 95 values of the cups stays nearly constant in the Zero-ELCA as compared with the LCA.

Table 9.10 Abatement exergy of the main harmful emissions in MJ

components	PS-cups	porcelain mug
CO ₂	148.8	79.9
SO ₂	22.0	3.4
NO _x	3.9	2.2
phosphate		0.8
total	174.7	86.3

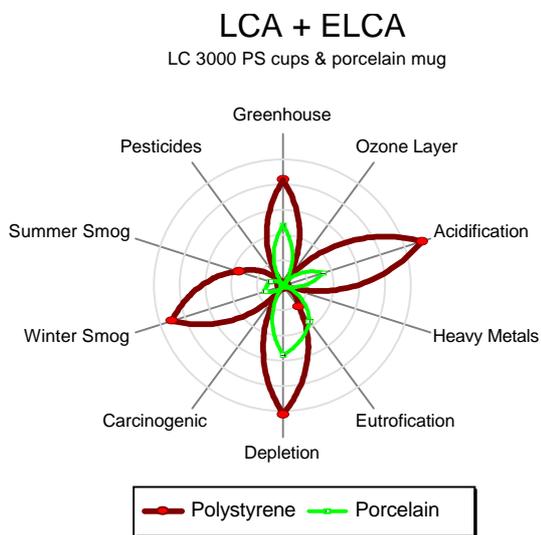


Figure 9.4 The normalised environmental effects, including depletion, each circle is, starting from the center, an increase of 0.001 PE.

¹¹ The irreversibility rate in Italy and Sweden is 8839 PJ/year. Extrapolating on the energy use gives a total irreversibility rate of 79517 PJ/year for Europa. Dividing by the 497 million inhabitants of Europe gives the value of 160 MJ per European a year

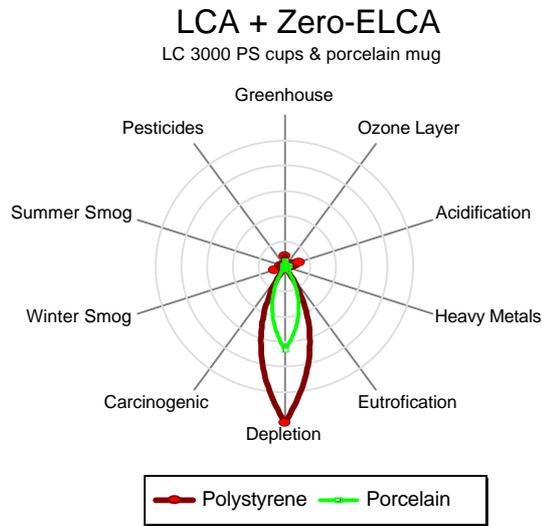


Figure 9.5 The normalised environmental effects in the Zero-ELCA, each circle is, starting from the center, an increase of 0.001 PE.

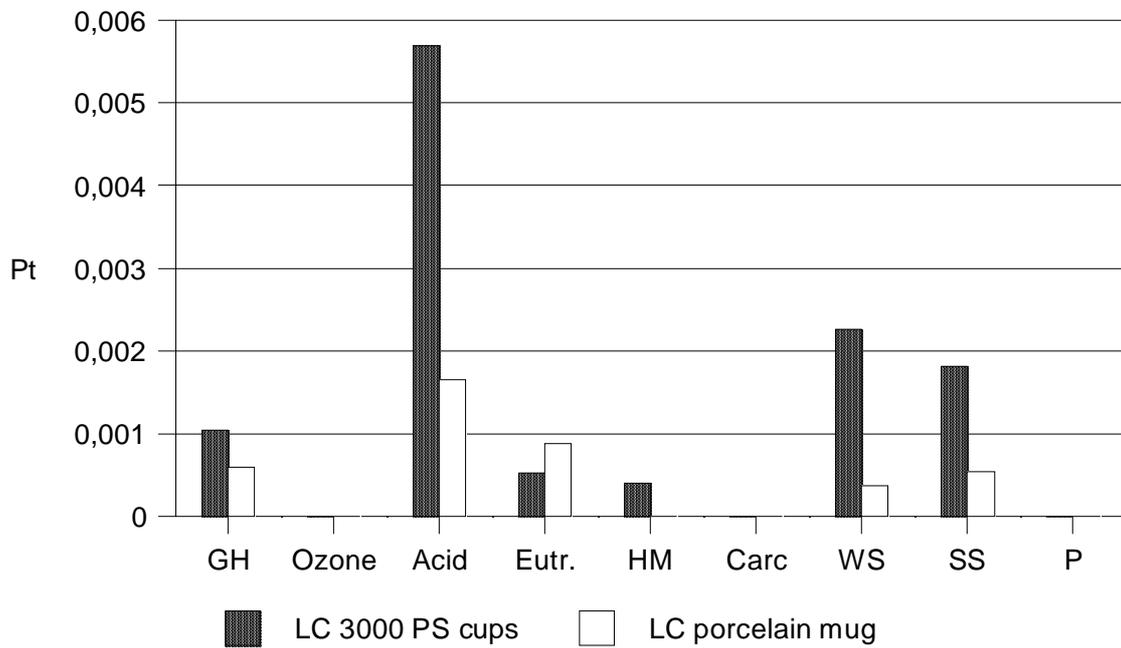


Table 9.11 The evaluated environmental effects in the Zero-ELCA according to the Eco-indicator 95 method

9.7 Discussion

In the ELCA of the porcelain mug the irreversibility due to the production of 750 gram of detergent has been neglected. In a more detailed analysis this should be included. The exergetic efficiency of the electricity production in both life cycles is too low.

The abatement exergy is determined when a reduction of 80% to 95% of the harmful emissions is obtained. There is no 100% reduction of the emissions. More research has to be done to determine the abatement exergy for 100% reduction. However, it could be argued that a complete 100% reduction of the emissions is not needed, because the sustainable emission levels require a lower reduction as shown in Weterings et al. (1992).

The environmental effects are analysed when there is no use of emission reduction techniques, like desuphurisation. In practice these techniques are in use in Western Europe. So no realistic picture is given for the environmental effects of the disposable cup and the porcelain mug. However, the comparison of the disposable cup and porcelain mug does not change much as these emission reduction techniques are included as shown in Table 9.7 and 9.11.

Because the main harmful emissions are caused by the burning of fossil fuels, which is also the main cause of irreversibility, the ELCA and the Zero-ELCA give the same ratio between irreversibilities of the two life cycles.

9.8 Results

The LCA shows that for the nine analysed environmental problems the porcelain mug has a lower impact on all, except for the problem of eutrofication, than the disposable cups. From the ELCA it can be concluded that the disposable PS-cups make two times greater use of the exergy reservoir of natural resources than the porcelain mug. The Zero-ELCA shows the environmental impact of the life cycles expressed in terms of the use of natural resources: the zero-exergy emission life cycle irreversibility. It is shown that with a relatively small increase in the use of natural resources the harmful emissions can be reduced greatly. The Eco-indicator 95, based on a subjective evaluation, and the Zero-ELCA, based on the effort required to reduce the emissions give roughly the same results, because the main harmful emissions are caused by the burning of fossil fuels.

9.9 Conclusions

By including the environmental effect of depletion of natural resources the ELCA is a valuable addition to the LCA. The Zero-ELCA shows that only a relatively small increase in the use of natural resources is needed to avoid 80 to 95% of the harmful emissions and reduce the associated environmental effects greatly. The Zero-ELCA is an objective criterion taken into account the irreversibility and the effort required to reduce

the environmental effects. So, an objective parameter for the aggregation of the different environmental effects is obtained and it should be considered to use this instead of arbitrary criteria, like the Eco-indicator 95.

9.10 Nomenclature

Acid	=	Acidification
AP	=	Acidification potential
BOD	=	Biological oxygen demand
Carc	=	Carcinogenic
COD	=	Chemical oxygen demand
Equiv.	=	Equivalent
Eutro	=	Eutrofication
GH	=	Green house effect
GWP	=	Global warming potential
HM	=	Heavy metals
LC	=	Life cycle
NP	=	Nitrification potential
ODP	=	Ozone depletion potential
P	=	Pesticides
PAH	=	Poly-aromatic hydrocarbons
PE	=	Personal equivalent
POCP	=	Photochemical ozone creation potential
PS	=	Polystyrene
Pt	=	Value of the Eco-indicator 95
SPM	=	Suspended particle matter
SS	=	Summer smog
WS	=	Winter smog

Greek

η = exergetic efficiency

9.11 References

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Conclusions

One of the keystones for obtaining sustainable development is the use of exergy analysis. Energy can never be lost as is stated in the first law of thermodynamics. Exergy can be lost and this loss is called irreversibility or exergy destruction. An important aspect of sustainable development is the minimisation of irreversibilities caused by the use of non-renewables. This irreversibility is minimised on different levels.

This is done with the aid of a meaningful and practical criterion of performance. For obtaining this criterion, which is called the rational efficiency, the function of the system has to be defined. The value of this exergetic efficiency is shown for a heat exchanger, where it is demonstrated that its behaviour is completely different from the effectiveness, a first law efficiency which is widely used nowadays.

By performing two conventional exergy analyses on process units the rational efficiencies and irreversibilities are obtained. It is shown that in the case of crude oil distillation great improvements are possible, while these are more limited in the case of cryogenic air separation. However, to minimise the total irreversibility, i.e. the life cycle irreversibility, not only processes that occur within the system have been shown to be taken into account, but also all kinds of interaction between energy and material flows outside the systems boundaries. It is demonstrated that the minimisation of the life cycle irreversibility is an important keystone to obtain sustainable development. The life cycle irreversibility is the measure for depletion of natural resources. This approach has been worked out in the Exergetic Life Cycle Analysis (ELCA).

An ELCA has been performed to optimise at the level of production processes, namely the textile washing machine. The minimisation of the life cycle irreversibility by varying the detergent use and washing temperature leads to the most efficient way of washing. Further reduction of the life cycle irreversibility can be achieved by replacing the electrical heating system by gas fired heating or district heating. The ELCA can also be used to compare different types of production processes as shown in the case of district heating systems, where as the reference situation high efficiency boilers are taken. The concept of exergetic pay back time is introduced to obtain the most exergy efficient investment. A financial analysis is included to obtain the most cost effective and exergy efficient system.

As an example of an ELCA of a single component, the tap water heat exchanger in a district heating system, is chosen. It is shown that the ELCA leads to the optimal design of a heat exchanger, where the influence of the systems parameters on the geometrical parameters are displayed. The life cycle irreversibility of this heat exchanger is, compared to the present situation, greatly reduced.

A comparison is made with the present environmentally orientated LCA and it is concluded that the ELCA should be part of the LCA to include the depletion of natural resources. A Zero-ELCA, based on zero-exergy emission processes, is introduced as an indicator for the effort required to reduce the emissions. This indicator, based on physical principles, can replace other indicators which are based on arbitrary criteria, like the Eco-indicator 95.

Appendices

Appendix 1: Bibliography of publications

The database

The abstracts have been collected from the DIALOG Database Energy Science & Technology File 103. This file is a multidisciplinary file containing world-wide references to basic and applied scientific and technical research literature and is one of the worlds biggest databases of abstract on the field of energy and related topics. The database contains more than 2.5 million abstracts of articles, report's, books, conference papers, patents and theses. This information has been provided by the US Department of Energy, its contractors, other government agencies, professional societies, and through the International Energy Agency's multilateral information programme and the International Atomic Energy Agency's. Approximately 50% of the references are from non-US sources.

The search action had taken place on the following words:

- EXERG?
- AVAILABILITY and SECOND (?) LAW

The set was shortened to the period 1985 - 1994. This gives a set of **825 abstracts**. These are stored in three files on two diskettes. The files are executable files and the instructions how to handle the program can be found in the help menu. The organisation of the files on the discs is as follows:

Disc 1: file:	lit1.exe	content:	Item 1 - 296
Disc 2: file:	lit2.exe	content:	Item 297 - 625
	lit3.exe		Item 626 - 825

The following abbreviations, listed in Table A.1, have been used in the files.

Table A.1 The abbreviations of the used indexes on the file

AB	Abstract
AN	Patent Application Number and Accession Numbers
AU	Author or for conference proceedings editor
AV	Availability
BN	International Standard Book Number (ISBN)
CL	Conference Location
CO	Code
CP	Country of Publication
CS	Corporate Source
CT	Conference Title
CY	Conference Year
DE	Descriptor
DT	Document Type
FN	File Number
GL	Geographic Location (Country or Origin)
JA	Journal Announcement
JN	Journal Name
LA	Language
PD	Patent Date
PG	Pages
PN	Patent Number
PR	USDOE Project/Non-DOE Project
PU	Publisher
PY	Publication Year
RN	Report Number
SC	Subject Code
SF	Sub File
SN	International Standard Serial Number (ISSN)
TI	Title
VO	Volume

Appendix 2: Flows in the distillation unit and the liquefaction unit

A2.1 Flows of components in the distillation unit

The minus sign in the mass rate means the flow is leaving the unit.

Table A2.1 The in- and outgoing flows of the lower distillation column (LC)

Components	Temp. (in °C)	Press. (in bar)	Flow (kg/sec)	Enthalpy (in kW)	Exergy (in kW)
cold air (AIRC)	-172.6	5.85	16.20	-3373.5	4688.0
Liquefied nitrogen (LQN)	-177.1	5.80	4.73	-1814.9	3188.3
Heat condenser	-177.0			-2097.8	4403.9
Top nitrogen (TN)	-177.0	5.85	4.73	1043.7	-1568.6
Distillate (D1)	-177.5	5.85	-7.00	2748.3	-4848.1
Bottom (B1)	-172.5	6.03	-9.20	3494.3	-5801.5
Total				0.1	62.0

Table A2.2 The in- and outgoing flows of the upper distillation column (UC)

Components	Temp. (in °C)	Press. (in bar)	Flow (kg/sec)	Enthalpy (in kW)	Exergy (in kW)
JT1out	-188.6	1.50	9.20	-3513.1	5745.8
JT2out	-191.6	1.60	7.00	-2841.1	5021.8
GOXC	-178.1	1.67	-2.82	527.4	-813.0
LOX	-178.1	1.67	-0.70	275.4	-510.7
Cold Purge (PCin)	-191.8	1.43	-3.49	782.8	-760.4
Cold Gan (GCin)	-192.8	1.40	-6.20	1422.2	-1472.4
LIN	-192.8	1.40	-2.86	1213.5	-2190.6
Crude Argon (CRAR)	-184.2	1.21	-0.13	15.0	-47.8
Heat reboiler	-178.1			2097.8	-4486.0
Total				-20.1	486.7

Table A2.3 The in- and outgoing flows of the heat exchange reboiler/condenser

Components	Temp. (in °C)	Press. (in bar)	Flow (kg/sec)	Enthalpy (in kW)	Exergy (in kW)
Heat reboiler	-178.1			2097.8	4486.0
Heat condenser	-177.0			-2097.8	-4403.9
Total				0	82.1

Table A2.4 The in- and outgoing flows of the throttling valve 1 (JT1)

Components	Temp. (in °C)	Press. (in bar)	Flow (kg/sec)	Enthalpy (in kW)	Exergy (in kW)
JT1in	-173.5	6.03	9.20	-3513.1	5838.9
JT1out	-188.6	1.50	-9.20	3513.1	5745.8
Total				0	92.3

Table A2.5 The in- and outgoing flows of the throttling valve 2 (JT2)

Components	Temp. (in °C)	Press. (in bar)	Flow (kg/sec)	Enthalpy (in kW)	Exergy (in kW)
JT2in	-183.5	5.85	7.00	2841.1	5053.9
JT2out	-191.6	1.60	-7.00	2841.1	5021.8
Total				0	32.1

*Table A2.6 The in- and outgoing flows of the heat exchanger
(not displayed in Figure 4.3)*

Components	Temp. (in °C)	Press. (in bar)	Flow (kg/sec)	Enthalpy (in kW)	Exergy (in kW)
Bottom (B)	-172.5	6.03	9.20	-3494.3	5801.5
Distillate (D)	-177.5	5.85	7.00	-2748.3	4848.1
Cold Purge (PCin)	-191.8	1.43	3.49	-782.8	760.4
Cold Gan (GCin)	-192.8	1.40	6.20	-1422.2	1472.4
JT1in	-173.5	6.03	-9.20	3513.1	-5838.9
JT2in	-183.5	5.85	-7.00	2841.1	-5053.9
Cold Gan (Gcout)	-180.6	1.43	-3.49	743.8	-665.9
Cold Purge (PCout)	-182.0	1.40	-6.20	1349.6	-1292.2
Total				0	31.5

A2.2 liquefaction unit

Table A2.7 The in- and outgoing flows of heat exchanger 1

Components	Temp. (in °C)	Press. (in bar)	Flow (kg/sec)	Enthalpy (in kW)	Exergy (in kW)
TOPN1 (IN1)	17.0	5.8	3.50	-34.9	626.3
Nitrogen 2 (N2)	-95.0	5.6	22.93	-2946.1	4863.5
Compressor3out (C3out)	33.0	45.8	18.11	-46.4	6541.8
Nitrogen 3 (N3)	33.0	37.4	8.32	-5.9	2858.8
Compressor1in (C1in)	26.0	5.5	-26.43	10.8	-4577.2
Nitrogen 4 (N4)	-88.5	45.4	-18.11	2682.9	-7190.3
HETin	-3.0	37.2	-8.32	339.6	-2867.0
Total				0	255.9

Table A2.8 The in- and outgoing flows of heat exchanger 2 (HE 2)

Components	Temp. (in °C)	Press. (in bar)	Flow (kg/sec)	Enthalpy (in kW)	Exergy (in kW)
Nitrogen 4 (N4)	-88.5	45.4	18.11	-2682.9	7190.3
CETout	-177.3	5.7	14.51	-3203.9	4789.8
Nitrogen 1 (N1)	-90.2	5.6	-14.51	1788.7	-3028.1
CETin	-104.2	45.2	-13.38	2283.2	-5516.4
JTin	-173.7	45.4	-4.73	1814.9	-3262.2
Total				0	173

Table A2.9 The in- and outgoing flows of compressor 1 (C1)

Components	Temp. (in °C)	Press. (in bar)	Flow (kg/sec)	Enthalpy (in kW)	Exergy (in kW)
Compressor1in (C1in)	26.0	5.5	26.43	-10.8	4577.2
Work input				6000.8	6000.8
Compressor1out (C1out)	26.0	30.6	-26.43	180.4	-8612.8
Heat transfer to environ.				-6170.5	0
Total				0.1	1965.2

Table A2.10 The in- and outgoing flows of the steam turbine (ST2)

Components	Temp. (in °C)	Press. (in bar)	Flow (kg/sec)	Enthalpy (in kW)	Exergy (in kW)
Steam in (ST1in)	420.0	40.00	5.99	-76119	7797.9
Steam out (STout)	36.0	0.05	-5.99	82120	-775.9
Work output				-6001	-6000.8
Total				0	1021.1

Table A.2.11 The in- and outgoing flows of compressor 2 (C2)

Components	Temp. (in °C)	Press. (in bar)	Flow (kg/sec)	Enthalpy (in kW)	Exergy (in kW)
Compressor2in (C2in = C1out)	26.0	30.6	26.43	-180.4	8612.8
Work input				672.2	672.2
Compressor2out (C2out)	33.0	37.4	-26.43	18.8	-9079.9
Heat transfer to environ.				-509.8	0
Total				0.8	205.1

Table A2.12 The in- and outgoing flows of compressor 3 (C3)

Components	Temp. (in °C)	Press. (in bar)	Flow (kg/sec)	Enthalpy (in kW)	Exergy (in kW)
Compressor3in (C3in)	33.0	37.4	18.11	-12.9	6221.1
Work input				461.9	461.9
Compressor3out (C3out)	33.0	45.8	-18.11	46.4	-6541.8
Heat transfer to environ.				-495.5	0
Total				0.1	141.1

Table A2.13 The in- and outgoing flows of the hot expansion turbine (HET)

Components	Temp. (in °C)	Press. (in bar)	Flow (kg/sec)	Enthalpy (in kW)	Exergy (in kW)
HETin	-3.0	37.2	8.32	-339.6	2867.0
HETout	-103.5	5.0	-8.32	1145.7	-1819.8
Work output				-806.1	-806.1
Total				0	241.1

Table A2.14 The in- and outgoing flows of the cold expansion turbine (CET)

Components	Temp. (in °C)	Press. (in bar)	Flow (kg/sec)	Enthalpy (in kW)	Exergy (in kW)
CETin	-104.2	45.2	13.38	-2283.2	5516.4
CETout	-177.3	5.7	-13.38	2955.3	-4416.4
Work output				-672.2	-672.2
Total				-0.1	428.4

Table A2.15 The in- and outgoing flows of the JT valve

Components	Temp. (in °C)	Press. (in bar)	Flow (kg/sec)	Enthalpy (in kW)	Exergy (in kW)
JTin	-173.7	45.4	4.73	-1814.9	3262.2
Liquefied nitrogen (LQN)	-177.1	5.8	-4.73	1814.9	3188.3
Total				0	73.9

Appendix 3: Relations for the detergency

Relations of the detergency for the different test pieces. The relations presented are the best fittings that could be obtained. The standard deviation is given with the calculated parameters. The validity of the relations is restricted to the temperature range from 30 to 60°C and the detergent quantity range from 18 to 108 grams.

Test piece 1 :

$$Det_1(T, D) = (a_1 + b_1 \cdot T) \cdot (1 - f_1 \cdot e^{-g_1 \cdot D}) \quad (A3.1)$$

a₁	49.3 ± 6%	[--]
b₁	0.551 ± 11%	[1/°C]
f₁	0.845 ± 12%	[--]
g₁	3.4 · 10 ⁻² ± 19%	[1/gr]

Test piece 2 :

$$Det(T, D) = (a_2 + b_2 \cdot T) \cdot (1 - f_2 \cdot e^{-g_2 \cdot D}) \quad (A3.2)$$

a₂	54.9 ± 6%	[--]
b₂	0.551 ± 13%	[1/°C]
f₂	1.25 ± 18%	[--]
g₂	5.23 · 10 ⁻² ± 18%	[1/gr]

Test piece 3 :

$$Det(T, D) = a_3 + b_3 \cdot T \cdot D \quad (A3.3)$$

a₃	13.2 ± 16%	[--]
b₃	6.9 · 10 ⁻³ ± 28%	[1/°C · gr]

Test piece 4 :

$$Det(D) = a_4 + b_4 \cdot D \quad (A3.4)$$

a₄	30.2 ± 7%	[--]
b₄	0.256 ± 13%	[1/gr]

Test piece 5 :

$$\begin{aligned}
 Det(D) &= a_5 + b_5 \cdot D && \{15 < T < 30\} \\
 Det(T, D) &= f_5 \cdot T + g_5 \cdot D && \{T > 30 \wedge D < 72\} \\
 Det(T) &= h_5 \cdot T && \{T > 30 \wedge D \geq 72\}
 \end{aligned}
 \tag{A3.5}$$

a₅	23.2 ± 13%	[--]
b₅	0.154 ± 27%	[1/gr]
f₅	0.461 ± 14%	[1/°C·gr]
g₅	0.342 ± 25%	[1/gr]
h₅	1.251 ± 18%	[1/°C·gr]

Test piece 6 :

$$Det(T, D) = b_6 \cdot T + f_6 \cdot (1 - e^{-g_6 \cdot D})
 \tag{A3.6}$$

b₆	0.295 ± 14%	[1/°C·gr]
f₆	72.9 ± 2%	[--]
g₆	7.25 · 10 ⁻² ± 6%	[1/gr]

Appendix 4: Calculation of the additional irreversibility connected with construction and dismantling

A4.1 Additional irreversibility due to the construction

The quantity of used material for the different alternatives is calculated according to Centre for Energy Conservation and Environmental Technology (1982).

Table A4.1 Amount of used material for the different alternatives (in tons)

Component	alt. 1	alt. 2	alt. 3	alt. 4
Steel	3520	7190	357	1513
PUR	329	1025	52	309
HDPE	803	2031	84	494
Sand digging	85000	390000	12000	130000
Sand supply	43000	185000	5000	64000
Sand returned	23000	179000	5000	60000
Sand removal	62000	209000	7000	70000

Using the data of Table 7.3 the irreversibility associated with the material use is calculated as shown in Table A4.2.

Table A4.2 Amount of irreversibility associated with the material use for the different alternatives (in TJ)

Component	alt. 1	alt. 2	alt. 3	alt. 4
Steel (sec)	35.2	71.9	3.6	15.1
PUR (prim)	23.4	72.8	3.7	21.9
HDPE (prim)	25.7	65.0	2.7	15.8
Sand digging	4.4	20.3	0.6	6.8
Sand supply	18.1	78.1	2.1	27.0
Sand return	1.2	9.3	0.3	3.1
Sand removal	26.2	88.2	3.0	29.5
Inv. irr.	134.2	405.5	15.9	119.3

However, in the case of alternative 2 and 4 a gas distribution system has been saved. So the irreversibility associated with this investment, displayed in Table A4.3, has to be subtracted.

Table A4.3 Amount of used material and irreversibility associated with the its use for the gas distribution system for the alternatives 2 and 4

Component	alt. 2		alt. 4	
	mass (in ton)	irr. (in TJ)	mass (in ton)	irr. (in TJ)
PVC (sec)	249	3.5	51	0.7
HDPE (sec)	106	1.5	38	0.5
Inv. irr.		5.0		1.2

A4.2 Additional irreversibility due to the dismantling

An additional irreversibility is caused by the burning of the HDPE and PUR in the waste incineration plant. It is taken into account that the power generation of this plant saves 0.64 MJ of exergy per MJ exergy of PUR and HDPE.

Table A4.4 Amount of irreversibility associated with the burning in the waste incineration plant for the different alternatives

Components	alt. 1	alt. 2	alt. 3	alt. 4
PUR	3.3	10.3	0.5	3.1
HDPE	14.4	36.5	1.5	8.9
Total	17.7	46.7	2.0	12.0

A4.3 The additional irreversibility

Table A4.5 Amount of additional irreversibility for the different alternatives

Components	alt. 1	alt. 2	alt. 3	alt. 4
Inv. irr. heat distribution	134.2	405.5	15.9	119.3
Inv. irr. gas distribution		-5.0		-1.2
Irr. waste incineration	17.7	46.7	2.0	12.0
Additional irr. (I_{inv})	152	447	18	130

Nomenclature

inv. = investment
 irr. = irreversibility
 prim = primary

sec = secondary

References

Centre for Energy Conservation and Environmental Technology, 1982, *Guide for optimal space heating of houses* (in Dutch), Delft, the Netherlands.

Appendix 5: Comparison with experimental results

The formulas used are compared with experimental results of the manufacturer for the heat exchanger SP 15 N (KVM, 1993), which is a heat exchanger with three inner tubes. The specifications are listed in Table A5.1

Table A5.1 Specifications of the SP 15 N in meters

Specification	meters
inner diameter (D_1)	$8.4 \cdot 10^{-3}$
outer diameter (D_2)	$2.77 \cdot 10^{-2}$
thickness of the inner tube (d_l)	$8.0 \cdot 10^{-4}$
length (L)	6.4

However, no sufficient experimental information about balanced operation is available for this heat exchanger, so the formulas have to be rewritten for the unbalanced situation.

For the unbalanced counter flow heat exchanger formula 8.1(a) is rewritten

$$\dot{i}^{\Delta T} = T_0 \left[C_1 \ln \frac{T_{1,\text{out}}}{T_{1,\text{in}}} + C_2 \ln \frac{T_{2,\text{out}}}{T_{2,\text{in}}} \right] \quad (\text{A5.1})$$

where $C_1 = \dot{m}_1 c_p$, $C_2 = \dot{m}_2 c_p$ and with help of the energy balance

$$T_{2,\text{out}} = T_{2,\text{in}} - \frac{C_1}{C_2} (T_{1,\text{out}} - T_{1,\text{in}}) \quad (\text{A5.2})$$

and the effectiveness

$$\varepsilon = \frac{C_1 (T_{1,\text{in}} - T_{1,\text{out}})}{C_{\min} (T_{1,\text{in}} - T_{2,\text{in}})} \quad (\text{A5.3})$$

where $C_{\min} = C_1$, when $C_2 > C_1$ and $C_{\min} = C_2$, when $C_2 < C_1$.

$T_{2,\text{out}}$ and $T_{2,\text{in}}$ can be replaced as shown in (A5.2) and (A5.3) to rewrite (A5.1) to

$$\dot{i}^{\Delta T} = T_0 \left[C_1 \ln \frac{T_{1,\text{out}}}{T_{1,\text{in}}} + C_2 \ln \left(1 - \frac{C_1}{C_2} \left(\frac{C_1}{\varepsilon \cdot C_{\min}} + \frac{T_1}{(T_{1,\text{out}} - T_1)} \right)^{-1} \right) \right] \quad (\text{A5.4})$$

with

$$\varepsilon = \frac{1 - e^{-N_{tu} \cdot x}}{1 - \frac{C_{\min}}{C_{\max}} e^{-N_{tu} \cdot x}}$$

where $N_{tu} = \frac{\alpha A}{C_{\min}}$ and $x = 1 - \frac{C_{\min}}{C_{\max}}$. N_{tu} is calculated with the help of formula 8.7. For the pressure losses formula (8.10b) can be used by filling in the correct mass flow. The irreversibility of experimental results calculated with (A5.1) is compared with the irreversibility calculated from the theory with formula A5.4 in Table 8.2 for $\dot{m}_1 = 5$ and $\dot{m}_2 = 4$ litre a minute. For the mechanical losses a balanced operation of $\dot{m}_1 = \dot{m}_2 = 7$ litre a minute is taken.

Table A5.2: Irreversibilities obtained by experimental results and theory in Watts

component	experimental	theoretical
thermal	1039 ± 50	1031
mechanical	1.8 ± 0.1	1.0

As can be seen in case of the thermal component the experimental and thermal results match perfectly, while in the case of the mechanical losses there is a big difference. Taking into account the bendings of the connecting tubes, which can be seen in Figure 8.1, for the theoretical values has a marginal effect. So this effect is not taken into account. Further contact with the manufactures has learned that the experimental values for the pressure drops are mean values and there are great deviations between different heat exchangers. Probably, these are the result of irregular position of the inner tubes in the outer tube.

References

KVM, 1993, *Vandvarmer SP 15 N* (in Danish), KVM-CONHEAT A/S, Vissenbjerg, Danmark.

Appendix 6: Environmental effects caused by emissions

The emissions are given per environmental effect in the normalised values, i.e. the share of the average emission of each European individual. Unless else wise notified, the emissions are airborne emissions.

Green house effect

Substance	LC PS cups	LC porc. mug
CO ₂	0.0042	0.00227
methane	1.96e-06	3.6e-06
N ₂ O	5.124-06	9.4e-05
total	0.00421	0.00240

Depletion of the ozone layer

No emissions

Acidification

Substance	LC PS cups	LC porc. mug
ammonia	1.4e-07	2.01e-06
HCl	4.8e-06	
HF	2.5e-07	
NO _x	0.00189	0.00107
SO ₂	0.00379	0.000584
Total	0.0057	0.00166

Eutrofication

Substance	LC PS cups	LC porc. mug
ammonia	7.69e-08	1.04e-06
NO _x	0.00104	0.000586
COD (w)	1.08e-05	0
NH ₄ ⁺ (w)	6.1e-06	0
phosphate (w)	0	0.001178
total	0.00105	0.00177

w = waterborne emission

Heavy metals

Substance	LC PS cups	LC porc. mug
heavy metals	1.062e-05	0
Ba (w)	0	2.4e-07
Cd (w)	1.18e-06	0
Cr (w)	0.0000388	0
Cu (w)	1.55e-07	0
Hg (w)	9.6e-07	0
Pb(w)	0.000031	1.38e-06
total	8.27e-05	1.62e-06

w = waterborne emission

Carcinogenic

substance	LC PS cups	LC porc. mug
benzene	8.1e-07	0
benzo[a]pyrene	2.6e-07	0
C _x H _y aromatic	7.87e-10	0
ethylbenzene	1.34e-07	0
fluoranthene	5.28e-06	0
total	6.48e-06	0

Winter Smog

substance	LC PS cups	LC porc. mug
carbon black	1.68e-07	0
dust (SPM)	0.000025	5.96e-05
SO2	0.00452	0.000698
total	0.00455	0.000758

Summer Smog

substance	LC PS cups	LC porc. mug
aldehydes	1.70e-07	1.0e-06
benzene	8.52e-06	0
C _x H _y	0.00180	0.0052
C _x H _y aromatic	1.65e-08	0
ethylbenzene	1.36e-06	0
methane	9.2e-07	1.68e-05
styrene	1.0e-06	0
toluene	0.000015	0
total	0.000182	0.00522

Pesticides

No emissions

PUBLICATIONS¹

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¹ Note: the chapters referring to the listed publications are no exact copy. Some minor changes have taken place to fit them into the underlying work

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About the author

René Cornelissen was born on August 28, 1969 in Bakkeveen (Friesland), the Netherlands. After attending secondary school Ichtus College in Drachten he went to the University of Groningen where he acquired a master degree in Applied Physics in 1993. His master thesis was about material substitution in automobiles and the associated environmental aspects. This study was performed at the Centre for Energy and Environmental Studies. Hereafter he started his Ph.D. study at the Chair of Energy technology, Laboratory of Thermal Engineering, Department of Mechanical Engineering. In this period he attended and presented his research at several conferences and symposia. Now, he is an independent consultant in the area of exergy analysis.